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Encyclopedia of Ocean Engineering

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Encyclopedia of Ocean Engineering

With 1737 Figures and 126 Tables

 Springer

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Preface

Although the planet on which we live is called *Earth*, more than 70% of her surface is covered by the ocean. With the constant increase of human population, people are striving to develop more living space in the ocean and take advantage of its resources. However, there is no doubt that during this process humans will meet many technological challenges and must find ways to protect the ocean environment, on purpose of reaching a sustainable and harmonious lifestyle with the ocean. To achieve this goal, we must know the ocean, gather sufficient knowledge about the ocean, and treat the ocean in a respectful and cautious manner, because otherwise, any violent and reckless exploration behavior that harms the ocean will lead to unaffordable consequences to humanity. Therefore, to understand the ocean and to master the fundamental technology relating to its development is of critical importance, especially this year when humanity is truly beginning to experience the obvious temperature rise due to global warming.

The writing of this encyclopedia, *Ocean Engineering*, has three aims. Firstly, to provide a useful introductory reading reference for people interested in, but not delving into, the field of ocean engineering. Secondly, to provide necessary scientific and technological knowledge for people who are delving into some specific aspects in ocean engineering and want to do interdisciplinary extensions. Thirdly, to contribute within our scope in helping people find sustainable and harmonious ways to develop and protect the ocean.

This encyclopedia discusses the main categories of fundamental knowledge on ocean engineering, which are extensively covered in 21 parts, including the parts on conventional naval architecture and offshore engineering such as ship design and offshore decommissioning; the parts on ocean exploration such as underwater engineering, AUV/ROV, and polar engineering; the parts on harvesting of ocean resources such as deepwater mining and on utilizing the ocean space such as very large floating structures (VLFS) and aquaculture engineering; and the parts on sustainable ocean development such as offshore renewable energy, green shipping, and ocean environmental protection. This encyclopedia contains numerous illustrations and examples to make it easy to understand.

With the publication of this encyclopedia, *Ocean Engineering*, the editorial team wants to thank all those who have contributed to it. Please forgive us for not being able to name all the section editors here since they are specified in the next page, but we would like to convey sincere appreciations to our excellent section editors (SEs). You have done great work, and the time and effort spent

in organizing and reviewing entries is much appreciated. Besides, please let us acknowledge every author's contribution. We would not be able to begin without the help of all our brilliant authors. Finally, we are also very grateful to the Springer Nature supporting team. Thank you for always being there for this encyclopedia. The experience of working together in this 3-year project leaves an indelible impression on us.

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May 2022

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Weicheng Cui joined Westlake University (WU) as a chair professor in September 2018, 1 month before its founding ceremony. The founding mission of WU is to become a global leader in frontier scientific research and a reformer in higher education in China. Dr. Cui's current research interests include (1) development of robotic fish-type submersibles and (2) fundamental research on the generalization of general system theory (GST) into theory of everything (TOE) for complex systems.

He got his B.Sc. from the Department of Engineering Mechanics at Tsinghua University in 1986 and his Ph.D. from the University of Bristol, England, in 1990. From 1990 to 1993, he did his postdoctoral research in the Department of Aerospace Engineering at the University of Bristol. He made some contributions in the aspects of measurement of interlaminar shear strength, nonlinear effect and size effect of the delamination strength, and the delamination mechanism for composite materials.

From February 1993 to May 1999, he worked at China Ship Scientific Research Center (CSSRC), and from June 1999 to September 2002, he was appointed as the Changjiang Professor at Shanghai Jiao Tong University. During these two periods, he mainly engaged in research in ship structural mechanics. He had made some contributions in the areas of the prediction of the ultimate strength of intact and damaged ship structures, fatigue strength assessment of ship structures, and reliability-based analysis and design of ship structures.

From October 2002 to March 2013, he worked at CSSRC again. Dr. Cui was the project leader and first deputy chief designer of Jiaolong deep manned submersible. He made some contributions in the application of multidisciplinary design optimization method and the establishment of a rational design standard for the manned cabin. Because of his outstanding contribution to the Jiaolong manned submersible development project, he won the Blancpain Hans Hass Fifty Fathoms Award, the First Prize in National Science and Technology Progress, and many honorary titles such as “The National Excellent Science and Technology Workers,” “The Deep Diving Hero” awarded by the Central Committee and the State Council, and the first batch of scholars of the National Innovation Talent Award.

From March 2013 to September 2018, he worked at Shanghai Ocean University, and from September 2018 onward, he has been working at Westlake University. He served as the project leader and chief designer of the Rainbowfish Challenging the Challenger Deep project from 2013 to 2020. In 2016, he was named one of “China’s Top Ten Science Stars” by *Nature*.

He is currently an editorial board member of six international journals, and the associate editor-in-chief of both *Shipbuilding of China* and the *Journal of Ship Mechanics*. He has published more than 400 papers in SCI and EI journals, being selected in Elsevier’s list of China’s Highly Cited Researchers continuously from 2017 to 2019.

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Professor Fu has been selected as the Academician of the Norwegian Academy of Technological Sciences in 2020, and was granted China National Funds for Distinguished Young Scientists in 2018.

Professor Fu is now the vice director of the State Key Laboratory of Ocean Engineering at Shanghai Jiao Tong University, and the dean of the Research School of Polar and Deep Sea Technologies (joint research school between SJTU and Second Institute of Oceanography). He is also the associate editor of *Marine Structures* and the *Journal of Offshore Mechanics and Arctic Engineering*.

Professor Fu's research interests include hydroelasticity of floating bridges/tunnels and large-scale fish cage, vortex-induced vibration of flexible structures, experimental methods of force feedback control, and hybrid full-scale model testing.

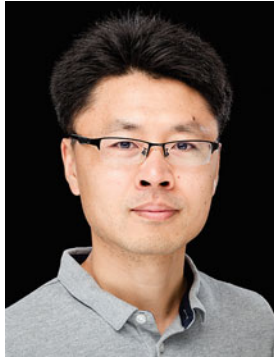
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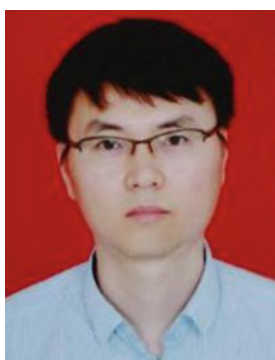
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A

Accidental Loads

- ▶ [Offshore Structure Design Under Ice Loads](#)

Acoustic Correlation Current Profiler (ACCP)

- ▶ [Underwater Information Sensing Technology](#)

Acoustic Deep Tow System

- ▶ [Deep Tow System](#)

Acoustic Doppler Current Profiler (ADCP)

- ▶ [Underwater Information Sensing Technology](#)

Acoustic Modems

- ▶ [Underwater Acoustic Sensor Network](#)

Acoustic Positioning System (APS)

- ▶ [Integrated Navigation](#)

Acoustic Synthetic Baseline (ASBL)

- ▶ [Long Baseline Underwater Acoustic Location Technology](#)

Acoustic Towed System

- ▶ [Deep Tow System](#)

Active Sonar

- ▶ [Sonar Technology](#)

Ad Hoc Network

- ▶ [Underwater Acoustic Sensor Network](#)

Advanced Control

- ▶ [Intelligent Control Algorithms in Underwater Vehicles](#)

AEM - Anion-Exchange Membrane

- ▶ [Salinity Gradient Power Conversion](#)

Aerodynamic Drag

- ▶ [Wind Tunnel Test](#)

Air Emissions

- ▶ [Impact of Maritime Transport](#)
- ▶ [Ship Propulsion System](#)

AIS (Automatic Identification System)

- ▶ [Ship Navigation System](#)

Alarm Pheromone-Assisted Ant Colony System (AP-ACS)

- ▶ [Obstacle Avoidance Technology for Underwater Vehicle](#)

ALS – Accidental Limit States

- ▶ [Design of Renewable Energy Devices](#)

Alternative Fuel

- ▶ [Ship Construction and Operation Impact on Energy Efficiency](#)

Alternative Fuel for Ship Propulsion

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Synonyms

[Fuel cell](#); [Hydrogen](#); [Liquefied natural gas \(LNG\)](#); [Liquefied petroleum gas \(LPG\)](#); [Methanol](#)

Definition

Alternative fuels for ship propulsion mean fuels which serve, at least partly, as a substitute for conventional fossil oil sources, normally heavy fuel oil (HFO), in the energy supply to shipping and which have the potential to contribute to its decarbonization and enhance the environmental performance of the shipping sector.

Driving Force for Alternative Fuels

Ships are a relatively fuel-efficient means of moving cargoes: for equivalent weights and distances, carbon dioxide emissions from shipping are considerably lower than air freight and road transport. At its best, emissions from shipping also undercut rail transport. This can be used as a rough proxy for transport costs and helps to illustrate the crucial role of international shipping in supporting global commerce, carrying as it does an estimated 90% of world trade. However, at the same time, the huge shipping quantity also makes shipping such a significant contributor to air pollution in terms of SO_x and NO_x emission, as well as

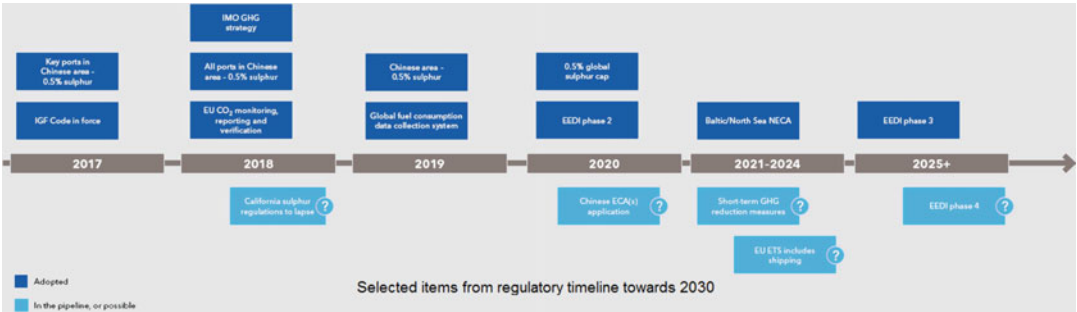
greenhouse gas emissions (accounting for 3% of global greenhouse gas emissions).

International Maritime Organization (IMO) and national environmental agencies have realized the impact of emission from shipping and therefore issued regulations and policies that drastically reduce emissions emanating from marine sources (IMO 2016; EU 2014, etc.). In particular, with the decision of IMO to limit the sulfur content of ship fuel from 1 January 2020 to 0.5% worldwide, more than 70,000 ships will be affected by the regulation. In addition, stricter limits on sulfur dioxide (SOx) emissions are already in place in Emission Control Areas (ECAs) in Europe and the Americas, and new control areas are being established in ports and coastal areas in China.

While these practical challenges related to sulfur reduction are knocking at the door, there is an accelerating worldwide trend toward pushing down CO₂ and nitrogen oxides (NOx) and particle emissions. In April 2018, IMO’s Marine Environment Protection Committee (MEPC), at its 72nd session, adopted an initial strategy on the reduction of greenhouse gas emissions from ships, setting out a vision to reduce GHG emissions from international shipping and phase them out, as soon as possible in this century. The vision confirms IMO’s commitment to reducing GHG emissions from international shipping, and, more specifically, under the identified “levels of ambition,” the initial strategy envisages for the first time a reduction in total GHG emissions from international shipping which, it says, should peak as soon

as possible and to reduce the total annual GHG emissions by at least 50% by 2050 compared to 2008 while, at the same time, pursuing efforts toward phasing them out entirely. The strategy includes a specific reference to “a pathway of CO₂ emissions reduction consistent with the Paris Agreement temperature goals.” The initial strategy represents a framework setting out the future vision for international shipping, the levels of ambition to reduce GHG emissions and guiding principles, and includes candidate short-, mid-, and long-term further measures with possible timelines and their impacts on States. A time map of actions from maritime authorities to promote greener shipping industry is presented in Fig. 1 (DNV GL 2019). The strategy also identifies barriers and supportive measures including capacity building, technical cooperation, and research and development (R&D).

Many ship operators, with present-day propulsion plants and marine fuels, cannot meet these new regulations without installing expensive exhaust after-treatment equipment or switching to more expensive marine fuels (such as MGO) or alternative fuels with properties that reduce engine emissions below mandated limits, all of which impact bottom-line profits (IMO 2016). The impact of these new national and international regulations on the shipping industries worldwide has brought alternative fuels to the forefront as a means for realizing compliance. The alternative fuels industry has grown dramatically for both liquid and gaseous fuels. In this context, the gradual adoption of alternative fuels



Alternative Fuel for Ship Propulsion, Fig. 1 Shipping becomes greener and more complex (Courtesy of DNV GL 2019)

by shipping would have a significant positive immediate environmental impact.

Basic Introduction to Alternative Fuels

There is a long list of alternative fuels that can be used in shipping. Among the proposed alternative fuels for shipping, LNG, methanol, hydrogen (particularly for use in fuel cells), and liquefied petroleum gas (LPG) are identified as the most promising solutions. All these fuels are virtually sulfur-free and can be used for compliance with sulfur content regulations, which is considered as the most urgent challenge for shipping. They can be used either in combination with conventional, oil-based marine fuels, thus covering only part of a vessel's energy demand, or to completely replace conventional fuels for some ships. The type of alternative fuel selected and the proportion of conventional fuel substituted will have a direct impact on the vessel's emissions, including GHG, NO_x, and SO_x.

Each of these alternative fuels has advantages and disadvantages from the standpoint of the shipping industry. One common challenge, however, posed by the adoption of most alternative fuels are their physicochemical characteristics, typically with associated low flashpoints, higher volatilities, different energy content per unit mass, and, in some cases, even toxicity. The adoption and entry into force of the draft International Code of Safety for Ships using gases or other low-flashpoint fuels (IGF Code), along with proposed amendments to make the code mandatory under SOLAS, by IMO Maritime Safety Committee 95 (MSC 95), on 11 June 2015, was a decisive step forward in addressing those challenges, at the regulatory level. The IGF Code includes mandatory provisions for the arrangement, installation, control, and monitoring of machinery, equipment, and systems using low-flashpoint fuels, such as liquefied natural gas (LNG), to minimize the risk to the ship, its crew, and the environment, having regard to the nature of the fuels involved. LNG has been the first focus of the IGF Code; however, provisions for methyl/ethyl alcohols, fuel cells, and low-flashpoint oil fuels are still being drafted as interim guidelines.

Key Applications

Liquefied Natural Gas (LNG)

LNG is a cryogenic fuel that is maintained at approximately -260°F (-162°C) at atmospheric pressure. The advantage of cooling and liquefying the fuel is that the volume is decreased approximately 600 times as compared to the gas. This advantage improves the energy density significantly for LNG. As a result, when compared to diesel fuel, LNG has about 2/3 as much energy on a volume basis and almost 90% as much energy on a weight basis. Natural gas consists primarily of methane, and typical composition is presented in Table 1 (Kolwzan and Narewski 2012).

The LNG fuel has a higher hydrogen-to-carbon ratio compared to oil-based fuels, which results in lower specific CO₂ emissions (kg of CO₂/kg of fuel). In addition, LNG is a clean fuel, containing no sulfur; this eliminates the SO_x emissions and almost eliminates the emissions of particulate matter. Additionally, the NO_x emissions are reduced by up to 90% due to reduced peak temperatures in the combustion process. Unfortunately, one of drawbacks of the use of LNG is the so-called methane slip during the valves overlap period for most four-stroke gas engines that is leading to the increase of the emissions of methane (CH₄), hence reducing the net global warming benefit from 25% to about 15%.

LNG as a liquid is not flammable or explosive. As any gas it has a flammability range. This range for LNG gas is between 5% and 15% when mixed with air. An explosion can only occur when the gas is in an enclosed space with air, the mixture is between 5 and 15%, and an ignition source is present. As with any flammable substance, proper design, regulations, and personnel training are needed to maintain a safe environment. Another major hazard for LNG as marine is that a spill of a cryogenic-like LNG

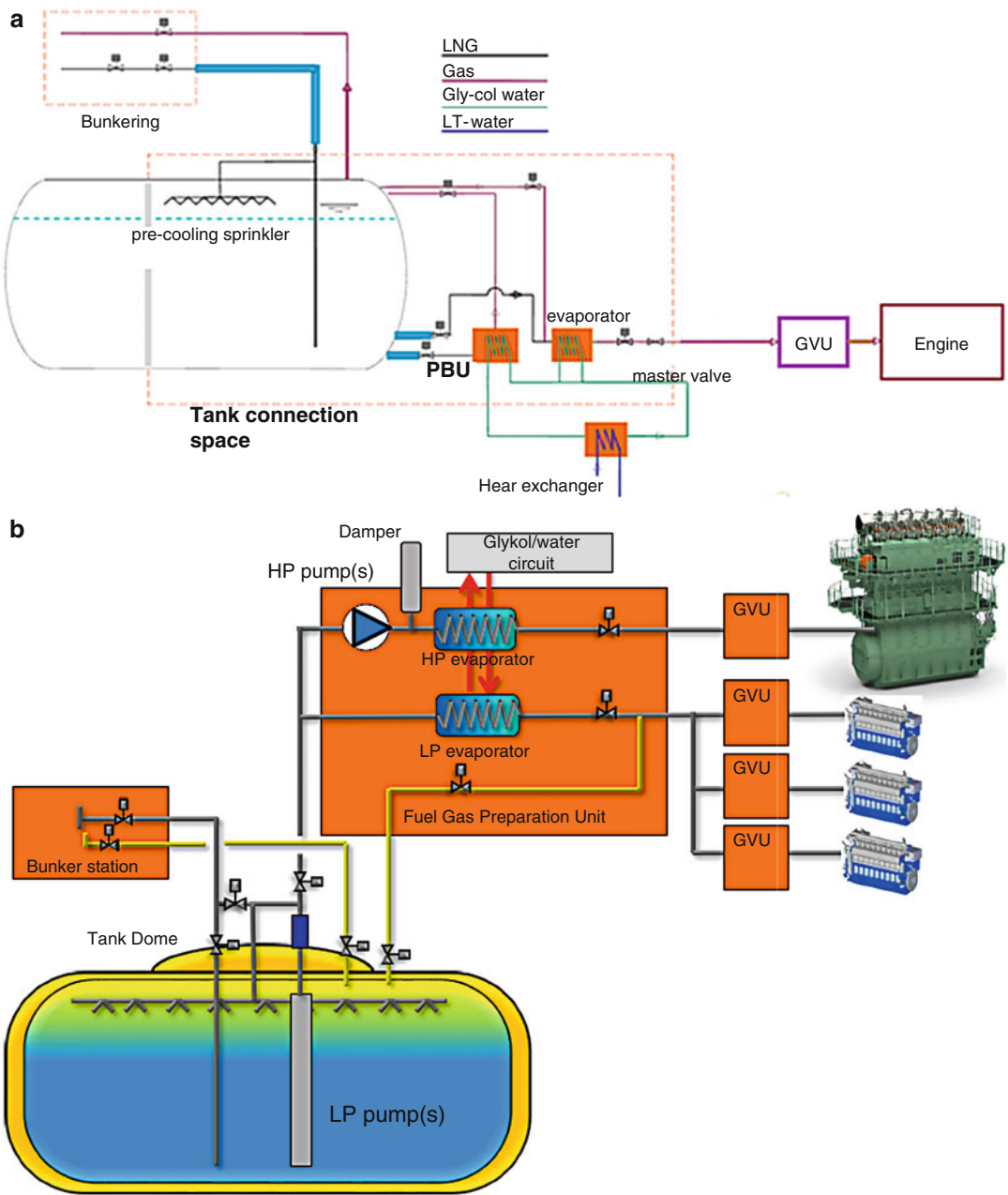
Alternative Fuel for Ship Propulsion, Table 1 Typical composition of natural gas in %

Methane	Ethane	Propane	Butane	Nitrogen
94	4.7	0.8	0.2	0.3

could result in structural damage of ships’ hull, deck, and other steel structures near the ship. Serious injuries could occur when LNG gets in contact with the human body.

An LNG fuel system onboard ships normally can be divided into four subsystems, namely,

LNG bunkering system, LNG storage system, fuel gas supply system (FGSS), and gas consumption installation system. Figure 2 shows the typical arrangement of low-pressure and high-pressure LNG fuel system onboard ships (McGill et al. 2013; DNV GL 2019).



Alternative Fuel for Ship Propulsion, Fig. 2 (a) Typical LNG fuel system (with low-pressure gas supply) onboard ships. (b) Typical LNG fuel system (with high-pressure gas supply) onboard ships

LNG is typically stored in highly insulated, spherical, or cylindrical tanks at low pressures (normally 1.05~5 bars). Fitting these tanks on a ship is quite feasible but requires some protection measures to reduce the risk of gas release due to collision with other ships or grounding. A detailed summary of various LNG fuel storage tanks is presented in Fig. 3.

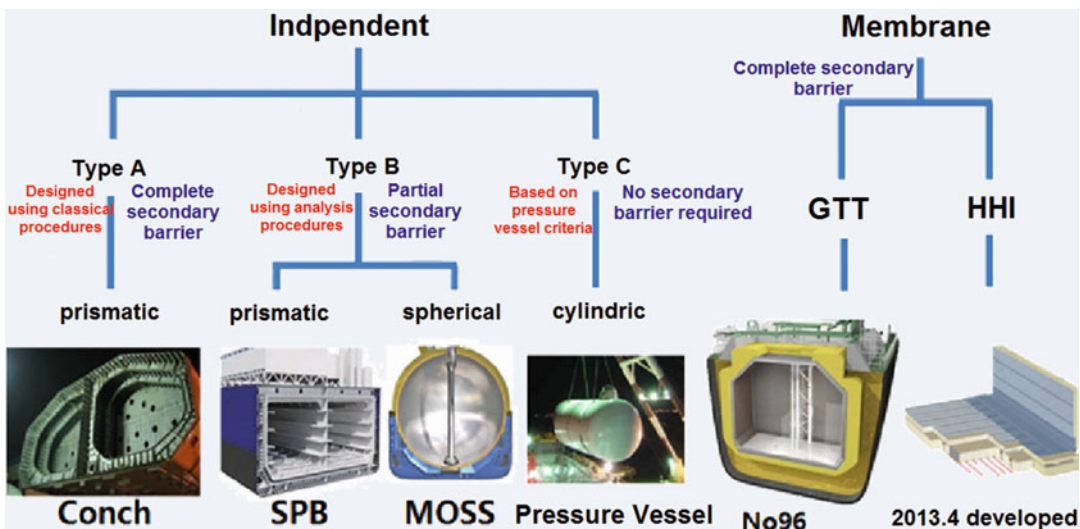
Depending on the chosen gas engines, both high-pressure (HP) and low-pressure (LP) FGSS are available. For LP FGSS, LNG extracted from LNG fuel tank by pump or pressure build-up unit, and then vaporized and heated by glycol water, finally supplied to gas engines. For HP FGSS, LNG is firstly pressurized and then vaporized, which this is quite different from LNG carriers, because pressurizing liquid is much easier than gas.

Gas engines, gas turbines, and LNG storage and processing systems have been available for land installations for decades. LNG sea transport by LNG carrier also has a history going back to the middle of the last century. Developments to use LNG fuel in general shipping began early in the current century. Today, the technology required for using LNG as ship fuel is readily available. Piston engines and gas turbines, several LNG storage tank types, and process equipment are also commercially available.

Redundancy of fuel gas system shall be considered in particular for pure LNG-fueled ships; IGF Code stipulates that for single fuel installations, the fuel supply system shall be arranged with full redundancy and segregation all the way from the fuel tanks to the consumer, so that a leakage in one system does not lead to an unacceptable loss of power. In this regard, one pure gas engine with two fuel supply systems or two pure gas engines that each with one independent fuel supply system are both acceptable.

Conversion from diesel to LNG propulsion is possible, but the LNG is mainly relevant for newly constructed ships, since substantial modification of engines, the related piping, and allocation of extra storage capacity is required.

LNG presents hazards that are different than conventional marine fuels, like heavy fuel oil (HFO) and marine gas oil (MGO). If released at normal ambient temperatures and pressures, it will form a flammable vapor, so the release of LNG or natural gas should be prevented at all time. Furthermore, in its liquid phase, LNG is cold enough that it can cause ordinary steel to become brittle and crack, so any contact with steel structures and decks should be avoided. Because of these hazards and others that can occur, safety and the prevention of leakage need



Alternative Fuel for Ship Propulsion, Fig. 3 Different LNG fuel storage tank types according to IGF Code

to be among the primary objectives in the design, construction, and operation of LNG-fueled ships.

The three primary safety objectives for LNG fuel system onboard a ship are as follows:

- Prevent the occurrence of any hazardous release of gas or liquid.
- In the event of a release, prevent or contain any hazardous situations.
- If a hazardous incident does occur, limit the consequences and harmful effects.

A potential disadvantage to using LNG is space. Since gas weighs more, volume-wise it requires more space as compared to bunker oil. The farther the journey, the equally larger amount of storage space is required. So far, tanks are designed to be built in the cargo spaces of the ships for using gas as fuel. This is a major setback for the ship operators in terms of freight earned by the cargo. Engineers and architects are working toward developing systems that would make room for storing LNG. This could be anywhere on the vessel, above deck, in the superstructures, beneath the cargo containers, astern of the vessel, etc., and this would also call for extra insulation, piping, and steelwork as far as construction of the vessels is concerned.

LNG fuel surely holds a promising future in the shipping industry. However, only time can tell as to how well it becomes an integral part of the shipping industry in the days to come.

Methanol/Methyl Alcohol

Methanol, with the chemical structure CH_3OH , is the simplest alcohol with the lowest carbon content and highest hydrogen content of any liquid fuel. Methanol is a liquid between 176 and 338 K (-93 to $+65$ °C) at atmospheric pressure; it is a basic building block for hundreds of essential chemical commodities that contribute to our daily lives, such as building materials, plastic packaging, paints, and coatings. It is also a transport fuel and a hydrogen carrier for fuel cells (SSPA 2015).

Methanol can be produced from several different feedstock resources, mainly natural gas or

coal, but also from renewable resources like black liquor from pulp and paper mills, forest thinning or agricultural waste, and even directly from CO_2 that is captured from power plants. When produced from natural gas, a combination of steam reforming and partial oxidation is typically applied, with an energy efficiency up to about 70% (defined as energy stored in the methanol versus energy provided by natural gas). Renewable methanol is produced from pulp mill residue in Sweden, waste in Canada, and from CO_2 emissions at a small commercial plant in Iceland.

As an alternative liquid fuel, methanol is clean-burning, biodegradable, and sulfur-free. The only PM produced from burning methanol is from the pilot fuel (usually diesel). Methanol is able to meet the increasingly stringent emission reduction requirement such as the global sulfur limit of 0.5% by 2020 for the international shipping and acting as potential candidate to meet further greenhouse gas reduction targets. Major OEMs have shown methanol can meet Tier III NOx requirements with the aid of emulsification.

There are two main options for using methanol as fuel in conventional ship engines: in a two-stroke diesel-cycle engine or in a four-stroke, lean-burn Otto-cycle engine. Similar to LPG, only a single two-stroke diesel engine is currently commercially available, the MAN ME-LGI series, which is now in operation on methanol tankers. Wärtsilä four-stroke engines are in operation onboard the passenger ferry *Stena Germanica*. Another possibility would be to use methanol in fuel cells. A test installation has been running on the Viking Line ferry *MS Mariella* since 2017.

Methanol is a liquid fuel and can be stored in standard fuel tanks for liquid fuels, with certain modifications to accommodate its low-flashpoint properties and the requirements currently under development for the IGF Code at the IMO. Fuel tanks should be provided with an arrangement for safe inert gas purging and gas freeing.

Typical methanol fuel system onboard a ship can be divided into the following system elements:

- Bunkering of methanol
- Storage of methanol onboard

Hydrogen is an energy carrier and a widely used chemical commodity. It can be produced from various energy sources, such as by electrolysis of renewables or by reforming natural gas. Today, nearly all hydrogen is produced from natural gas.

For applications in the transport sector, production by reforming from natural gas is currently the most common method. If the resulting CO₂ would be captured, this could result in a zero-emission value chain for shipping.

When used in combination with marine fuel cells, the emissions associated with other marine fuels could be minimized or eliminated entirely. If H₂ is generated using renewable energy, nuclear power, or natural gas with carbon capture and storage, zero-emission ships are possible.

Power generation systems based on H₂ may eventually be an alternative to today's fossil-fuel-based systems. While fuel cells are considered the key technology for hydrogen, other applications are also under consideration, including gas turbines or internal combustion engines in stand-alone operation or in arrangements incorporating fuel cells. Hydrogen-fueled internal combustion engines for marine applications are said to be less efficient than diesel engines. Hydrogen-fueled piston engines for ships are not available in the market. On land development is ongoing. Possibly larger-scale industrial and maritime applications combined with waste heat recovery solutions might be better suited for high-temperature technologies such as solid oxide fuel cells (SOFC) or even industrial systems using molten carbonate fuel cells.

Fuel cells combined with batteries (and possibly super capacitors) adding peak-shaving effects are a promising option. Even proton-exchange membrane fuel cells (PEMFC), thanks to their flexible materials, could improve fuel cell lifetime significantly when protected against the harshest load gradients. SOFC must be applied in a hybrid environment using peak-shaving technology to be a realistic alternative for shipping.

Regarding the marine application, the European Commission FCSHIP study concluded in 2004 that the use of fuel cells in ships was feasible, with the usual caveats about availability and fuel supply. The subsequent METHAPU project sets out to

evaluate solid oxide fuel cell technology running on methanol for ship auxiliary power. For example, a 20 KW fuel cell system from Finnish company Wärtsilä was installed on the deck of the car carrier MV Undine. The 2009–2010 trial showed that the use of fuel cell technology and an alternative fuel poses no more of a risk to a commercial vessel than conventional equipment and fuel, laying the foundation for further deployment.

The FellowSHIP project has tested fuel cell technology integrated with a ship's propulsion system. In September 2009, a 330 KW molten carbonate fuel cell from MTU Onsite Energy was installed onboard the Viking Lady, an LNG-fueled ship with gas-electric propulsion system, without the need for pre-reforming. During the trial, the fuel cell logged 18,500 successful operating hours, providing supplementary power to the ship at an electrical efficiency of over 52% at full load. The next phase of FellowSHIP, now underway, is installing a battery pack for energy storage to create a true hybrid propulsion system for the Viking Lady.

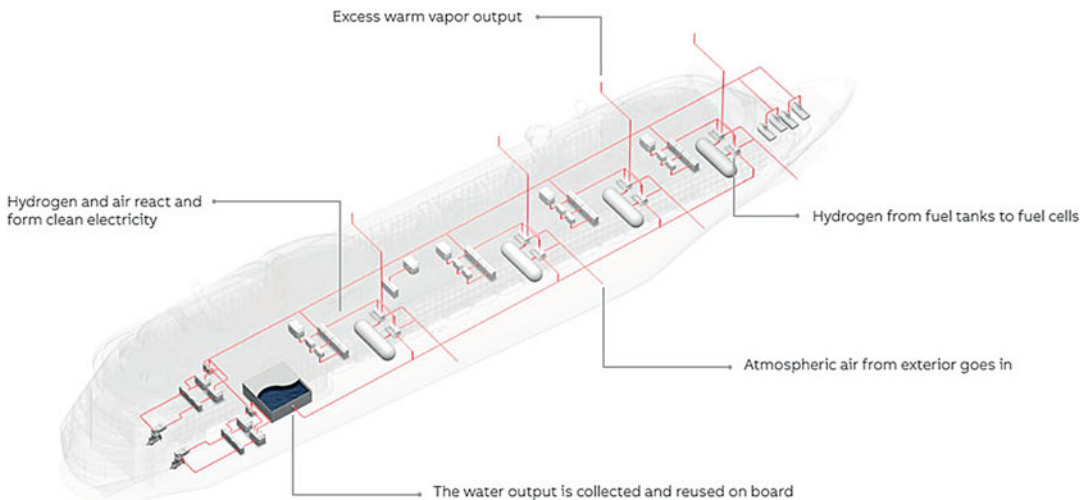
Backed by the Norwegian Maritime Authority (NMA) and led by shipbuilder Fiskerstrand (see Fig. 5), the HYBRIDShips innovation project aims to have a pilot version of a hybrid hydrogen/battery-powered ferry operational by 2020. HYBRIDShips will establish the knowledge base necessary to build zero-emission propulsion systems for larger vessels and longer voyages. The pilot ferry will be the world's first such hybrid to combine hydrogen fuel cell power working alongside batteries.

In 2018 June, ABB and Ballard Power Systems decided to jointly develop zero-emission fuel cell power plant for shipping industry as in Fig. 6. The two companies propose to develop an electrical generating capacity of 3 MW (4000 horsepower) module and fit within a single module no bigger in size than a traditional marine engine running on fossil fuels.

Typically, a fuel cell power installation is shown as in Fig. 7. The IMO has looked at fuel cells as part of its Carriage of Cargoes and Containers Sub-Committee, with a view to include safety provisions in the International Code of Safety for Ships using gases or other low-flashpoint fuels (IGF) code.

Alternative Fuel for Ship Propulsion,

Fig. 5 Project of hydrogen-powered ferry in Norway (Courtesy of Fiskerstrand)



Alternative Fuel for Ship Propulsion, Fig. 6 The installation of proton-exchange membrane fuel cells onboard ship (Courtesy of ABB)

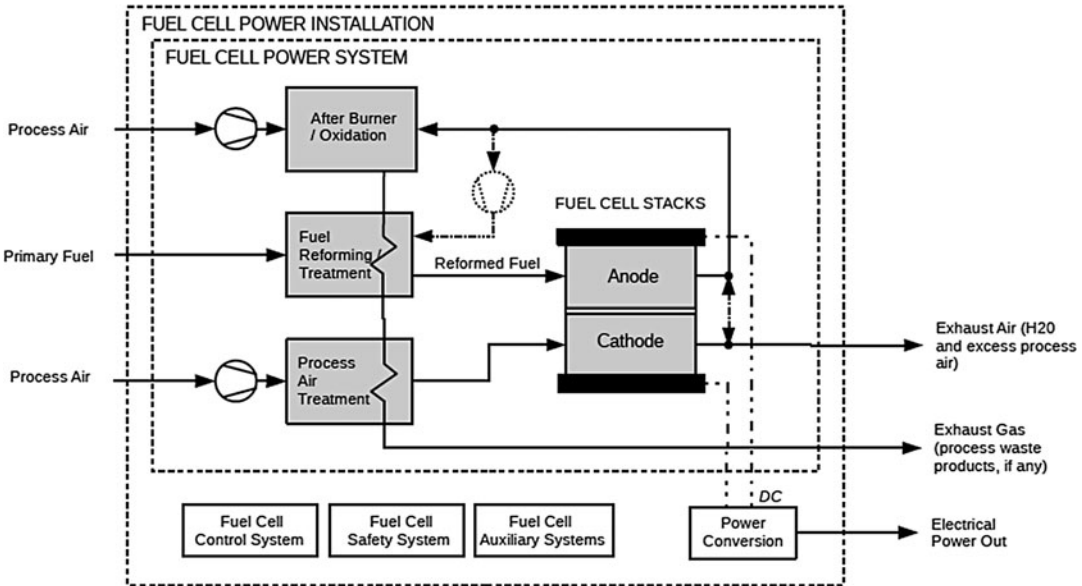
Liquefied Petroleum Gas (LPG)

Liquefied petroleum gas (LPG) is any mixture of propane and butane in liquid form. In the USA, the term LPG is generally associated with propane. Specific mixtures of butane and propane are used to achieve desired saturation, pressure, and temperature characteristics. Propane is gaseous under ambient conditions, with a boiling point of -42°C . It can be handled as a liquid by

applying moderate pressure (8.4 bar at 20°C). Butane can be found in two forms: n-butane or isobutane, which have a boiling point of -0.5°C and -12°C , respectively. Since both isomers have higher boiling points than propane, they can be liquefied at lower pressure. Regarding land-based storage, propane tanks are equipped with safety valves to keep the pressure below 25 bar. LPG fuel tanks are larger than oil tanks due to the lower density of LPG.

COMPONENTS OF A TYPICAL FUEL CELL POWER INSTALLATION

Notes: 1) The boundaries show the groups of components as defined and do not reflect a spatial arrangement!
2) Subject to FC technology, not all components are applicable and configuration may vary.



Definitions: FUEL CELL POWER SYSTEM is the group of components which may contain fuel or hazardous vapors.

FUEL CELL SPACE(s) are defined as any space containing components of the group FUEL CELL POWER SYSTEM (but may contain additional components of the group FUEL CELL POWER INSTALLATION)

Alternative Fuel for Ship Propulsion, Fig. 7 Components of a typical fuel cell power installation

There are two main sources of LPG: it occurs as a by-product of oil and gas production or as a by-product of oil refinery. It is also possible to produce LPG from renewable sources, for example, as a by-product of renewable diesel production.

Basically, there are three main options for using LPG as ship fuel:

- In a two-stroke diesel-cycle engine
- In a four-stroke, lean-burn Otto-cycle engine
- In a gas turbine

Currently, only a single two-stroke diesel engine model is commercially available, the MAN ME-LGI series. In 2017, a Wartsila four-stroke engine was commissioned for stationary power generation (34SG series). This engine had to be derated to maintain a safe knock margin. An alternative technology offered by Wartsila consists in the installation of a gas reformer to turn

LPG and steam into methane by mixing them with CO₂ and hydrogen. This mixture can then be used in a regular gas or dual-fuel engine without derating.

LPG can be stored under pressure or refrigerated. It will not always be available in the temperature and pressure range a ship can handle. Therefore, the bunkering vessel and the ship to be bunkered must carry the necessary equipment and installations for safe bunkering. A pressurized LPG fuel tank is the preferred solution due to its simplicity and because the vessel can bunker more easily using either pressurized tanks or semi-refrigerated tanks without major modifications.

The cost of installing LPG systems onboard a vessel (e.g., internal combustion engine, fuel tanks, process system) is roughly half that of an LNG system if pressurized type C tanks are used in both cases. This is because there is no need for special materials that can handle cryogenic temperatures. On large ships, the cost difference

between LNG and LPG systems is lower if the LPG is stored in pressurized type C tanks, which are more expensive than large prismatic tanks. Alternatively, LPG can be stored at low temperatures in low-pressure tanks, which require thermal insulation.

Cross-References

- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)

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American Petroleum Institute (API)

- [Christmas Tree](#)
- [Installation of Offshore Pipelines](#)

Analog-to-Digital (A/D)

- [Underwater Acoustic Communication](#)

Analysis of Renewable Energy Devices

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Synonyms

BEM – Blade element momentum; CFD – Computational fluid dynamics; DLL – Dynamic-link library; FEM – Finite element method; GDW – Generalized dynamic wake theory; HAWT – Horizontal-axis wind turbine; MBS – Multibody simulation; ME – Morison's equation; OWT – Offshore wind turbine

Definition

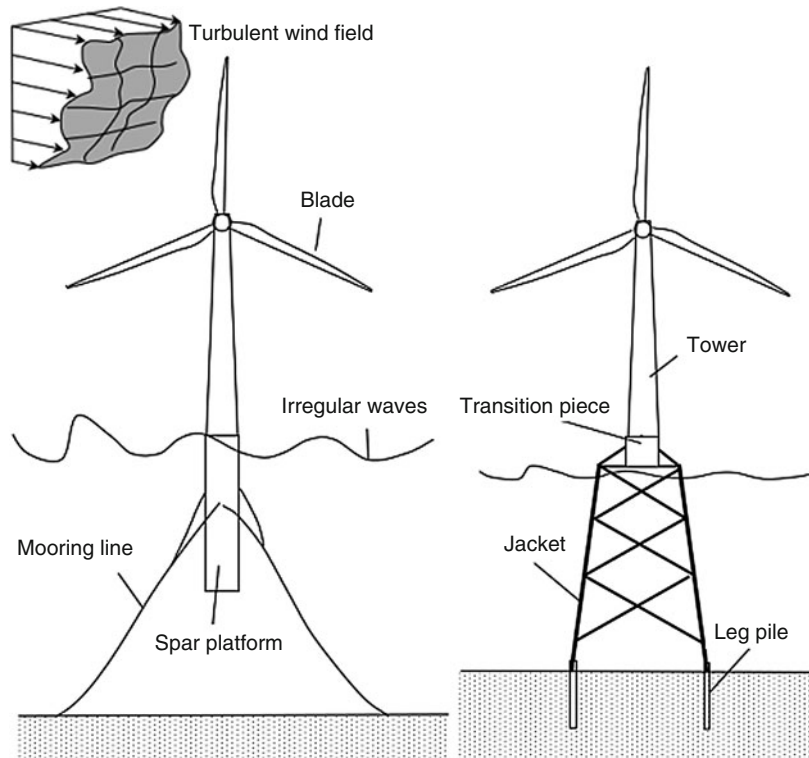
Analysis refers to the application of scientific and engineering principles and processes to reveal the properties of a system.

Introduction

Renewable energy devices may operate in complex internal and external conditions, and engineering analysis is a key element of the design process. Because of the sophisticated physics associated with specific renewable energy devices, simplified analysis is often not adequate for a full understanding of the system behavior. With the advent of information and digital technologies, analysis at various fidelity levels can be achieved using simulation tools. To reduce the computational costs and to validate the accuracy of simulation tools, significant efforts have been invested from both industry and academia.

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Fig. 1 Schematic of a spar floating wind turbine and a jacket-supported offshore wind turbine



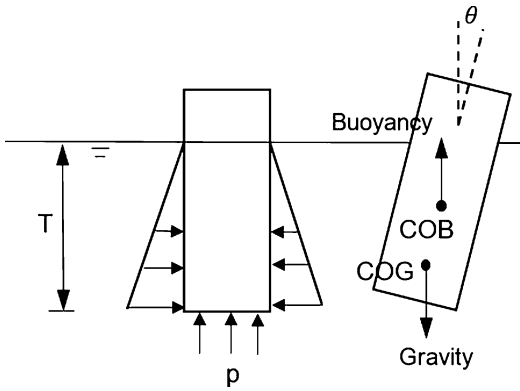
This chapter will focus on the analysis of renewable energy devices that operate in offshore environments. Such devices may experience hydrodynamic and aerodynamic load actions and have large dynamic motions and structural deformations. Figure 1 (left) presents an example of a floating wind turbine which is originally designed for the North Sea environments (Nielsen 2013). A bottom-fixed offshore wind turbine (OWT) is shown in Fig. 1 (right) which is intended for intermediate water depths below 50 m. At the design stage, dynamic response analysis must be performed to assess the power production quality, blade deflections in extreme wind conditions, platform pitch motions, mooring line dynamics, and so forth (Jiang et al. 2018).

Hydrostatic Analysis

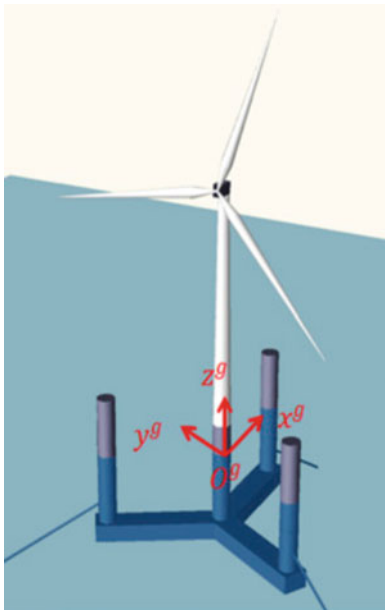
Traditionally, hydrostatic analysis deals with naval architectures including ships and offshore

drilling units. To ensure safety of such floating installations, classification authorities (DNV GL 2013; ABS 2018) dictate that all offshore units have positive metacentric height in calm water position for afloat conditions and intact stability and damage stability be checked. Thus, hydrostatic analysis is also involved when renewable energy devices are supported on floating foundations. As shown in Fig. 2, a spar buoy has a simple geometry with a draft of T has hydrostatic pressure (p) acting on the submerged hull. The pressure is a function of the submerged depth. After a static heel within certain range, the buoy should be able to return to its equilibrium position with the assistance of the restoring moment created by the buoyancy and gravity forces. Details of hydrostatics can be found in Biran and Pulido (2013).

Note that hydrostatic analysis can be among the first steps in dimensioning the floating platforms. Luan et al. (2016) checked the intact stability of a semisubmersible wind turbine (Fig. 3) considering the overturning moment from



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Fig. 2 Schematic of a buoy and hydrostatic pressure



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Fig. 3 Illustration of the 5-MW CSC wind turbine (Luan et al. 2016)

aerodynamic loads and the righting moment by hydrostatic pressure. As shown in Fig. 4, the area under the righting moment curve (ϕ) is relatively large compared to the area under the design overturning moment (straight line), and large safety margin is indicated. Similar analyses can also be found in the works of Lefebvre and Collu (2012) and Karimirad and Michailides (2015).

Hydrodynamic Analysis

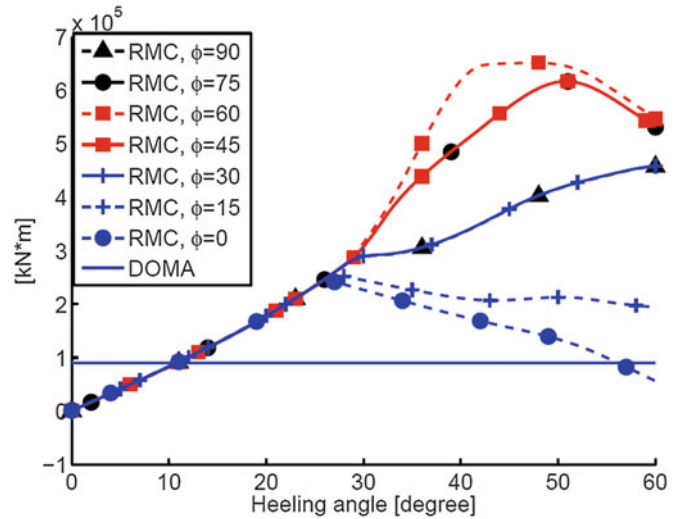
For renewable energy devices that are subjected to ocean waves and currents, hydrodynamic load can be an important source of excitation. Both viscous effects and potential flow effects (Faltinsen 1993) may play a role in determining the wave-induced loads and motions on certain types of renewable energy devices including wind turbines and wave energy converters. Figure 5 (upper) shows a vertical cylinder with diameter D that stands on the seabed, and an incident wave with wavelength of λ and wave height of H is propagating toward the cylinder. Based on Fig. 5 (lower), it is possible to judge which wave forces are of greater significance to the structure. If the cylinder is slender with large λ/D ratio, then viscous forces are prominent due to flow separation. For wind turbines supported by jacket or monopile foundations, the hydrodynamic forces acting on the foundations can often be represented by Morison's equation (ME); see Morison et al. (1950). Suppose that the slender structure can be divided into many strips, the hydrodynamic force per unit length normal to each strip can be expressed as

$$f_s = \rho C_M \frac{\pi D^2}{4} \ddot{x}_w - \rho (C_M - 1) \frac{\pi D^2}{4} \ddot{\eta}_1 + \frac{1}{2} \rho C_D D (\dot{x}_w - \dot{\eta}_1) |\dot{x}_w - \dot{\eta}_1| \quad (1)$$

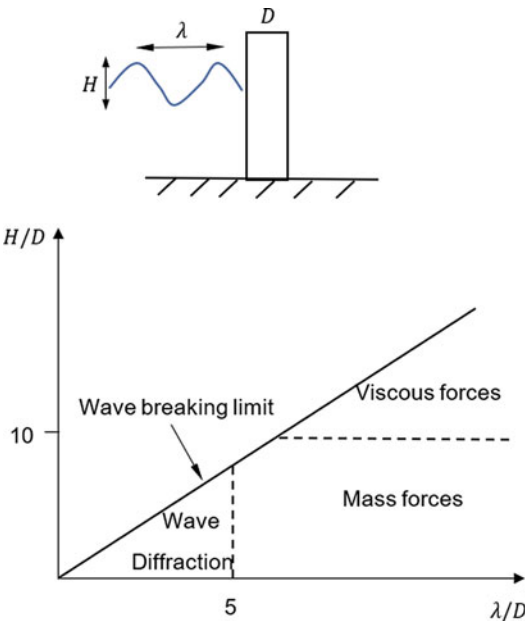
where ρ is the density of seawater, D is the diameter of the slender structure, C_M is the mass coefficient, and C_D is the drag coefficient, $\dot{x}_w$ and $\ddot{x}_w$ are the velocity and acceleration of water particles at the strip center, and $\dot{\eta}_1$ and $\ddot{\eta}_1$ are the velocity and acceleration of the strip. In Eq. (1), C_M and C_D are dependent on the Reynolds number, the Keulegan-Carpenter number, and surface roughness. Reference values are given by offshore standards (NORSOK 2007; DNV 2010). As the strip velocity and acceleration are included in Eq. (1), this equation is also applicable to moving structures. Note that although the ME has been widely applied by industry and academia, this equation ignores lift forces, slamming forces, and axial Froude-Krylov forces.

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Fig. 4 Intact stability analysis of the 5-MW-CSC, ϕ , heeling angle; RMC, righting moment curve; DOM, design overturning moment curve (Luan et al. 2016)



A



Analysis of Renewable Energy Devices,
Fig. 5 Importance of mass, viscous, and diffraction forces on marine structures (Faltinsen 1993)

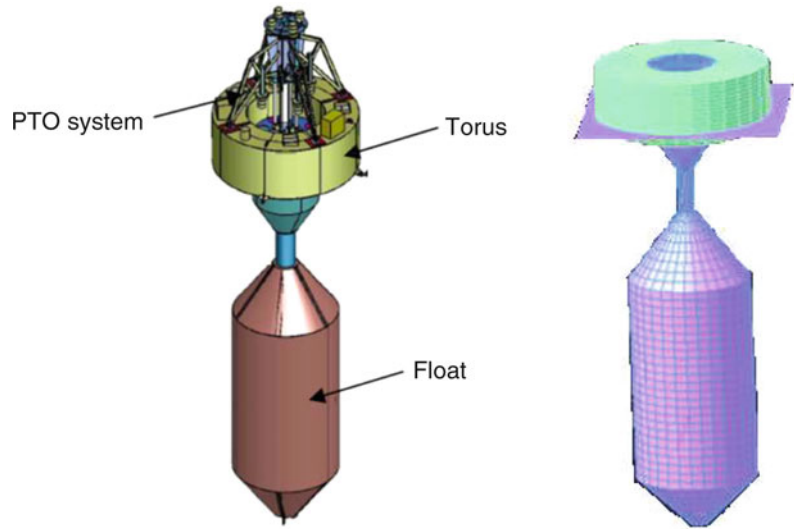
Hydrodynamic analysis of monopile-type wind turbines using the ME can be found in Veldkamp and Van Der Tempel (2005), Shirzadeh et al. (2013), and Jiang (2018). For monopile foundations supporting 10-megawatt (MW) wind turbines, the diameter can reach 10 m (Velarde 2016), and the ME is not necessarily applicable

for short waves. For jacket-type wind turbines, the tubular members have relatively small diameters. For example, the maximum leg diameters of traditional jackets supporting 5-MW wind turbines generally do not exceed 2 m (Chen et al. 2016; Dong et al. 2011). Thus, the hydrodynamic loads are drag-dominated for extreme waves, and the ME is well-suited for the analysis. Shi et al. (2013a, b) performed dynamic loads analysis of jacket-type wind turbines extensively and applied the ME during the hydrodynamic analysis. The ME has also been considered in the hydrodynamic analysis of floating platforms including spar buoy (Jonkman 2007; Jiang et al. 2013b), tension leg platform (Bachynski and Moan 2012), as well as mooring systems (Kvittem et al. 2012).

For floating renewable energy devices supported by large-volume structures, the diffraction effects are important, and potential flow theory is often used. Potential flow theory is realized numerically through the panel method (boundary element method), which solves the boundary-value problem for the interaction of wave-waves with prescribed motions in finite and infinite water depth (Lee 1995). Figure 6 schematizes a two-body wave energy converter which produces power using the relative heave motion between the torus and the float. A panel model is shown of the submerged parts. For a low-order panel method, the velocity potential is constant over the panel, and the diagonal length of the panel mesh is

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Fig. 6 Schematic of a two-body wave energy converter and the panel model (Muliawan et al. 2013a)



recommended to be below $1/6$ of the smallest wavelength analyzed (DNV 2010). Potential theory assumes ideal flow and ignores viscous effects. For floating structures with sharp corners, viscous effects may be more important, and Morison-type elements can be applied in time-domain simulations to complement the potential-flow solution. Kvittem et al. (2012) adopted this approach for a semisubmersible wind turbine with heave damping plates.

The panel method is a frequency-domain approach applicable to weakly nonlinear hydrodynamic problems. If there is highly nonlinear interaction between waves and floating bodies, analytical approaches (Faltinsen et al. 2004), the time-domain boundary element approaches (Schlør et al. 2016; Salehyar et al. 2017), or computational fluid dynamics methods (Li and Yu 2012) can be used. Chella et al. (2012) presented an overview of the wave impact forces on OWT substructures. Saletti (2018) studied the bottom slamming phenomenon for a combined wind and wave energy converter.

Additionally, a wave theory must be involved to calculate the wave kinematics which is closely correlated with the hydrodynamic loads. The linear wave theory, or the airy wave theory (Airy 1841; Craik 2004), is the simplest solution for the flow field and is only applicable for small-amplitude waves. Figure 7 shows the applicability

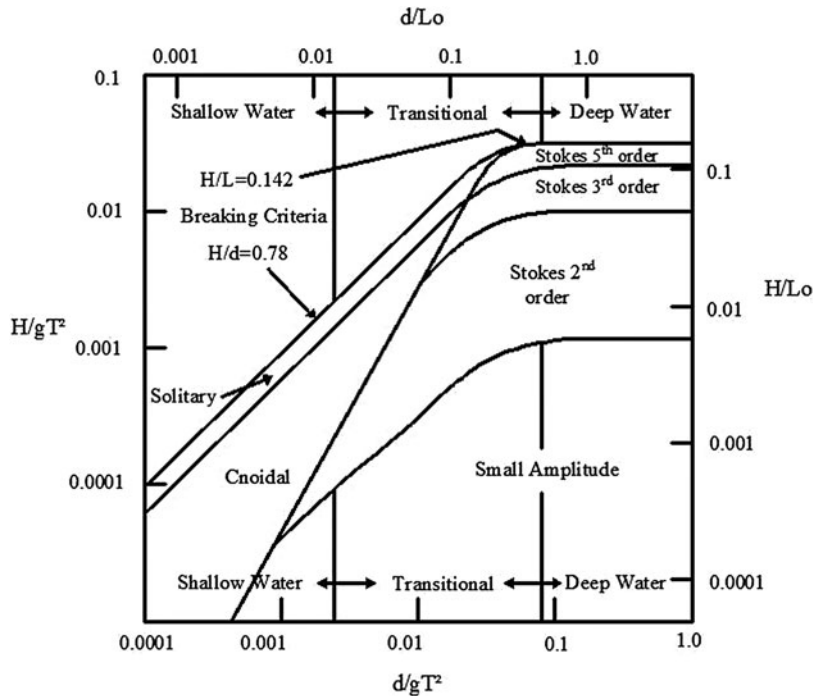
of various wave theories. Here, H denotes wave height, T is wave period, d is water depth, and L_0 is wavelength. Details on wave kinematics can be found in Dean and Dalrymple (1991).

Aerodynamic and Aeroelastic Analysis

For renewable energy devices that are exposed to wind loads, analysis of wind effect is indispensable. Wind turbines are particularly designed to harness the kinetic energy from the wind, and modern wind turbine blades are long and slender. For horizontal-axis wind turbines (HAWTs), the longest blade announced approaches 90 m for an 8-MW wind turbine (LM Wind Power 2018). Such flexible blades may experience large deformation under the combined effect of wind excitations, centrifugal forces, gravitational forces, and control actions. Aerodynamic analysis helps to understand the behavior of the airflow and the forces acting on the blades and the performance of the wind turbine. The classical blade element moment (BEM) was initially proposed by Glauert (1983) and modified for wind turbine analysis. The basic assumption of the BEM theory is that the force of a blade element is solely responsible for the change of axial momentum of the air which passes through the annulus swept by the elements, and there is no radial interaction between the

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Fig. 7 Applicability of wave theories (Sarpkaya 2010)



flows through contiguous annuli (Burton et al. 2011). BEM can be used to calculate the steady loads, the thrust, and the power of HAWTs. A simple BEM algorithm to find the axial and tangential induction factors is presented in Hansen (2008). A BEM algorithm with improved convergence rate is presented in Ning (2014). The classical BEM needs to be corrected by Prandtl's tip loss factor and Glauert correction to get reasonably good results, as compared to the measurements. Due to unsteadiness of the wind seen by the rotor, the classical BEM cannot realistically capture the aeroelastic behavior of wind turbines, and the unsteady BEM method should be considered. The unsteady BEM, albeit still efficient, considers the time behavior of loads and power by the dynamic wake model and the dynamic change of angle of attack by the dynamic stall model (Hansen 2008). Further, physical phenomena like wake meandering can also be incorporated as engineering corrections to BEM (Larsen et al. 2013).

The BEM theory considers uniform pressure distribution across a rotor plane. Unlike BEM, the generalized dynamic wake (GDW) method, also

known as the acceleration potential method, allows for a more general distribution of pressure across a rotor plane and includes inherent modeling of the dynamic wake effect, tip losses, and skewed wake aerodynamics (Moriarty and Hansen 2005). However, the GDW method was developed for lightly loaded rotors at high wind speeds, and the induced velocities are small relative to the mean inflow velocity. Detailed descriptions of the GDW theory can be found in Pitt and Peters (1980) and Suzuki (2000).

Over the past decades, computational fluid dynamics (CFD) has been widely used in aerodynamic analysis of rotors, and actuator disc and actuator line methods are special types of CFD methods (Hansen and Aagaard Madsen 2011). Krogstad and Eriksen (2013) presented a summary of different computational methods that were applied to predict the performance and wake development of a tested model wind turbine. De Vaal et al. (2014) used the actuator disc model to study the effect of surge motion of a floating wind turbine on rotor thrust and induced velocity. Wen et al. (2018) applied the free vortex method to study the power coefficient

overshoot of a floating wind turbine in surge oscillations.

Aeroelastic analysis refers to the type of analysis that deals with the interaction between the inertial, elastic, and aerodynamic forces when the structure is exposed to a fluid flow. For commercial wind turbines, stall-induced vibrations and classical flutter are two categories of instabilities that have been observed for stall- and pitch-regulated wind turbines, respectively. Hansen (2003, 2007) presented a state-of-the-art review on aeroelastic stability analysis of wind turbines.

Integrated Dynamic Analysis

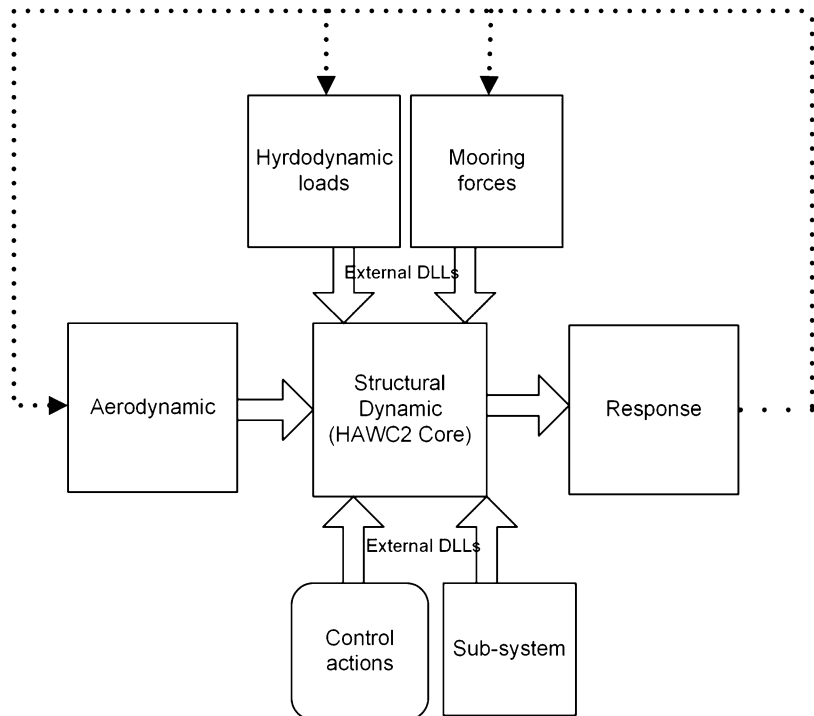
A complex renewable energy system may comprise structural components, mechanical parts, electrical devices, and control systems. Modeling and simulation of such systems can be achieved by using numerical methods such as multibody simulation (MBS) or finite element method (FEM). Integrated dynamic analysis of renewable energy devices under external load conditions is usually performed in the time domain and

provides global and structural responses. For OWTs, the integrated dynamic analysis is also addressed as aero-hydro-servo-elastic analysis (Jonkman et al. 2008) as aerodynamics, hydrodynamics, servo dynamics, and structural dynamics are involved in the analysis. Figure 8 illustrates the computational flowchart of an integrated dynamic analysis in the HAWC2 program (Larsen 2009). As shown, user-specified control actions including mechanical system faults can be added to the main program via external dynamic-link libraries (DLLs) written in a programming language. For HAWTs, blade pitch control and generator torque control are also achieved through DLLs. There exist many engineering challenges with regard to modeling complexity, coupling of different submodules, nonlinear responses, and time-domain efficiency; see Butterfield et al. (2007).

Global motions and structural responses are two main aspects of concern. The global motions refer to the horizontal displacement and rotation of structural members or bodies. For floating wind turbines, rigid body motions of the floating platforms under dynamic loading can provide insights

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Fig. 8 Modularized computational flowchart for floating wind turbine simulation (Jiang et al. 2013a)



into system dynamics. Nielsen et al. (2006) conducted dynamic response analysis of the floating wind turbine concept HYWIND under wind, wave, and current conditions and compared the simulated and experimental decay tests of tower pitch angle. Pereya et al. (2018) analyzed the nacelle acceleration and platform pitch motion of the TetraSpar floating wind turbine. Such global motion responses, albeit not stated explicitly, can form design constraints for drivetrain components. Kurniawan et al. (2012) performed modeling and global motion analysis of a pitching wave energy converter and discussed dynamics of such a system with different hydraulic components. Muliawan et al. (2013b) conducted dynamic response analysis of a combined wind-wave energy converter and uncovered positive synergy between the two floating bodies by global motion analysis. Shi et al. (2016) developed an ice load force module for an aero-hydro-servo-elastic program and identified important response characteristics of a monopile-type wind turbine under combined ice and wave loads. From an integrated dynamic analysis, structural responses are available. To ensure structural integrity of renewable energy devices, the structural responses need be checked against possible failure modes including fatigue and ultimate limit states. Dong et al. (2011) checked the long-term fatigue damage of tubular joints of a jacket-type OWT (see Fig. 1 right) after performing integrated dynamic

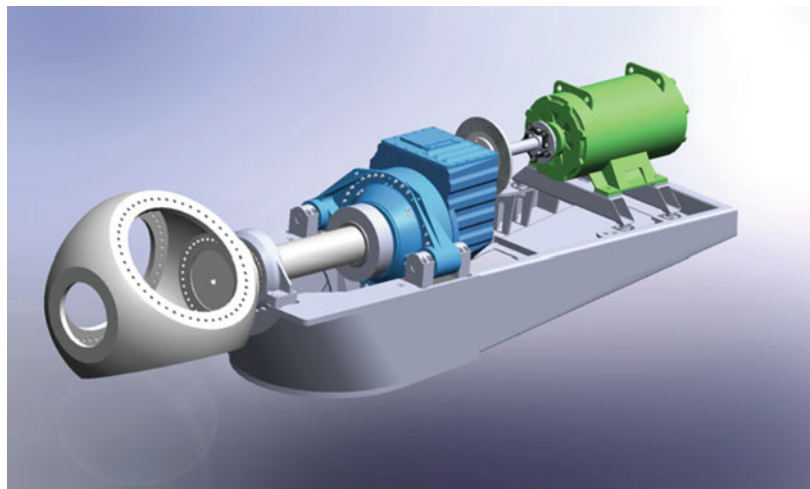
analysis. Jiang et al. (2015) analyzed the short-term fatigue damage of mooring lines of a floating wind turbine during shutdown. Wei et al. (2014) calculated the structural capacity of jacket support structure of an OWT under extreme wind and wave loading.

Analysis of Mechanical Components

For many renewable energy devices, mechanical components are widely used to transfer the energy to the generator side. Figure 9 demonstrates a three-stage gearbox of a 750-kilowatt stall-regulated wind turbine. The gearbox has one planetary stage and two parallel stages. High failure rates of gearbox components have been observed for the wind industry since its inception (McNiff et al. 1991), and analysis of mechanical components sometimes requires knowledge of both external loading conditions from the rotor side and internal conditions like friction, oil viscosity, and lubricant temperature (Harris and Kotzalas 2007). Numerical methods like FEM and MBS are useful tools for modeling the detailed setup of mechanical components and to study the internal responses under dynamic conditions. Using FEM and MBS, Xing and Moan (2013) showed that the main shaft non-torque loads can substantially contribute to the bearing loads and gear displacements. Dong et al. (2012) investigated gear

Analysis of Renewable Energy Devices,

Fig. 9 Illustration of a wind turbine gearbox (Image source: (Jiang et al. 2014a), courtesy of National Renewable Energy Laboratory)



contact fatigue in a wind turbine drivetrain using a decoupled analysis approach. Jiang et al. (2014a) proposed using multilevel integrated analysis for contact fatigue analysis of planetary bearings. Nejad et al. (2014, 2016) analyzed dynamic load effects of a drivetrain components and performed reliability analysis of wind turbine gears.

Hydraulic components including valves, pumps, accumulators, pipelines, and motors are also mechanical components that have been suggested for use in wave energy converters and wind turbines. Analysis of hydraulic systems often involves mathematical modeling and numerical simulations. Henderson (2006) presented both numerical simulation and laboratory tests of the hydraulic system employed in the Pelamis wave energy converter. Yang et al. (2010) investigated the wear damage in the piston ring and cylinder bore of a heaving-buoy wave energy converter. Numerical simulations of hydraulic transmission of wind turbines can be found in the works of Jiang et al. (2014b), Yang et al. (2015), Buhagiar et al. (2016), and Buhagiar and Sant (2017).

Code Verification and Validation

A multitude of design codes have been developed and extensively used for analysis of renewable energy devices. In general, many design codes adopt simplified physical representations of actual systems with reduced degrees of freedom but account for most prominent system features. Before being put into use, a new code should be verified against other state-of-the-art codes with adequate model fidelity levels or validated against experimental results. Larsen et al. (2013) showed good comparison between HAWC2 and the CFD code EllipSys3D for aerodynamic forces on a blade. Extensive benchmark work usually involves international collaboration among various academic and industrial partners. Passon et al. (2007) introduced the first international investigation and verification of aeroelastic codes for OWTs. Modeling capabilities of offshore environment, structural modeling, and rotor aerodynamics were compared among nine design codes for

four different support structures. Later, code-to-code verifications were conducted of other types of foundations with an increased number of participants and additional load cases; see Jonkman et al. (2008), Popko et al. (2012), and Vorpahl et al. (2014).

Experiments at model scale or full-scale testing are other effective means of code verification. Discrepancies in results between model testing and numerical codes are not uncommon, especially for renewable energy devices that have both aerodynamic and hydrodynamic excitations, because similarity between inertia and viscous forces of the models cannot be achieved simultaneously. Li and Calisal (2010) developed numerical codes using a discrete vortex method for vertical axis tidal current turbines and verified the two- and three-dimensional codes with experiments. Preliminary verification of a wave energy converter design tool with experimental wave tank results is presented in Ruehl et al. (2014). Coulling et al. (2013) verified a numerical model constructed in the design code FAST (Jonkman and Buhl 2005) with 1/50th-scale model test data for a semisubmersible floating wind turbine system. Luan et al. (2018) compared the simulated sectional responses of a semisubmersible using a nonlinear finite element code SIMO-Riflex with the 1/30th-scale model test results.

Cross-References

- [Design of Renewable Energy Devices](#)
- [Design Rules and Standards](#)

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Application of Image Processing in Ice–Structure Interaction

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Introduction

The understanding of Arctic physical processes and sustainable exploration, exploitation, and management of Arctic resources require more

detailed, precise, and continuous measurements of sea ice parameters. Because various types of ice are in the ice-covered regions and the sizes of the ice floes can range from about 1 meter to kilometers, the temporally and spatially continuous field observations of sea ice are necessary for marine activities. One of the best ways of observing the ice conditions in the oceans is by using aerial or nautical imagery. The use of cameras as sensors on mobile sensor platforms (e.g., unmanned vehicles) will aid the development of sea ice observation. It has the potential of continuous measurements with high precision, which is particularly important for providing detailed localized information of sea ice to ensure safe operations of structures in ice-covered regions (Haugen et al. 2011).

The collected ice images or videos must be analyzed by dedicated computer algorithms to extract useful ice information to dynamic ice estimators and for decision support in Arctic offshore engineering. Therefore, this article introduces image processing algorithms for providing the ice parameters that are important factors in the analysis of ice–structure interaction in an ice field. These useful ice parameters include ice concentration, ice types, ice floe size, and floe size distribution, and they are defined as follows.

Ice Concentration

Ice concentration (IC) is the ratio of ice on unit area of sea surface; it has been identified as one of the most influencing parameters on the magnitude of experienced forces during model tests (van der Werff et al. 2012; Comfort et al. 1999). To obtain ice concentration from a visual ice image, only the visible ice can be considered, including brash ice, and submerged ice if visible in the image. With the image area, the height of image taken above the ice sheet, and the segmentation which is the identification of the ice from water, the actual area of sea ice and sea surface can be derived. However, the actual domain area is not necessary for calculating ice concentration. In simplified terms, ice concentration from a digital visual image is defined as the area of sea surface covered by visible ice observable in the 2D visual image taken vertically from above, as a fraction of the

whole sea surface domain area. Hence, it is the ratio of the number of pixels of visible ice to the total number of pixels within the image domain. Note that the domain area is an effective area within the image, excluding land or other nonrelevant areas. The ice concentration is then given by (Zhang et al. 2012a)

$$\begin{aligned}
 IC &= f(\text{image area, height above ice sheet, segmentation}) \\
 &= \frac{\text{Area of all visible ice}}{\text{Actual domain area}} \\
 &= \frac{\text{No. of pixels of visible ice in image domain}}{\text{Total no. of pixels in image domain}}
 \end{aligned} \tag{1}$$

Ice Types

Sea ice is any form of ice found at sea which has originated from the freezing of sea water. Different types of sea ice have different physical properties. Since one generally assumes that brash ice has a dampening effect in models for calculating ice pressure and ice forces, it may be more convenient to estimate the distribution between three classes, that is, the ratio of ice floes, the ratio of brash ice, and the ratio of water. As defined (Løset et al. 2006):

- Floe is any relatively flat piece of sea ice 20 m or more across. It is subdivided according to horizontal extent. A giant floe is over 10 km across; a vast floe is 2–10 km across; a big floe is 500–2000 m across; a medium floe is 100–500 m across; and a small floe is 20–100 m across.
- Ice cake is any relatively flat piece of sea ice less than 20 m across.
- Brash ice is accumulations of floating ice made up of fragments not more than 2 m across and the wreckage of other forms of ice. It is common between colliding floes or in regions where pressure ridges have collapsed.
- Slush is snow which is saturated and mixed with water on land or ice surfaces or as a viscous floating mass in water after heavy snowfall.

For simplicity, the size of sea ice piece is the only criterion to distinguish ice floe and brash ice

in this research. That is, any relatively flat piece of sea ice 2 m or more across is considered as “ice floe,” while any relatively flat piece of sea ice less than 2 m across is considered as “brash ice (piece).” And the rest of ice pixels, e.g., single ice pixels or the ice pieces that are too small to be treated as brash ice, are considered as “slush.”

Ice Floe Size and Floe Size Distribution

The estimation of ice floe size and floe size distribution among the “ice floes” gives an important set of parameters from ice images. In image processing, the ice floe size can be determined by the number of pixels in the identified floe. If the focal length f and camera height are available, the actual size in SI unit of the ice floes and floe size distribution can also be calculated (Lu and Li 2010) by converting the image pixel size to its SI unit size.

Image Processing Methods

A digital image is a numeric representation of a two-dimensional picture, and it is composed of pixels which are the smallest individual elements in the image. A pixel holds quantized values that represent the color or gray level of the image at a particular point.

Ice Pixel Detection

Ice concentration, as defined, is a binary decision of each pixel to determine whether it belongs to the class “ice” or to the class “water.” From Eq. 1, it is clear that the detection of the ice pixels from water pixels is crucial to obtaining the ice concentration from an ice image.

Otsu Thresholding

The pixels in the same region have similar intensity. Based on that sea ice is whiter than open water, ice pixels have higher intensity values than those belonging to water in a uniform illumination ice image. Thus, the thresholding, which is based on the pixel’s gray-level to turn a grayscale image into a binary image (whose pixels have only two possible intensity values, e.g., “0” and “1”), is a natural way to segment ice regions from water regions.

Assuming that an object is brighter than the background, the object and background pixels have intensity levels grouped into two dominant modes. A threshold value T is chosen to separate an image into an “object region” and a “background region.” Individual pixels are marked as “object” pixels if their value is greater than the threshold value and as “background” pixels otherwise, that is:

$$g(x, y) = \begin{cases} 1 & \text{if } f(x, y) > T \\ 0 & \text{if } f(x, y) \leq T \end{cases} \quad (2)$$

where $g(x, y)$ and $f(x, y)$ are the pixel values located in the x^{th} column and y^{th} row of the binary and grayscale image, respectively. Then, the grayscale image is turned into a binary image. The key of thresholding method is how to select the threshold value.

The Otsu thresholding method (Otsu 1975) is one of the most common automatic threshold segmentation algorithms. It requires that the histogram (the distribution of gray value) is bimodal and the illumination is uniform. In this method, the histogram of the image is divided into two classes (i.e., the pixels are classified as either foreground or background), and the goal is to find the threshold value that minimizes the within-class variance, given by (Otsu 1975)

$$\sigma_w^2(T) = \omega_1(T)\sigma_1^2(T) + \omega_2(T)\sigma_2^2(T) \quad (3)$$

where ω_1 and ω_2 are the probabilities of the two classes separated by a threshold T and σ_1 and σ_2 are the variances of these two classes. The threshold with the maximum between-class variance also has the minimum within-class variance. The between-class variance is given by (Otsu 1975)

$$\begin{aligned} \sigma_b^2(T) &= \omega_1(T)(\mu_1(T) - \mu(T))^2 + \omega_2(T)(\mu_2(T) - \mu(T))^2 \\ &\cong \omega_1(T)\omega_2(T)(\mu_1(T) - \mu_2(T))^2 \end{aligned} \quad (4)$$

where μ_1 and μ_2 are the means of these two classes and $\mu(T) = \omega_1(T)\mu_1(T) + \omega_2(T)\mu_2(T)$. This expression can also be used to find the best threshold and to update the threshold value iteratively.

K-means Clustering

Clustering is a statistical data analysis method that divides a data set into many groups (Basak et al. 1988), and it has been widely used in image segmentation, especially classifying the objects into many groups. This method is based on the mathematical distance measure between individual observations and groups of observations to find hidden structures in unlabeled data and assign the unlabeled data into groups, so that the data in one group are more similar to each other than to those in other groups.

Among various clustering algorithms, *k*-means is one of the simplest but most popular clustering algorithms. The goal of *k*-means clustering is to minimize the within-cluster sum of distance to partition a set of data into *k* clusters (MacQueen 1967). The step-by-step algorithm for this method in image segmentation is described below:

Step 1:

For image processing, a set of gray levels is given:

$$f(x_1, y_1), f(x_2, y_2), \dots, f(x_n, y_n). \quad (5)$$

Step 2:

Partition this set into *k* clusters:

$$\begin{aligned} f_i(x_1, y_1), f_i(x_2, y_2), \dots, f_i(x_{n_i}, y_{n_i}) & \quad i \\ = 1, 2, \dots, k. \end{aligned} \quad (6)$$

Step 3:

Calculate the local means of each cluster:

$$c_i = \frac{1}{n_i} \sum_{m=1}^{n_i} f_i(x_m, y_m) \quad i = 1, 2, \dots, k. \quad (7)$$

Step 4:

Gray level $f(x_j, y_j)$ ($j = 1, 2, \dots, n$) belongs to set p ($p = 1, 2, \dots, k$) if it has the shortest distance to set p than any other sets:

$$\begin{aligned} |f(x_j, y_j) - c_p| & \leq |f(x_j, y_j) - c_i| \quad i \\ & = 1, 2, \dots, k. \end{aligned} \quad (8)$$

Iterate Steps 3 and 4 until the local means are unchanged.

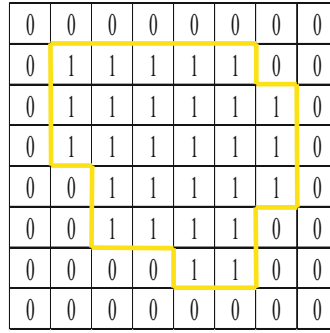
Ice Floe Boundary Detection

In image processing, the detection of ice floe boundaries can be used to distinguish individual ice floes. With the individual ice floe identification result, ice floe characteristics, such as location, area, perimeter, and shape measurements of each ice floe, together with the floe size distribution can thereby be estimated. Thus, ice floe boundary detection is a vital for extracting information of ice floes from ice images.

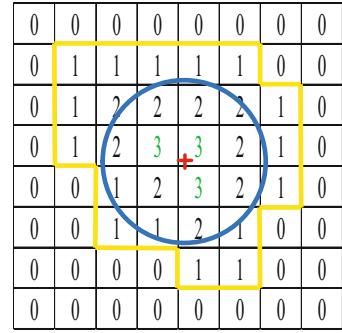
In an actual ice-covered environment, especially in marginal ice zone (MIZ), ice floes typically touch each other, and the edges between touching floes may be difficult to identify in digital images. This issue significantly affects the analysis of individual ice floe properties and floe size distribution. Among various floe boundary detection methods, for example, derivative boundary detection (Zhang et al. 2012a, b), morphology-based method (Zhang et al. 2012a, b; Banfield 1991; Banfield and Raftery 1992), watershed-based algorithms (Blunt et al. 2012; Zhang et al. 2013), etc., the GVF (gradient vector flow) snake-based approach (Zhang et al. 2015; Zhang and Skjetne 2015) is the most advanced method nowadays.

The GVF snake algorithm (Xu and Prince 1998) is an extension of the traditional snake (also known as active contours) algorithm (Kass et al. 1988) (the details of the traditional and GVF snake algorithms can be found in “Appendix A”). It has a good capability in the detection of weak boundaries. However, proper initial contours (an initial contour is a starting set of snake points for the evolution) are required by the GVF snake algorithm in ice floe boundary detection, especially when detecting the boundaries of massive ice floes. For each ice floe, it was showed that the initial contour close to the actual floe boundary, located inside the floe and centered as close to the ice floe center, is most effective (Zhang and Skjetne 2015). To accomplish the requirements of the initial contour without manual interaction, an automatic contour initialization algorithm based on the distance transform (the details of the distance transform can be found in “Appendix B”) and its regional maxima (a regional maximum is a connected component of pixels with a value

Application of Image Processing in Ice–Structure Interaction,
Fig. 1 Contour initialization algorithm



(a) Binary image matrix.



(b) Distance transform of Fig. 1(a), regional maximum, seed, and initial contour.

greater than any of its neighbors) is concluded as follows (Zhang and Skjetne 2014, 2015):

Step 1:

Convert the ice image into binary image after separating the ice from the water, in which case the pixels with value “1” indicate ice and pixels with value “0” indicate water; see Fig. 1a.

Step 2:

Perform the distance transform to the binary image, and find the regional maxima shown as the green numerals in Fig. 1b.

Step 3:

Merge the regional maxima into a big one if they have a short distance to each other, and then find the “seeds” that are centers of the regional maxima (including the merged ones), shown as red “+” in Figs. 1b and 2b.

Step 4:

Initialize the circular-shaped contours located at the seeds with the radii selected according to the pixel value at the seeds in the distance map; see the blue circles in Figs. 1b and 2b.

After initializing the contours, the GVF snake algorithm is run on each contour to detect the floe boundary. By superimposing all the detected boundaries over the binarized ice image, it allows to separate touching ice floes and thereby be able to identify individual floes, as shown in Fig. 2.

Floe Shape Enhancement

Some segmented floes may contain holes or smaller ice floes inside after boundary detection, and the shape of the segmented ice floe is rough. To smoothen the shape of the ice floe, morphological cleaning is used after ice floe boundary detection.

Morphological cleaning is a combination of first morphological closing and then morphological opening (Soh et al. 1998) on an image. Assume A is a binary image and B is a chosen structuring element (a structuring element is a shape that is used to probe an input image and draw conclusions on how the structuring element fits or misses the shapes in the input image. It can be represented as a matrix of 1 s, indicating the points that belong to the structuring element, and 0 s indicating otherwise); the morphological closing of A by B , denoted $A \bullet B$, is a dilation followed by an erosion (the details of dilation and erosion operations can be found in “Appendix C”):

$$A \bullet B = (A \oplus B) \ominus B \quad (9)$$

where $\oplus$ and $\ominus$ denote dilation and erosion, respectively. The morphological closing is the complement of the union of all translations of B that do not overlap A . It tends to smooth the contours of objects, generally joins narrow breaks, fills long thin gulfs, and fills holes smaller than the structuring element as seen in Fig. 3b.

(a) Sea ice floe image.

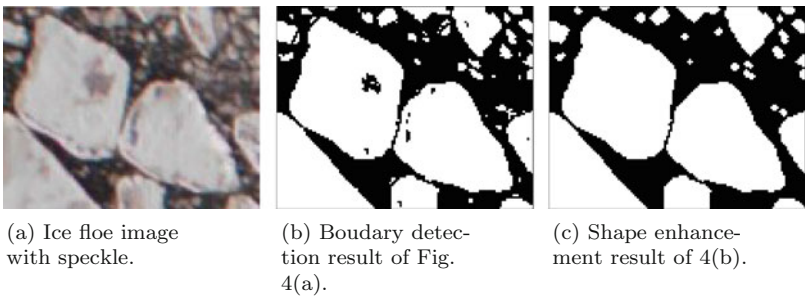
(b) Binary image with initial contours.

(c) separation result, individual floes are labeled in different colors.

[illegible][illegible][illegible]

To ensure that smaller ice floes contained in larger floes are removed, the morphological cleaning should be performed on smaller floes

Application of Image Processing in Ice–Structure Interaction, Fig. 4 Ice shape enhancement



Application of Image Processing in Ice–Structure Interaction, Table 1 Managed ice conditions in the test runs, target values (full scale)

Run no.	IC [%]	Floe size 1 (45%) [m]	Floe size 2 (40%) [m]	Floe size 3 (15%) [m]
5100	86	0.50	1.00	1.50
5200	70	0.50	1.00	1.50
5300	70	0.25	0.50	0.75
5400	86	0.25	0.50	0.75

first. Thus, the shape enhancement includes following two steps (Zhang and Skjetne 2014):

Step 1:

Arrange all the segmented ice floes from small to large.

Step 2:

Perform the morphological cleaning to the arranged ice floes in sequence.

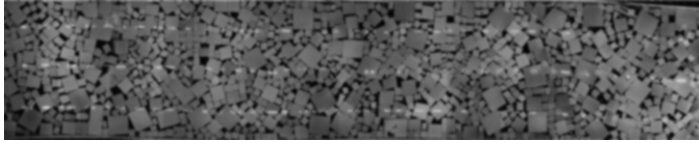
The shape enhancement result is shown in Fig. 4c.

Applications in Arctic Offshore Engineering

Model Ice Image Processing Applications

Before performing an analysis at full scale, the dynamic positioning (DP) experiments in model ice at the Hamburg Ship Model Basin (HSVA) in May 2011 allow for the testing of relevant image processing algorithms. This section shows the applications of the image processing techniques for determining important ice parameters from model ice data in the model-scale ice–structure analysis.

In these DP experiments, a managed ice condition was obtained by cutting the level ice layer into predefined ice floe shapes, and the behavior of two different model ships (an Arctic drillship and a polar research vessel) in a broken-ice field was studied. Four different types of ice fields were tested, varying in ice concentration and ice floe size distribution, as shown in Table 1. The model ice image data provided from the experiments includes two complete overview pictures from run nos. 5100 and 5200 and the videos of each of the four model test runs (see Table 1). The overall tank images were retrieved by stitching 28 top view pictures taken before execution of the model tests, showing a complete overview of the ice floe distribution in the ice tank, as seen in Fig. 5. The videos captured the local conditions around the fixed model vessel with a constant heading of 180° during each run by a top view video camera moving along with the carriage and model, as seen in Fig. 6. The four model ice videos are more than 24 min long with a frame rate of 25 fps. Since a video is composed from a sequence of frames, by gathering the result for each analyzed frame, the model ice parameters over time are retrieved. Before applying the algorithms to these videos, one frame per second is



Application of Image Processing in Ice-Structure Interaction, Fig. 5 Overall tank image for run no. 5100. Target ice concentration of 86%



Application of Image Processing in Ice-Structure Interaction, Fig. 6 One frame in the original video. Run no. 5400

found sufficient, and each frame was fed to the program for further processing (Zhang et al. 2012b).

Ice Concentration

Both Otsu thresholding and k -means clustering methods have been used to determine the ice concentration in model basin for run nos. 5100 and 5200. The analysis results are compared with the target ice concentration values, as presented in Table 2. Because the intensity values of all the model ice pixels are significantly higher than water pixels, both methods give similar result (Zhang et al. 2012b) and are both effective, as shown in Fig. 7.

When analyzing the ice concentrations in the vicinity of the model ship, the impediments around the tank in the videos are removed, and the vessel in the middle bottom of the tank is eliminated by a black rectangle, as seen in Fig. 8a. Then, the Otsu's and the k -means

Application of Image Processing in Ice-Structure Interaction, Table 2 Ice concentrations derived from different methods

Methods	Target value (%)	Otsu (%)	K-means (%)
Run no. 5100	86	83.17	82.86
Run no. 5200	70	62.50	62.00

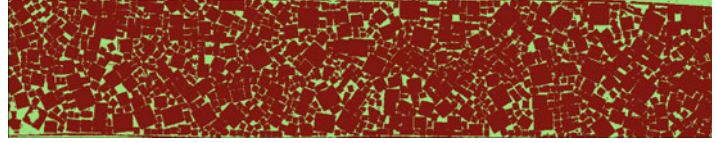
methods are applied in the video processing to calculate the ice concentration as a function of time. Figure 9 shows the variation of the calculated ice concentration in time for all test runs based on the Otsu's method, and the average ice concentrations after reaching the limiting values in all test runs are summarized in Table 3. Reduced ice concentration in the initial part of the test runs (before convergence) is related to the model ship positioning. It is an unwanted phenomenon, since it reduces the effective length of the ice tank. In all test runs, it was observed that the ice concentration in the near vicinity of the model was reaching a limiting value of approximately 80–89%, irrespective of the starting ice concentrations and floe sizes. This phenomenon can be explained by the tank's wall effect. That is, the ice floes were compacted by the model ship toward the end of the basin, such that the ice concentration asymptotically approached a limiting value.

Within the image analysis results, the time series of the ice concentration have been compared with the absolute total hull forces that were measured during the tests to investigate the relation between the ice concentration and the hull forces (see Fig. 10, for instance) (van der Werff et al. 2012). Further investigations are under consideration.

Application of Image Processing in Ice-Structure Interaction,
Fig. 7 Ice pixel detection for run no. 5100

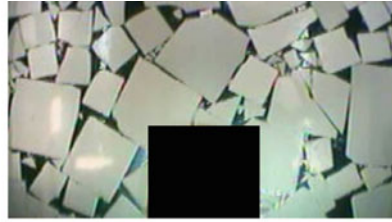


(a) Otsu thresholding method, $IC = 83.17\%$, Threshold = 84.

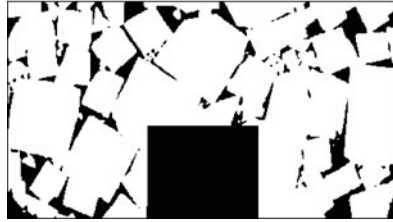


(b) K-means method, 2 clusters, $IC = 82.86\%$.

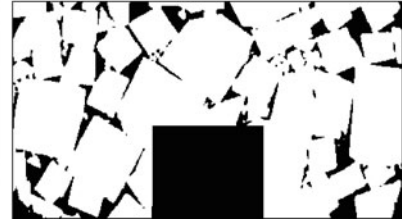
Application of Image Processing in Ice-Structure Interaction,
Fig. 8 Frames of run no. 5100 at 816 s and ice detection



(a) Pre-processed frame.



(b) Otsu, $IC = 87.26\%$, threshold = 100.



(c) K-means, $IC = 86.91\%$.

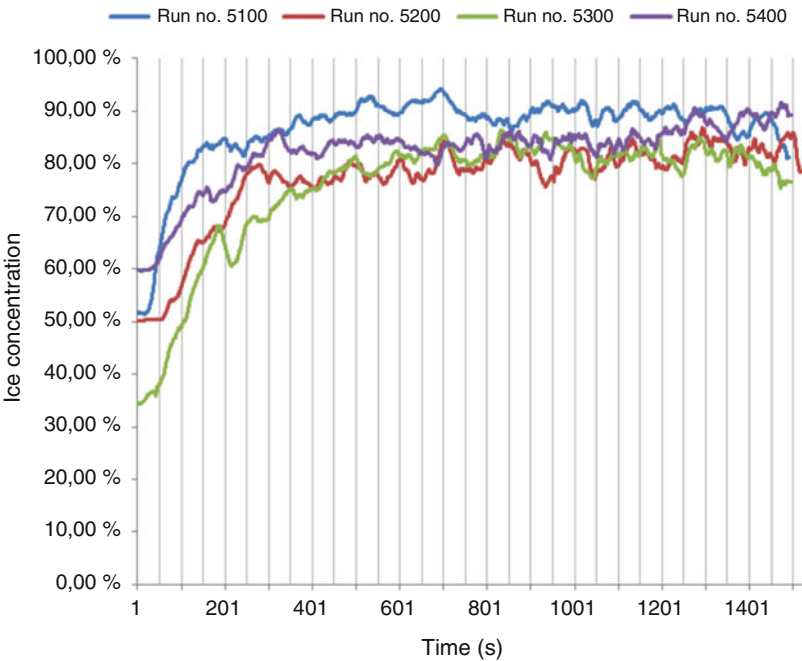
Model Ice Floe Monitoring

In one of the model ice tests as seen in Fig. 11a, the ice floes were modeled square shapes with predefined side lengths, and the largest floe has an area less than a predefined value. To evolve the GVF snake algorithm effectively for identifying ice floe boundaries, an automatic initial contour initialization method was introduced in section “Ice Floe Boundary Detection.”. However, it is difficult for the contour initialization method to initialize the contours for each square-shaped ice floe as shown in Fig. 11b, so that the GVF snake algorithm is unable to find all floe boundaries and many floes still touch each other as shown in Fig. 11c. Thus, an additional round of contour initialization and segmentation is required.

In this model ice test, although these model ice floes are not perfect squares, most of the floes could be approximated as rectangles with a length-to-width ratio less than the given threshold. Based on the characteristics of these rectangle-shaped model ice floes, the following three criteria can be used to determine whether it is necessary to initialize the contours and conduct a second segmentation (Zhang et al. 2015):

- The ice floe area is less than a given threshold.
- The ice floe has a convex shape (it could be done by determining if the ratio between the floe area and its minimum bounding polygon area is larger than the threshold).

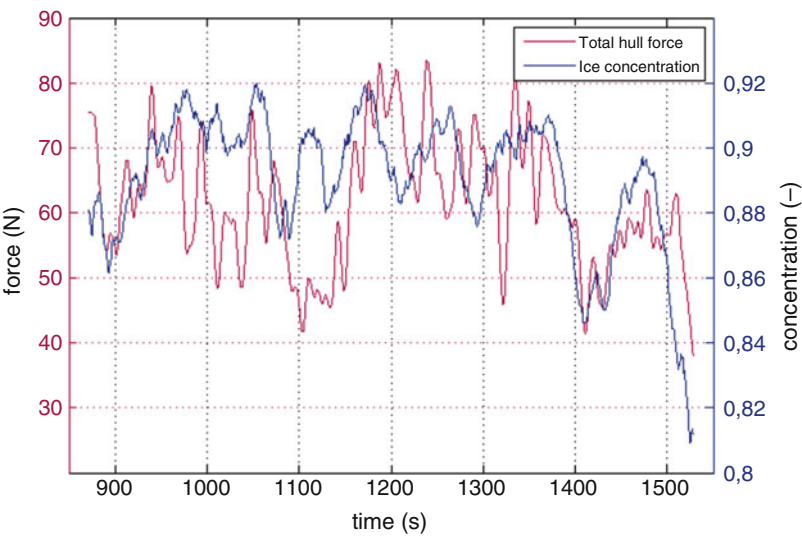
Application of Image Processing in Ice–Structure Interaction, Fig. 9 Time-varying *IC* of run nos. 5100–5400 based on Otsu thresholding



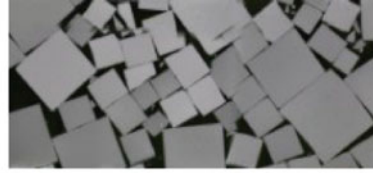
Application of Image Processing in Ice–Structure Interaction, Table 3 Average *IC* after reaching saturation in all test runs

Run no.	5100	5200	5300	5400
Start time (s)	200	300	600	300
Average <i>IC</i>	88.93%	80.39%	81.69%	84.83%

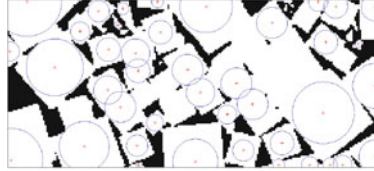
Application of Image Processing in Ice–Structure Interaction, Fig. 10 Hull force and ice concentration time series for run no. 5100 (high *IC*, large floes)



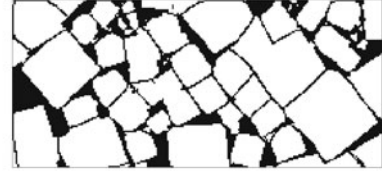
Application of Image Processing in Ice–Structure Interaction, Fig. 11 Crowded model ice floe segmentation



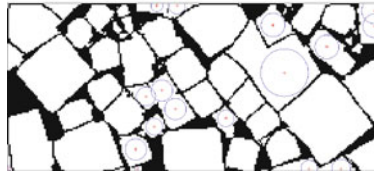
(a) Model ice image with crowded floes.



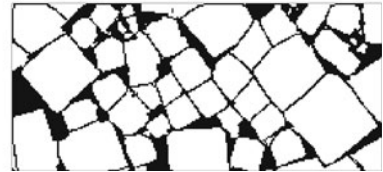
(b) Initializing the contours once.



(c) Results after first round.



(d) Initializing the contours twice.



(e) Final result.

- The length-to-width ratio of the minimum bounding rectangle of the ice floe is less than the threshold.

Note that these criteria are designed for segmenting the rectangular-shaped crowded model ice floes only. For crowded model ice floes with other shapes, the criteria can be replaced by the corresponding shape criteria.

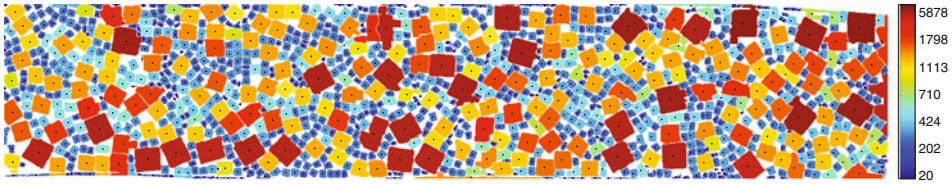
After a segmentation step, the algorithm will stop if all the segmented floes satisfy these criteria. Otherwise, the algorithm must find the floes that do not satisfy any of these criteria, find their seeds, initialize new contours, and perform the segmentation again (e.g., seen in Fig. 11d, e). After several segmentation steps, some segmented floes may still not satisfy the criteria. This is mainly because the boundaries of those floes are too weak to be detected. However, the total number of segmented floes will converge to a final solution. Therefore, the algorithm is made to stop if the total number of floes segmented after steps N and $N + 1$ are equal, in combination with an absolute stop criterion.

Finally, the ice shape enhancement algorithm is performed on the overall segmented model ice floe image to obtain the final model ice floe identification result. Figure 12 presents an example showing the crowded model ice floe identification result from Fig. 5 and the corresponding floe size distribution of an overall ice tank image (Zhang et al. 2015).

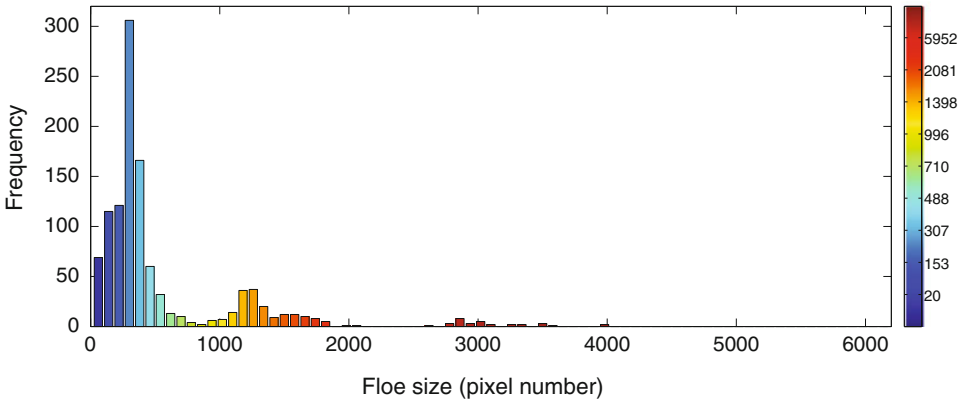
This model ice floe identification algorithm has been applied to an ice surveillance video to monitor the maximum floe size entering the protected vessel from a physical ice management operation, as seen in Fig. 13b. The maximum floe size for each frame is calculated as a function of time as seen in Fig. 14. Based on this result, a warning can be sent to the risk management system if the estimated risk based on the maximum floe size is too large.

Sea Ice Image Processing Applications

The Norwegian University of Science and Technology (NTNU) expedition Oden Arctic Technology Research Cruise 2015 (OATRC'15) was



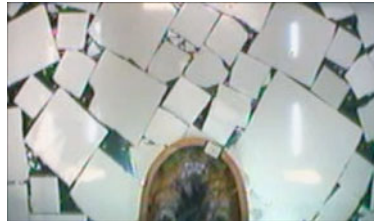
(a) Ice floe identification. The floe positions, found by averaging the positions of the pixels of each floe, are denoted using black dots.



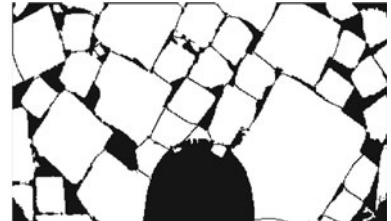
(b) Floe size distribution histogram.

Application of Image Processing in Ice-Structure Interaction, Fig. 12 Crowded model ice floe image identification and floe size distribution from Fig. 5

Application of Image Processing in Ice-Structure Interaction, Fig. 13 Model ice video processing



(a) Pre-processed frame at 965s.

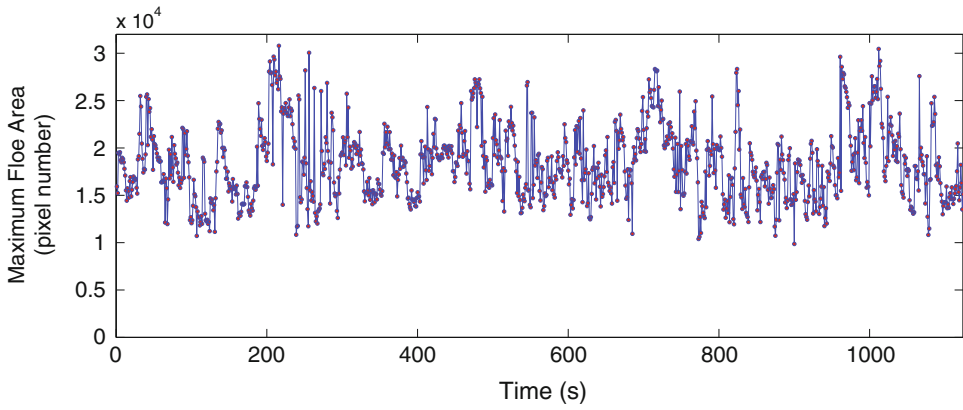


(b) Segmentation result of (a).

carried out in the Arctic region in September 2015. During the research cruise, one of the helicopter flight missions was to capture ice conditions in the marginal ice zone (MIZ). The images contain valuable ice information in the MIZ. In this section, several methods are sequentially presented with an example to demonstrate an automated procedure for ice floe and brash ice identification, their numerical representation, and forthcoming ice field generation.

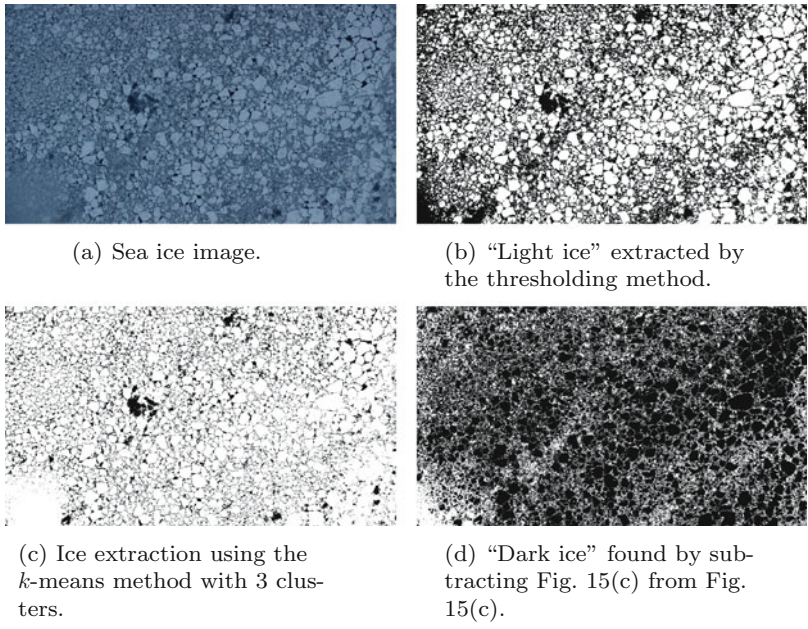
Sea Ice Image Processing

Most of the ice, we called “light ice,” can be identified by Otsu thresholding method. However, the “dark ice,” whose pixel intensity values are close to water pixels, may be lost. To determine more ice pixels, the k -means clustering method is used to divide the image into three or more clusters, considering the cluster with the lowest average intensity value to be water and the other clusters to be ice. The “dark ice” is then obtained



Application of Image Processing in Ice–Structure Interaction, Fig. 14 Maximum floe size entering the protected vessel

Application of Image Processing in Ice–Structure Interaction, Fig. 15 Sea ice image processing result



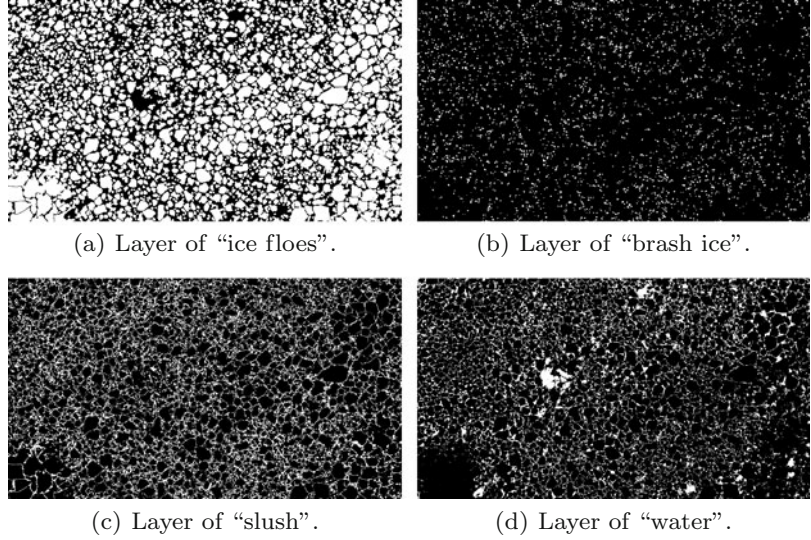
by comparing the difference between Otsu threshold detection result and *k*-means clustering detection result, as seen in Fig. 15d.

Thereafter, the GVF snake-based floe boundary detection method is run on the image layers of “light ice” and “dark ice” to individually derive “light ice” segmentation and “dark ice” segmentation. Collecting all the segmented ice pieces from both “light ice” and “dark ice” layers, the shape enhancement is then performed to complete the identification. According to section “Ice

Types,” the identified ice pieces are simply distinguished into ice floes and brash ice by defining a brash ice threshold parameter (e.g., number of pixels or area). Then the remaining of detected ice pixels, most of which are the detected boundary pixels between the touching floes, are considered to be “slush.” The sea ice image processing result is then four layers of ice floes (58.00%), brash ice (4.85%), slush (21.21%), and water (15.94%). The processed result of Fig. 15a can be found in Fig. 16, where a total of 2888 ice floes

Application of Image Processing in Ice–Structure Interaction,

Fig. 16 Sea ice image processing result of Fig. 15a

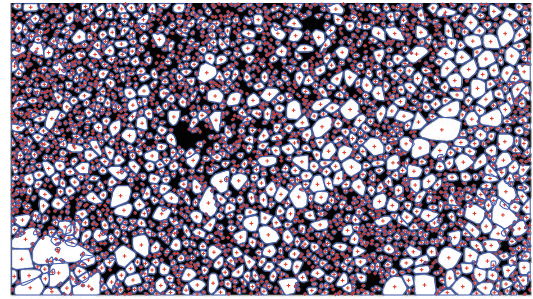


and 3452 brash ice pieces are identified from Fig. 15a.

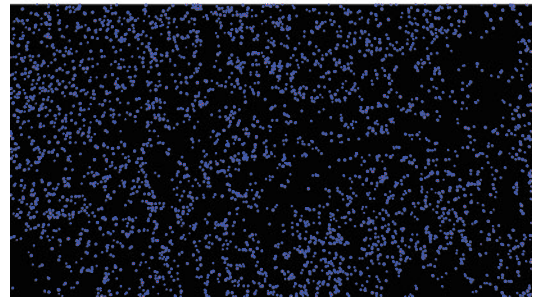
Sea Ice Numerical Modeling

Based on the image processing results, the identified ice floes and brash ice pieces are further simplified for the numerical simulation of the ice–structure interaction. In this modification, each sea ice floe is represented by a bounding polygon, and the brash ice pieces were reshaped by circular disks of equivalent area (Zhang 2015).

Figure 17 shows an example of sea ice modeling for Fig. 15a. A close-up view of a few ice floes and brash ice in the middle of Fig. 15a is given in Fig. 18, with the blue boundaries of the polygons/circles superimposed on top of the original segmented ice pixels. It is obvious to see that the polygonized floes will not be smaller than the actual identified floes and may overlap with other floes and brash ice pieces. Identifying the overlaps of floe-floe, floe-brash, and brash-brash is important when using the identified ice floes and brash ice as a starting condition for the initialization of an ice field in a numerical simulation for ice–structure interactions (Lubbad et al. 2015). To indicate these overlaps, an “overlap flag” is added to each polygonized floe to record the serial number of the floes and brash ice pieces with which the current floe overlaps and the “overlap flags” of



(a) Polygon fitting to ice floes in Fig. 16(a).



(b) Circle fitting to brash ice in Fig. 16(b).

Application of Image Processing in Ice–Structure Interaction, Fig. 17

Sea ice modeling for Fig. 15a

brash-brash and brash-floe are also registered in this modeling (Zhang and Skjetne 2018).

Ice Field Generation

The numerical representation of sea ice is utilized to generate its corresponding ice field to bridge the

gap between a natural ice field and its numerical applications, e.g., simulations involving ice–structure interactions. A major challenge of utilizing the digitalized ice field is overlaps among ice floes and brash ice. These overlaps are the consequence of both the input image’s visual noise (e.g., the foggy bottom-left corner) and inaccuracies introduced by the adopted image processing technique (e.g., the procedure to polygonize the segmented ice floes). A non-smooth discrete

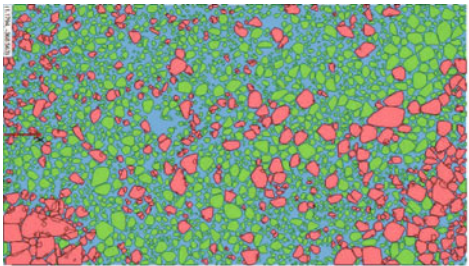
element method (DEM) is adopted to resolve all the overlaps among different bodies and assign basic physics to each ice floe and brash ice (Zhang and Skjetne 2018; Zhang 2020).

Ice floes are treated as discrete bodies after importing the ice field’s numerical representation into the non-smooth DEM-based simulator, and floe pairs involving overlap are labeled with red color in Fig. 19a. Afterward, for each calculation iteration, the collision detection algorithm identifies existing contacts; and the collision responses are calculated and applied to eliminate the overlaps. Figure 19b shows one snapshot of the ice field domain, within which overlaps are gradually resolved. Notably, for saving computation resources, not all the ice floes are involved in the calculation of each iteration. For ice floes without overlap and that are far away from the overlapped ice floe clusters, they are in “sleeping mode” in the adopted algorithm (see Fig. 19b).

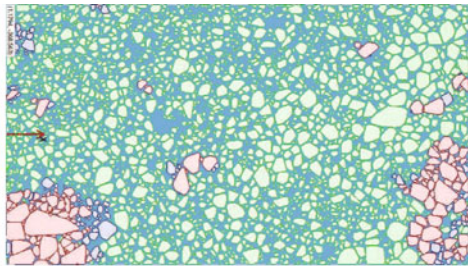
Figure 19a shows that ice floes in the ice field’s bottom-left corner have more overlaps. This is mainly because of the input image’s visual noises.



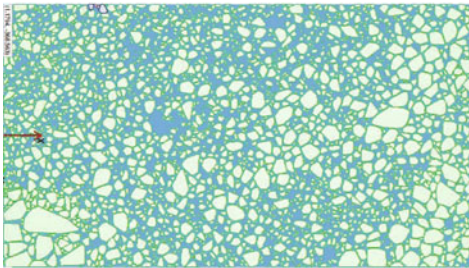
Application of Image Processing in Ice–Structure Interaction, Fig. 18 A close-up view of floe ice, brash ice, and their corresponding simplifications in modeling



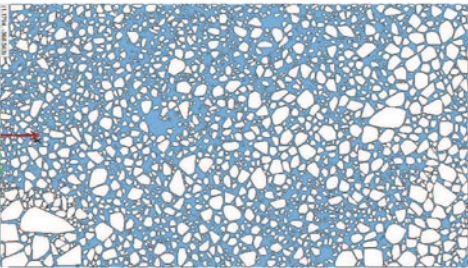
(a) Initial phase of the ice floe field with overlap.



(b) Calculation phase of the ice floe field with overlap.



(c) All overlaps are resolved.

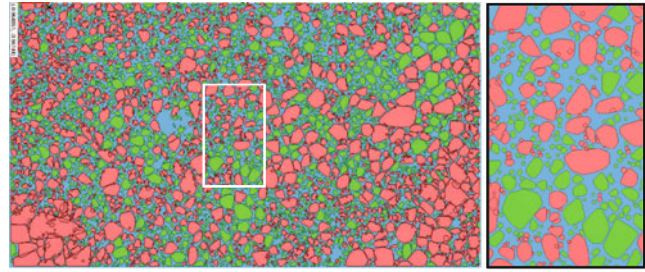


(d) Finally generated ice floe field.

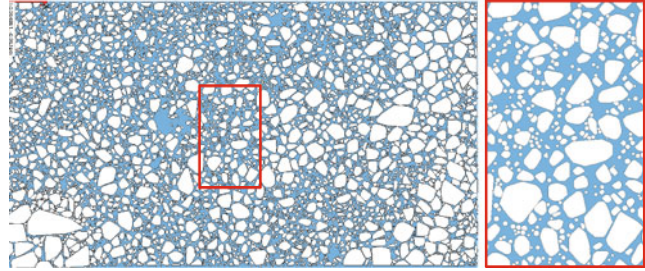
Application of Image Processing in Ice–Structure Interaction, Fig. 19 Ice floe field generation

Application of Image Processing in Ice–Structure Interaction,

Fig. 20 Ice field generation with both floe ice and brash ice



(a) Initial ice condition with overlaps.



(b) Final ice condition without overlaps.

Nevertheless, applying the above non-smooth DEM calculation procedures, all the overlaps are eventually resolved in Fig. 19c, and its corresponding final ice field is shown in Fig. 19d. After resolving the overlaps, the exact location of each ice floe in Figs. 19d and 17a is not the same, but with only minor differences. On the other hand, each ice floe's shape and size and the overall ice field's ice mass are conserved.

Similarly, brash ice can be imported into the same non-smooth DEM-based simulator and be treated as discrete bodies. From a non-smooth DEM calculation's point of view, the simplification of each brash ice as a disk with equivalent area makes the collision detection and consequent collision response calculation much easier comparing to arbitrary polygons. Given the amount of brash ice and its relatively small mass, this simplification is reasonable and has been adopted in previous studies (Konno 2009; Konno et al. 2011, 2013).

For the current demonstration, the identified brash ice in Fig. 16b and its numerical representation in Fig. 17b are additionally imported to the ice field in Fig. 19a. This is illustrated in Fig. 20a, which shows relatively much more overlaps. An enlarged view within the field center is also

presented. The circular disk-shaped bodies are the brash ice representations.

It is computationally efficient to make the circular disk-shaped simplification for brash ices. For the current ice field composition, i.e., 58.00% ice floe and 4.85% brash ice, the calculation time to resolve all overlaps for the cases with and without brash ice poses no significant difference. In both cases, the bottleneck for calculation time is on the overlap resolution in the bottom-left corner's large ice floes. However, it is expected that as the amount of brash ice increases, the calculation time would also increase, which eventually becomes the decisive bottleneck for the calculation. To certain point, it might be more efficient to model brash ice as a continuum, e.g., a viscous flow, which is governed by conservation laws as a material collection.

Conclusion

Various image processing techniques have been introduced in this entry to extract useful ice information from the collected ice image data to support the estimation of ice forces that are critical to marine operations in the Arctic. The introduced

methods have been applied to both model and sea ice image data to give some results applicable for ice engineering. More results and better information of ice from visual images will be investigated by further development of these image processing techniques.

Appendix A Traditional and GVF Snake Algorithms

Traditional Snake Algorithm

A traditional snake is a curve $\mathbf{C}(s) = (x(s), y(s))$ with the normalized arc length $s \in [0, 1]$ that moves through the spatial domain of an image to minimize the sum of the internal and external energy, given by

$$\mathbf{E} = \int_0^1 (\mathbf{E}_{\text{int}}(\mathbf{C}(s)) + \mathbf{E}_{\text{ext}}(\mathbf{C}(s))) ds \quad (11)$$

where $\mathbf{E}_{\text{int}}$ is the internal energy

$$\mathbf{E}_{\text{int}} = \frac{1}{2} \left(\alpha |\mathbf{C}'(s)|^2 + \beta |\mathbf{C}''(s)|^2 \right) \quad (12)$$

where α and β are weight parameters that control the snake's tension and rigidity, respectively. $\mathbf{C}'(s)$ denotes the first derivatives of $\mathbf{C}(s)$ with respect to s , making the snake act as a membrane, and $\mathbf{C}''(s)$ denotes the second derivatives, making the snake act as a thin plate.

$\mathbf{E}_{\text{ext}}$ is the external energy defined in the image domain. It attracts snakes to salient features in the image, such as boundaries. To find boundaries in a grayscale image, $\mathbf{I}(x, y)$, the image gradient is typically chosen as the external energy (Kass et al. 1988):

$$\mathbf{E}_{\text{ext}} = -|\nabla \mathbf{I}(x, y)|^2 \quad (13)$$

where $\nabla \mathbf{I}(x, y) = \left(\frac{\partial \mathbf{I}}{\partial x}, \frac{\partial \mathbf{I}}{\partial y} \right)$ is the image gradient that represents a directional change in the brightness of the image with the gradient angle $\theta = \arctan \left(\frac{\partial \mathbf{I}}{\partial y} / \frac{\partial \mathbf{I}}{\partial x} \right)$. When also considering the

image noise, the external energy is defined as (Kass et al. 1988)

$$\mathbf{E}_{\text{ext}} = -|\nabla \mathbf{G}_\sigma(x, y) * \mathbf{I}(x, y)|^2 \quad (14)$$

where $\nabla \mathbf{G}_\sigma(x, y)$ is a two-dimensional Gaussian function with a standard deviation σ and “*” denotes convolution.

To minimize the energy $\mathbf{E}$, a snake must satisfy the Euler equation:

$$\alpha \mathbf{C}''(s) - \beta \mathbf{C}''''(s) - \nabla \mathbf{E}_{\text{ext}} = 0. \quad (15)$$

Let $\mathbf{F}_{\text{int}} = \alpha \mathbf{C}''(s) - \beta \mathbf{C}''''(s)$ denote the internal force and $\mathbf{F}_{\text{ext}} = -\nabla \mathbf{E}_{\text{ext}}$ denote the external force. Then Eq. 15 can be written as the force balance:

$$\mathbf{F}_{\text{int}} + \mathbf{F}_{\text{ext}} = 0. \quad (16)$$

The internal and external forces are defined such that the snake will conform to an object boundary (or other desired features) within an image. The internal force $\mathbf{F}_{\text{int}}$ discourages stretching and bending, while the external potential force $\mathbf{F}_{\text{ext}}$ pulls the snake toward the desired image boundaries. Eq. 16 implies that the initial curve given in the snake algorithm will move under the influence of internal forces from the curve itself and external forces computed from the image data until the internal and external forces reach equilibrium.

To find a solution for Eq. 15, $\mathbf{C}(s)$ is treated as a discrete system of normalized arc length s and time t :

$$\frac{\partial \mathbf{C}(s, t)}{\partial t} = \alpha \mathbf{C}''(s, t) - \beta \mathbf{C}''''(s, t) - \nabla \mathbf{E}_{\text{ext}}. \quad (17)$$

When the solution $\mathbf{C}(s, t)$ becomes stationary, $\frac{\partial \mathbf{C}(s, t)}{\partial t}$ tends to zero, the energy $\mathbf{E}$ reaches a minimum, and the curve converges toward the target boundary.

The traditional snake algorithm is able to detect weak boundaries. However, there are two key limitations: the capture range of the external force fields is limited, and it is difficult for the

snake to progress into boundary concavities. The traditional snake algorithm is, therefore, sensitive to the initial contour, and the initial contour should be somewhat close to the true boundary.

GVF Snake Algorithm

To overcome the limitations of the traditional snake algorithm, the GVF snake algorithm introduces a spatial diffusion of the gradient of an edge map (which is derived from the image data) to expand the capture range of external force fields from boundary regions to homogeneous regions (Xu and Prince 1998).

The GVF is defined to be the vector field $\mathbf{v}(x, y) = (u(x, y), v(x, y))$ that minimizes the energy functional:

$$\epsilon = \iint \left[\mu (u_x^2 + u_y^2 + v_x^2 + v_y^2) + |\nabla f|^2 |\mathbf{v} - \nabla f|^2 \right] dx dy \quad (18)$$

where u_x, u_y, v_x and v_y are the derivatives of the vector field, μ is a parameter that controls the balance between the first and second term in the integrand, and f is an edge map (which could be the image gradient $|\nabla \mathbf{I}(x, y)|^2$) that is larger near the edges of objects in the image.

In Eq. 18, $|\nabla f|$ becomes large close to the object boundaries, in which case the second term dominates the integrand and is minimized by $\mathbf{v} = \nabla f$. Otherwise, $|\nabla f|$ is small, and the first term dominates the integrand to ensure that the external force field varies slowly and still acts in the homogeneous regions.

The GVF field can be found by solving the Euler equations:

$$\mu \nabla^2 u - (u - f_x) (f_x^2 + f_y^2) = 0 \quad (19a)$$

$$\mu \nabla^2 v - (v - f_y) (f_x^2 + f_y^2) = 0. \quad (19b)$$

A solution to Eq. 19a and Eq. 19b can be obtained by introducing a time variable, t , and finding the steady-state solution of the following partial differential equations:

$$u_t(x, y, t) = \mu \nabla^2 u(x, y, t) - (u(x, y, t) - f_x(x, y)) (f_x(x, y)^2 + f_y(x, y)^2) \quad (20a)$$

$$v_t(x, y, t) = \mu \nabla^2 v(x, y, t) - (v(x, y, t) - f_y(x, y)) (f_x(x, y)^2 + f_y(x, y)^2) \quad (20b)$$

Compared to the external force field in the traditional snake model having only f_x and f_y , the new vector fields, u and v in the GVF, are derived using an iterative method to find a solution for f_x and f_y . The result is that the capture range is effectively enlarged, as seen Fig. 21. Therefore, the GVF snake releases the requirements of the initial contour, so that the initial contour no longer needs to be as close to the true boundary.

Appendix B Distance Transform

Given a binary image $f(x, y)$, whose elements only have values of “0” and “1,” the pixels with a value of “0” indicate the background, while the pixels with a value of “1” indicate the object. Let $B = \{(x, y) | f(x, y) = 0\}$ be the set of background pixels and $O = \{(x, y) | f(x, y) = 1\}$ be the set of object pixels. The distance transform of a binary

Application of Image Processing in Ice-Structure Interaction,
Fig. 21 External forces

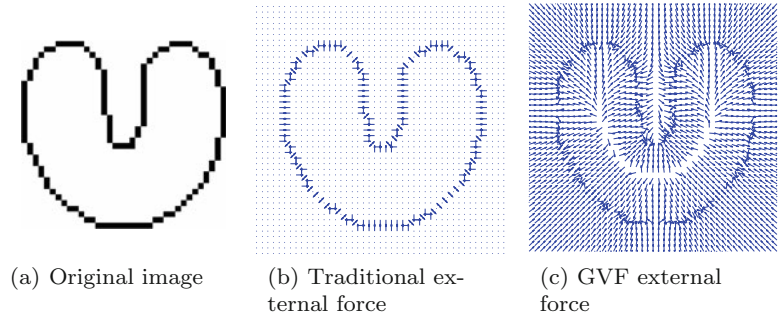


image f , $\mathbf{D}(x, y)$ is the minimum distance from each pixel in f to the background B , that is:

$$\mathbf{D}(x, y) = \begin{cases} 0 & \text{if } (x, y) \in B \\ \min_{b \in B} d[(x, y), b] & \text{if } (x, y) \in O \end{cases} \quad (21)$$

where $d[(x, y), b]$ is some distance measure between pixel (x, y) and b (Rosenfeld and Pfaltz 1968).

Appendix C Dilation and Erosion

A dilation operation “grows” or “thickens” objects by adding pixels to the object boundaries. The mathematical definition of dilation of A by B is defined as follows (Gonzalez et al. 2003):

$$A \oplus B = \{z | (\hat{B})_z \cap A \neq \emptyset\} \quad (22)$$

where $(B)_z$ is the translation of B by the point $z = (z_1, z_2)$, defined as

$$(B)_z = \{b + z | b \in B\} \quad (23)$$

and $\hat{B}$ is the reflection of B , defined as

$$\hat{B} = \{w | -w \in B\}. \quad (24)$$

The output of dilation is a binary image having a value of 1 at each location of the origin of the structuring element, such that the reflected and translated structuring element overlaps at least one 1-valued pixel in the input image.

An erosion operation “shrinks” or “thins” objects by removing pixels on object boundaries. The mathematical definition of erosion A by B is as follows (Gonzalez et al. 2003):

$$A \ominus B = \{z | (B)_z \cap A^c \neq \emptyset\} \quad (25)$$

where A^c is the complement of A (0-valued pixels set to 1-valued and 1-valued pixels set to 0-valued, for a binary image), defined as

$$A^c = \{w | w \notin A\}. \quad (26)$$

The output of erosion is a binary image having a value of 1 at each location of the origin of the structuring element, such that the element overlaps only 1-valued pixels of the input image (i.e., it does not overlap any of the image background).

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Approximate Technology

► Multidisciplinary Design Optimization (MDO)

Aquaculture Structures: Experimental Techniques

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Synonyms

Force oscillation experiment; Seakeeping experiment; Towing experiment

Definition

Various experimental techniques for aquaculture structures have been developed for different purposes. Towing and force oscillation tests on an integrated fish cage system or a single fish net plane are utilized to investigate the mathematical models of the hydrodynamic force coefficients in current and waves, respectively, which are an essential input for numerical simulations. Such experiments can also be implemented in a circulating water channel, which can produce uniform and oscillatory currents. Seakeeping experiment technologies for aquaculture structures in waves and currents are also very popular and have mainly been used for the analysis of the dynamic performance of integrated structures; these technologies can provide important verification data for numerical methods.

Scientific Fundamentals

Experimental Techniques for Aquaculture Structures in Current

In a towing test, an integrated fish cage model or a fish net panel model is installed beneath a towing carriage driven by servo motors. The carriage normally runs at a constant velocity to simulate the pure current in a real sea state. The main purpose is to investigate the hydrodynamic force or deformation for integrating a gravity fish cage or the drag forces of a fish net or semi-submersible fish cage. The circulating water channel test is an alternative approach to the towing test in the simulation of current. The difference is that the model is fixed, and a current with a constant velocity is generated in the water channel.

Experimental Study of a Fish Net Panel in Current

An integrated fish cage can be assumed to be a combination of a number of independent net panels. The drag force on a net cage is then given as the sum of the drag forces on the different net panels. One of the most popular towing experiments on a fish net panel model was conducted by Aarsnes, Rudi, and Løland (Aarsnes et al. 1990). These researchers investigated the flow through and around fish net panels and derived the mathematical function of the drag and lift force coefficient for the screen model with the relation of the solidity ratio and incident current angle:

$$\begin{aligned} C_D &= 0.04 + (-0.04 + 0.33Sn + 6.54Sn^2 - 4.88Sn^3) \cos \theta \\ C_L &= (-0.05Sn + 2.3Sn^2 - 1.76Sn^3) \sin 2\theta \end{aligned}$$

(1)

Here, θ refers to the angle between the vector of the fish net plane and the current direction. Sn represents the solidity ratio. The drag and lift forces acting on the fish net panel can then be calculated as

$$F_D = \frac{1}{2} \rho C_D A U^2; F_L = \frac{1}{2} \rho C_L A U^2 \quad (2)$$

where A is the projected area of the net panel and U represents the current velocity.

Experimental studies of fish net panels have also been investigated by many other researchers. Tsukrov et al. (2011) performed an experimental study of the hydrodynamic forces of copper nets with different solidities, as shown in Fig. 1. Swift et al. (2006) conducted a field test of the drag forces acting on both clean and biofouled net panels. The field testing apparatus is shown in Figs. 2 and 3.

Experimental Study of an Integrated Fish Cage in Current

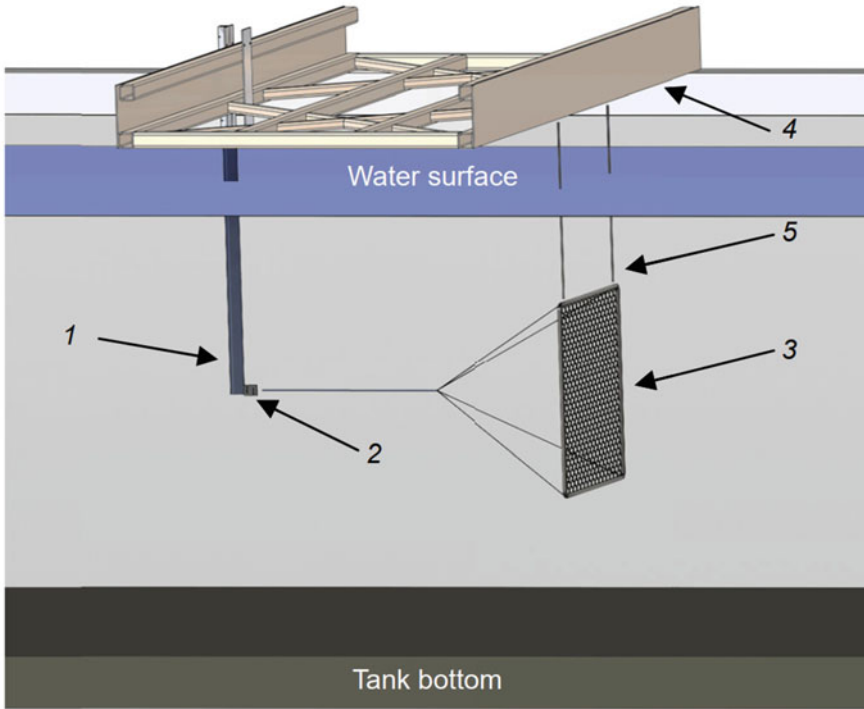
Towing experiments with integrated fish cages, for instance, a gravity fish cage or a semi-submersible fish cage, have been conducted by many researchers. The main purpose also includes the investigation of the deformation of the fish net, which is slightly different from that in a towing test of the fish net panel.

Lader and Enerhaug (2005) carried out an experiment of a gravity fish cage with a scale ratio of 1:7.1 in uniform current in a water channel driven by four impellers, as shown in Fig. 4. Three different sets of weights with cylindrical shapes made of steel were used in the experiment. The tension in the mooring lines was measured by load cells, and simultaneously, the deformation of the cylindrical net was captured by cameras. The dependency between the forces and the geometry and the effects of the weights on it were the main focus. Kristiansen et al. (2015) performed a model test of a floating fish cage in current in a towing tank, as shown in Fig. 5. The current was simulated by running a towing carriage in calm water.

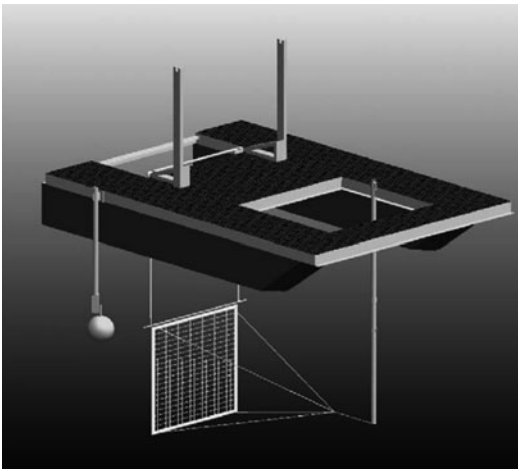
Experimental Techniques of Aquaculture Structures in Waves

Experimental Study of Hydrodynamic Forces on a Fish Net Panel in Waves

Uncertainties exist if the empirical formula that was developed based on towing experimental data is utilized for predicting the hydrodynamic forces acting on a fish net in waves. Therefore, many researchers have conducted experimental investigations on the hydrodynamics of fish nets in waves. In the experiment, the fish net panel is normally pretensioned in a rectangular steel



Aquaculture Structures: Experimental Techniques, Fig. 1 Experiment apparatus on the hydrodynamic forces of copper nets with different solidities (Tsukrov et al. 2011)



Aquaculture Structures: Experimental Techniques, Fig. 2 Field test apparatus on the drag forces acting on both clean and biofouled net panels (Swift et al. 2006)

framework and then fixed in a water channel. By generating regular or random waves, the wave elevation before and after the net and the

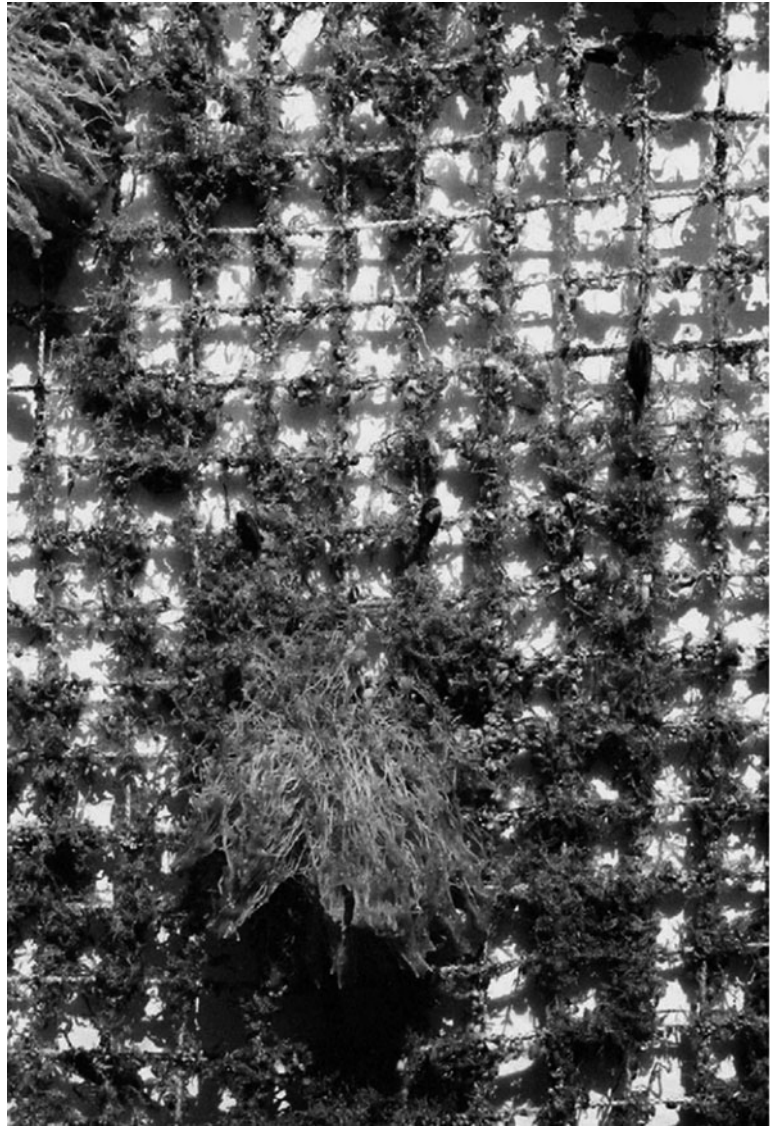
hydrodynamic forces acting on the net panel are measured simultaneously. Compared with the model in the current-only case, two new key parameters should be considered, i.e., the Keulegan–Carpenter (KC) number and wave steepness. These parameters can be written as

$$KC = \frac{\pi H}{D}, \quad s = \frac{H}{L} \quad (3)$$

where H and L refer to the wave height and wavelength, respectively, and D represents the net twine diameter.

Song (2006) performed an experimental study on the effect of horizontal regular waves on fish net panels, as shown in Fig. 6. By using dual series relations, a least squares approximation, and a multiple stepwise regression analysis, this scholar established an empirical relation between the horizontal wave force and the wave period and wave height. Lader et al. (2007) conducted a very similar experiment and measured the forces acting on

**Aquaculture Structures:
Experimental
Techniques, Fig. 3** A
biofouled net panel (Swift
et al. 2006)

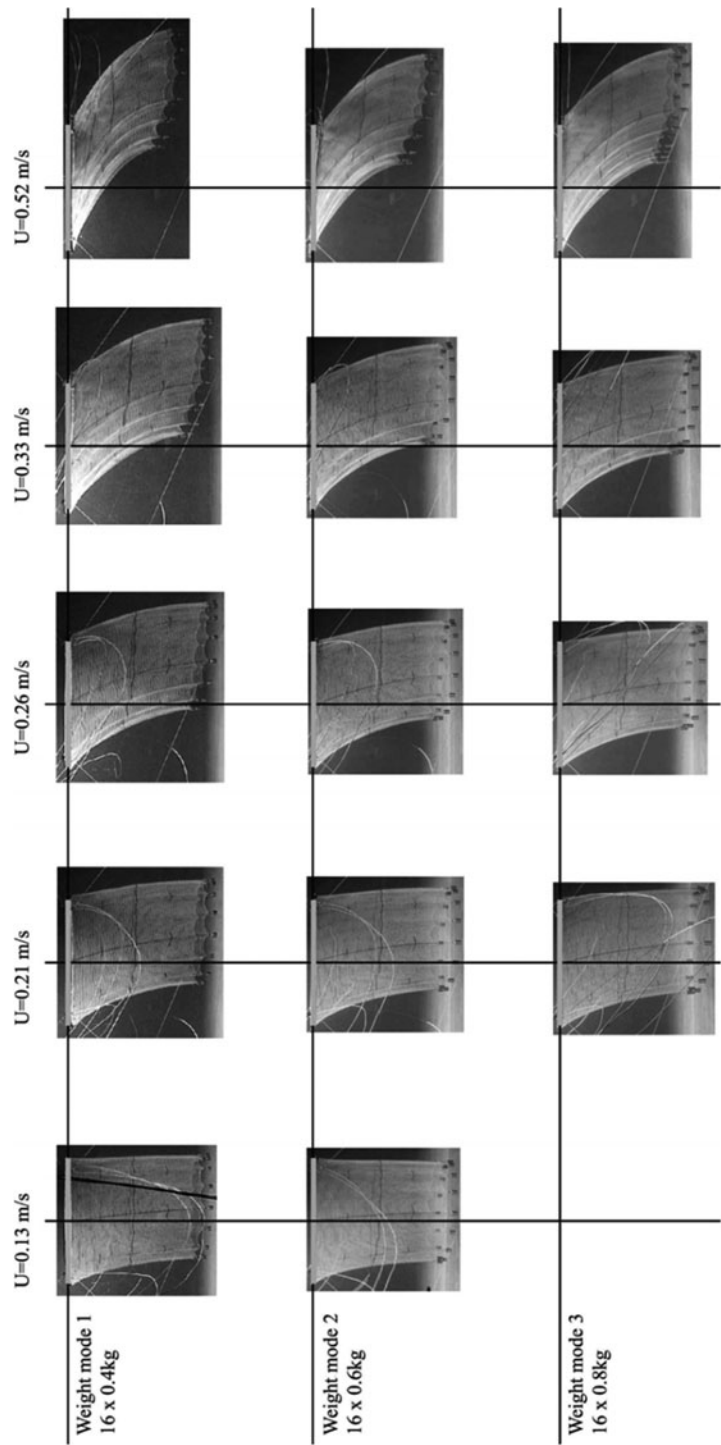


A

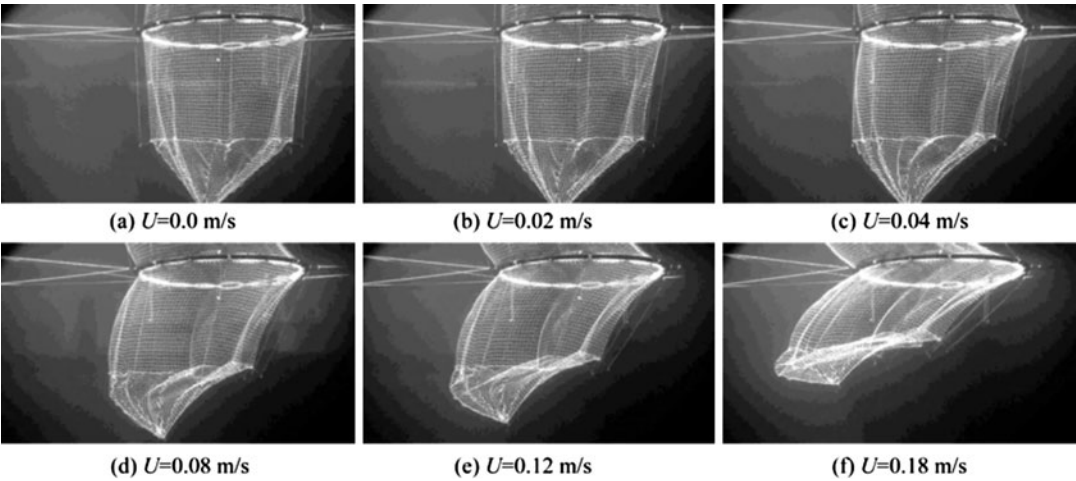
net panels with three different solidity ratios when exposed to five different regular wave cases, as shown in Fig. 7. The measurements were compared with three different empirical load models, i.e., a simple drag model, the Aarsnes model, and the Tsukrov model. The simple drag model yielded the best results for two of the net cases for the horizontal forces, while the Aarsnes model gave the best results for the high solidity net. For the vertical force, the Tsukrov model provided the best results.

Experimental Study of Hydrodynamic Forces on a Semi-Submerged Cylinder in Waves

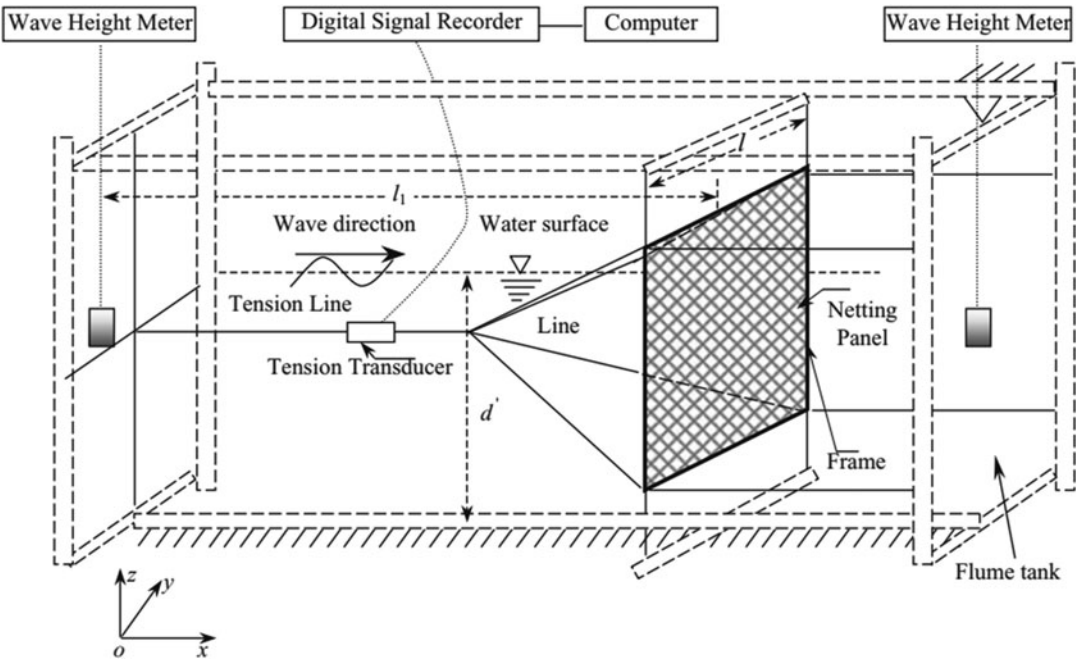
In addition to the fish net, the gravity fish cage also contains a floating collar, which is the main structure resisting wave loads. Slender structures float on the free water surface and continuously undergo locomotion and elastic deformation in waves and current. Although the floating collar is usually made of a circular cylinder and substantial experimental data based on Morison equations are available for the circular cylinder's



Aquaculture Structures: Experimental Techniques, Fig. 4 Deformation of the net cylinder for different weigh configurations and current velocities in a water channel (Lader and Enerhaug 2005)



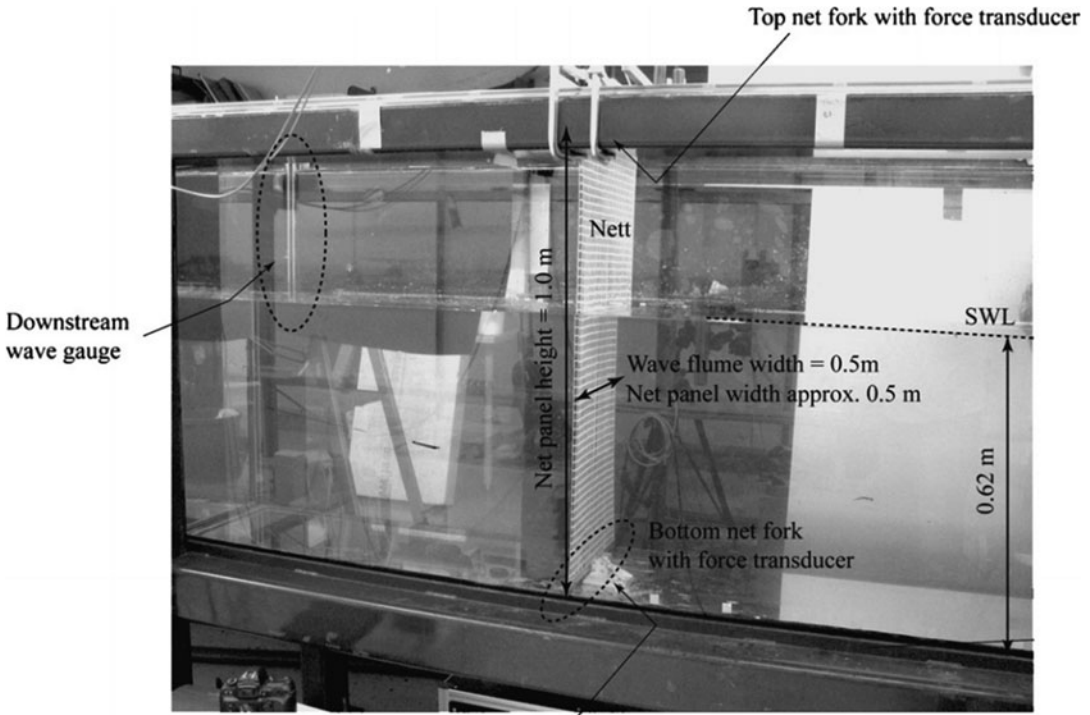
Aquaculture Structures: Experimental Techniques, Fig. 5 Deformation of the gravity fish cage from towing test in calm water (Kristiansen et al. 2015)



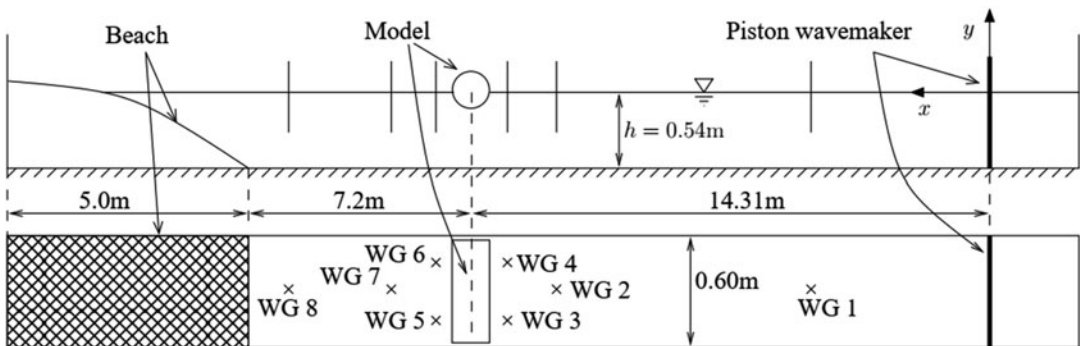
Aquaculture Structures: Experimental Techniques, Fig. 6 Experimental study of fish net panels in regular waves (Song 2006)

hydrodynamic coefficients, it is still difficult to accurately predict the hydrodynamic forces on the floating collar due to its particularities compared with the fully immersed circular cylinder, i.e., the free surface effect.

To understand the hydrodynamic characteristics of floating cylinders, Kristiansen and Faltinsen (2008) conducted research on wave loads on a fixed circular cylinder and the heave and sway motion of a floating cylinder in regular



Aquaculture Structures: Experimental Techniques, Fig. 7 Experiment study of hydrodynamic forces acting on net panel under regular wave (Lader et al. 2007)



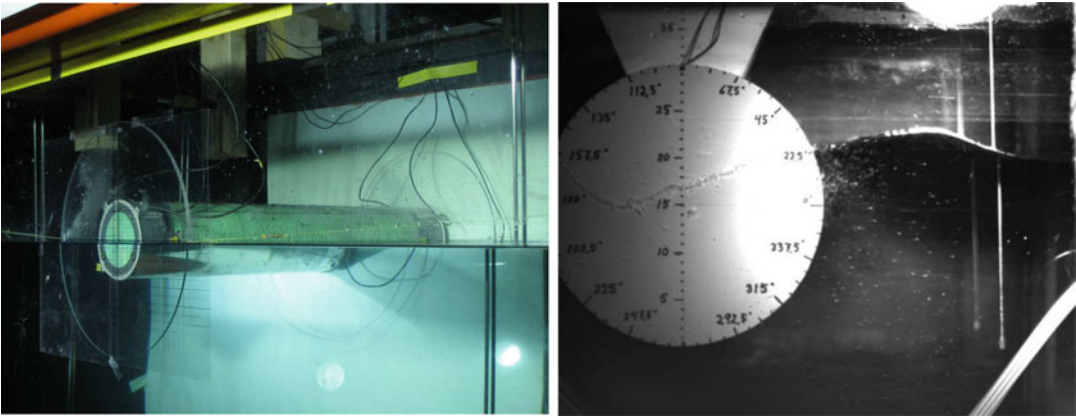
Aquaculture Structures: Experimental Techniques, Fig. 8 Experimental study on the wave loads on a cylinder (Kristiansen and Faltinsen 2008)

waves in a wave flume shown in Figs. 8, 9, and 10. In such experiments, the circular cylinder segment model can be fixed or elastically restrained by mooring lines with a specified water depth. As the model is fixed, the main objective is to focus on nonlinear effects such as wave overtopping on the models and how such events influence the wave loads. The experiment makes significant

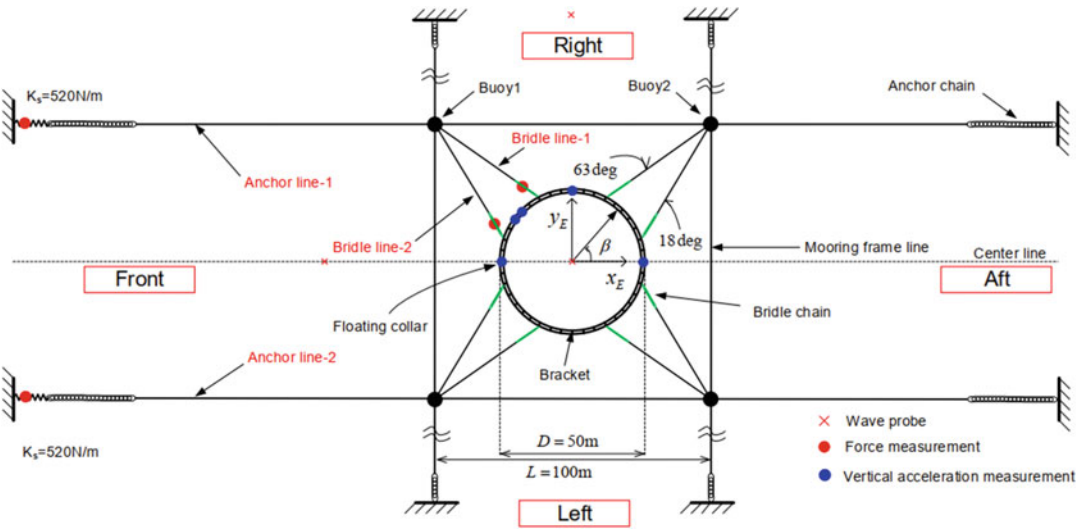
contributions to the design of gravity fish cages with partially submerged floating collars.

Experimental Study of the Dynamic Response of Fish Cages in Waves

Experimental studies of the dynamic response of a whole fish cage model in waves are essential for the design of aquaculture structures. A physical



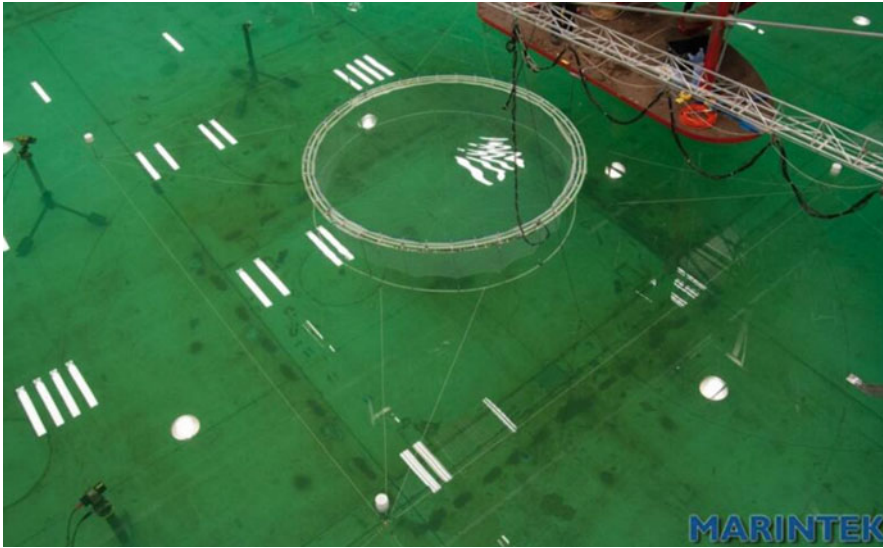
Aquaculture Structures: Experimental Techniques, Fig. 9 Photos of a partially submerged cylinder in waves (Kristiansen 2010)



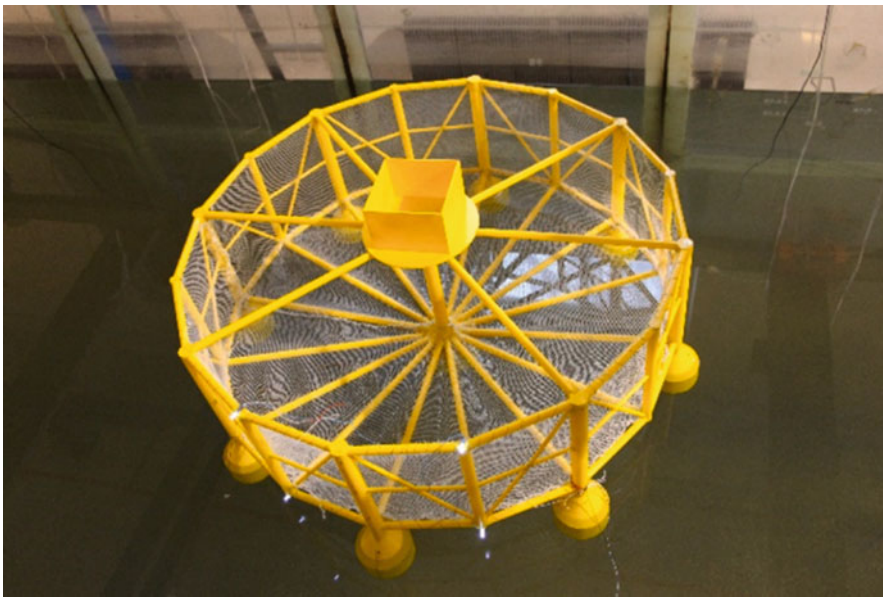
Aquaculture Structures: Experimental Techniques, Fig. 10 Model test of gravity fish cages containing floating tubes (Shen et al. 2018)

model normally contains the fish cage, and the mooring system is arranged in an ocean basin. Regular or irregular waves can be generated by wave makers. The motions of the fish cage model and the tensions in the mooring lines were measured, providing important test data to validate the accuracy of the numerical methods. The procedure is quite similar to that for traditional marine structures. The main difference is the existence of a super-flexible fish net. The physical model of traditional marine structures, such as ocean

platforms or ships, is considered to be a rigid body, and wave actions are dominated by diffraction and mass forces. However, the fish net encounters very large deformations, and the viscous force is dominant. Many researchers have conducted various experimental studies on the wave-induced response of different types of fish cages, such as the gravity fish cage, semi-submersible fish cage, and closed flexible cage. Nygaard (2013) carried out a model test of gravity fish cages containing



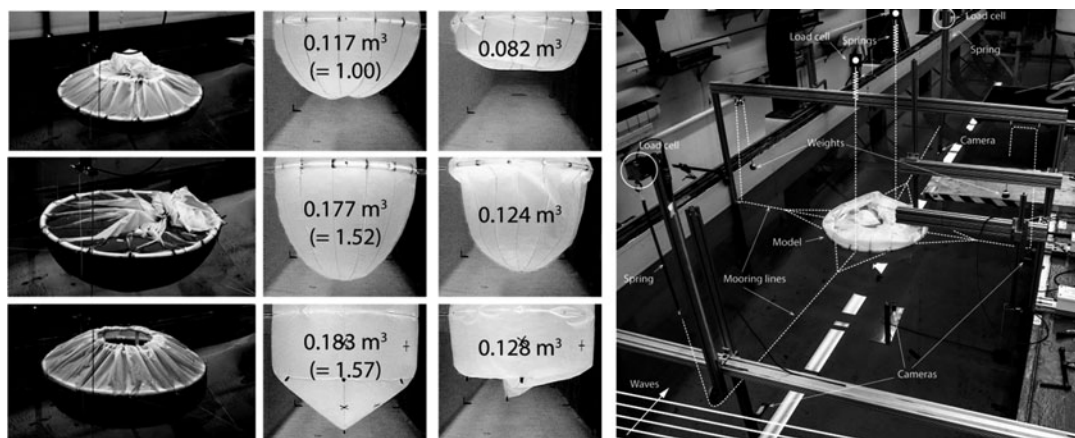
Aquaculture Structures: Experimental Techniques, Fig. 11 Model test of a gravity fish cage (Shen et al. 2018)



Aquaculture Structures: Experimental Techniques, Fig. 12 Model test of a semi-submersible fish cage (Zhao et al. 2019)

floating tubes, an elastic sinker tube, a cylindrical net cage, and a mooring system with a scale ratio of 1:16 in irregular wave, regular wave, and current-only cases, as shown in Figs. 10 and 11. Eight linear accelerometers were arranged to capture the deformation of the floating tubes. In addition, axial force transducers were used to measure

the tensions in the mooring lines. Zhao et al. (2019) performed an experimental study of the hydrodynamic characteristics of a semi-submersible fish cage in waves with different wavelengths, as shown in Fig. 12. The motion response and the mooring line tension are measured and used to verify the rationality of the numerical model.



Aquaculture Structures: Experimental Techniques, Fig. 13 Model test of a closed flexible bag (Lader et al. 2017)

Lader et al. (2017) investigated the behavior of closed flexible bags experimentally which is a very new and novel design proposed due to increasing environmental considerations as salmon lice, escape of farmed fish, and release of nutrients. Three different bag shapes with a scale ratio of 1:30 in Fig. 13 were studied: cylindrical, spherical, and elliptical. The response in vertical direction and the tension in the mooring line were measured simultaneously. The deformation of the bag was only recorded qualitatively by using cameras in the test, and no quantitative data were derived from the images.

Forced Oscillation Experiment of Aquaculture Structures

It is hard to get large KC number as real sea state via traditional wave making method. Currently, two ways are available for this problem: one is to force the model harmonically oscillating in still water; another is to fix the model in the oscillatory flow in a U-shaped water tunnel. The latter one is very creative and widely used for the fully immersed cylinder in oscillatory flow. Yet for the cylinder floating on the free surface, e.g., floating collars of the fish farming structures, the waves and current will be always co-existent (where current generally maintains a quite large velocity and hence becomes non-negligible); thus, the wave making and overtopping on the cylinder should be taken into consideration. As the U-shaped water tunnel cannot simulate the free

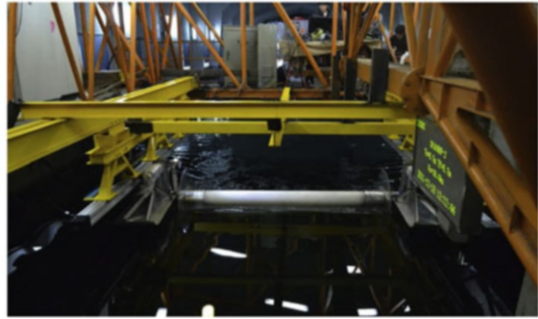
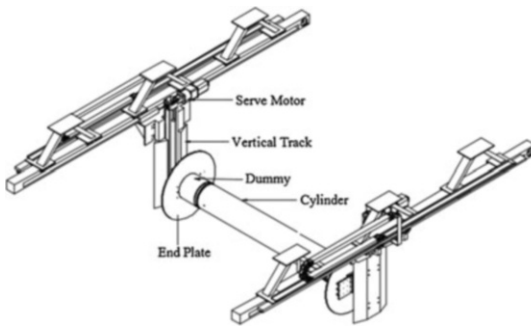
surface, the first experimental method is presently introduced. Specifically, the forced oscillation of the cylinder fixed on the free surface is used to simulate the water particle motions around the floating cylinder in waves; and the forward towing of the cylinder is used to simulate the steady current speed, so that the effect of the wave making and overtopping can be easily achieved in the experiment.

Fu et al. (2013) and Ren et al. (2019) investigated the hydrodynamic characteristics of the segment model of a floating collar and floater-net system in oscillatory and steady flows through forced oscillation experiments in a towing tank, as shown in Fig. 14. The Reynolds number is from 0.5×10^5 to 7.5×10^5 , and the KC number varies from 6.28 to 37.70. The effects of KC number, Reynolds number, reduced velocity, and overtopping phenomenon on the drag and inertia force coefficients are studied. Also, hydrodynamic forces on the floater and net are compared in detail to analyze their respective contributions to the total drag force.

Key Applications

Experiment of a Fish Net Panel in Waves and Current

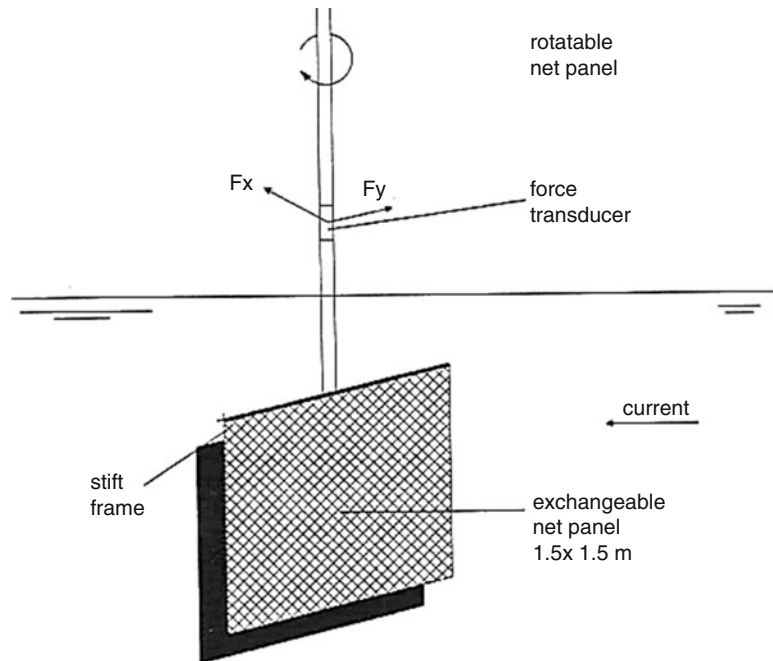
An extensive model test of a single net panel in current-only and combined wave and current conditions was performed by J.V. Aarsnes, H. Rudi,



Aquaculture Structures: Experimental Techniques, Fig. 14 Forced oscillation experiment of a partially submerged floating collar (Fu et al. 2013; Ren et al. 2019)

**Aquaculture Structures:
Experimental
Techniques, Fig. 15**

Model test of a single net panel (Løland 1991)



and G. Løland (Aarsnes et al. 1990) at the Ocean Basin Laboratory of Marintek. The experimental apparatus is shown in Fig. 15. The tests were run in a towing tank by mounting the physical model with six different incident angles on the towing carriage. Three different constant towing speeds were used. A full-scale net with dimensions of 1.5×1.5 m and six different solidity ratios was used. A total of 108 possible combinations were carried out for the current-only condition. In the combined wave and current condition, different wave heights, wave periods, and current velocities were used. During the test, the hydrodynamic

forces acting on the net panel in the normal and tangential directions were measured, which were named the drag and lift forces.

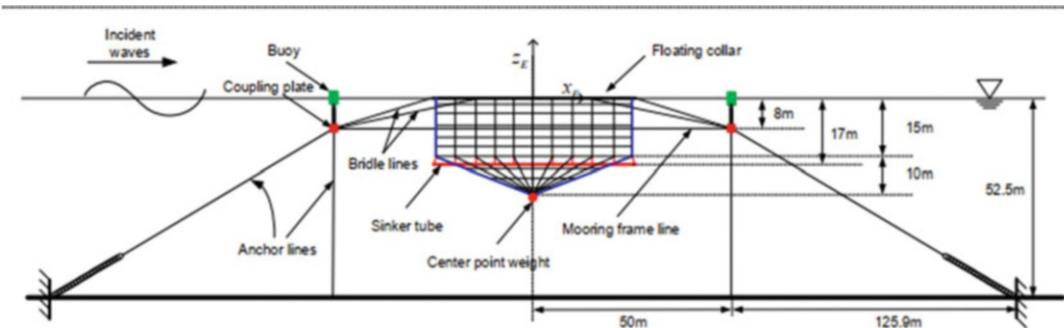
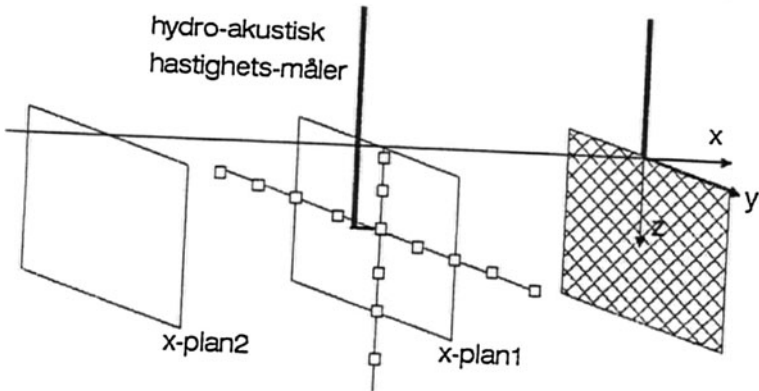
By using the experimental data, the drag and lift force coefficients can then be derived by using, for instance, the least squares method, and the empirical relations between the hydrodynamic force coefficients, and the solidity ratios and incident angle can be established, as shown in Eq. 1.

As the flow passes through the fish net, the velocity decreases significantly due to the shielding effect. The quantification of the velocity

reduction is very important for the numerical analysis of the whole fish cage. Therefore, in the model test of the single net panel, the velocity distribution behind the net was also measured, as shown in Fig. 16. An empirical function between the current velocity reduction factor and the drag coefficient was proposed.

**Aquaculture Structures:
Experimental
Techniques, Fig. 16**

Experimental apparatus on the velocity distribution behind a net panel (Løland 1991)



Aquaculture Structures: Experimental Techniques, Fig. 17 The experimental arrangement of a gravity fish cage (Shen et al. 2018)



Aquaculture Structures: Experimental Techniques, Fig. 18 From left to right: the gravity fish cage model, the floating collars, the elastic sinker tube, and the buoy (Shen et al. 2018)

Norway, including the floating collars, a fish net cage, an elastic sinker tube, a center point weight, and a mooring system. The mooring system is quite complex and is composed of bridle lines, anchor lines, mooring frame lines, buoys, coupling plates, and anchor lines. The scale ratio is 1:16, and Froude scaling with geometric similarity except for the net twines was adopted. For the net twines, the diameter is too small, and the geometric similarity is almost impossible to satisfy. Only the solidity ratio was designed to be the same as the real-scale structure to ensure the similarity of the hydrodynamic forces acting on the fish net.

A comprehensive measurement system was arranged during the experiment, which includes eight linear accelerometers, fourteen force transducers, and three wave gauges. The tension forces in the bridle lines, anchor lines, ropes connecting the buoy and coupling plate, ropes connecting the sinker tube and net, and ropes in the net cage were measured simultaneously. Furthermore, the motion response and the deformation of the floating collar were captured by using accelerometers. The dynamic response of the gravity fish cage under long-crested irregular

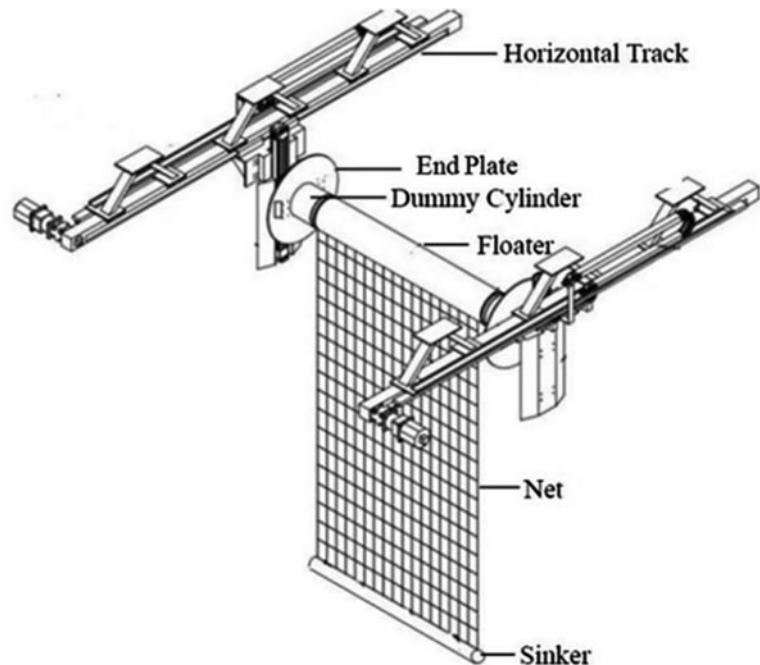
waves, regular waves, and current-only cases was studied.

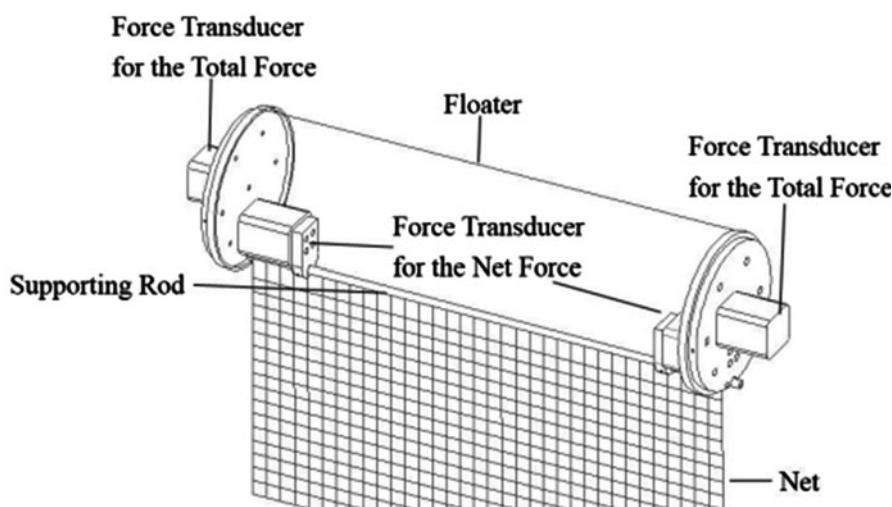
Force Oscillation Experiment of Floater-Net System

A forced oscillation experiment of floater-net system was carried out by Fu et al. (2014) in a towing tank in Shanghai Ship and Shipping Research Institute in China, which is 192 m long, 10 m wide, and 4.2 m deep; and the maximum velocity of the towing carriage is 9 m/s. The experimental apparatus includes the towing carriage, the forced oscillation apparatus, and a cylinder model, as shown in Fig. 19. The horizontal track length is 3.5 m and the diameter of cylinder is 0.25 m.

The 2 m long smooth cylinder model is manufactured by polypropylene material and installed on the forced oscillation apparatus, passing through two three-component force transducers in use to measure the hydrodynamic force of the floater-net system, as shown in Fig. 2. The model will harmonically oscillate in horizontal direction driven by a servo motor. Oscillating displacement and velocity are controlled according to the experimental design and measured by the encoders of the servo-motor.

Aquaculture Structures: Experimental Techniques, Fig. 19 The experimental apparatus for forced oscillation of a floater-net system (Fu et al. 2014)





Aquaculture Structures: Experimental Techniques, Fig. 20 The arrangement of the measurement system (Fu et al. 2014)

The net is manufactured by nylon material and is 2 m wide and 3.5 m length. The netting is knotless with a mesh size of 20 mm and twine thickness of 2 mm. Mounted as square meshes, the solidity ratio of the netting is 0.4. The net structure is supported by a slender steel rod which is installed on three-component force transducers used to measure the force of net structure, as shown in Fig. 20.

Cross-References

- [Aquaculture Structures: Numerical Methods](#)
- [Modern Aquaculture Structures](#)
- [Net Structures: Biofouling and Antifouling](#)
- [Net Structures: Design](#)
- [Net Structures: Hydrodynamics](#)
- [Traditional Aquaculture Structures](#)

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Aquaculture Structures: Numerical Methods

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Synonyms

Computational fluid dynamics (CFD); Finite element method (FEM); Hydroelasticity theory; Modal superposition method; Morison-force model; Screen-force model

Definition

For *aquaculture structures*, numerical analysis with proper methods is essential for safe structural design. In a numerical method for aquaculture system, a *structural model* and a *hydrodynamic-force model* are needed to describe the structure and evaluate the loads on the structure, respectively. A typical fish-farm system comprises different components, and numerical method for each individual component is required.

Scientific Fundamentals

Traditional Fish-Farm Concepts

There exist a variety of fish-farm concepts, and these concepts can be classified in different ways. For example, Loverich and Gace (1997) classified sea cages into four types based on the structural means used to hold the shape of the cage, i.e., *gravity cages*, *anchor-tension cages*, *semi-rigid cages*, and *rigid cages*. The gravity cages are by far the most widely used concepts in the fish farming industry and thus are explained in detail here. They rely on the force of gravity to maintain net volume, by providing a surface buoyancy system and an underwater weighting system for the net. There exist different materials and geometries of the buoyancy system. The most commonly used are the *circular plastic floaters* and the *interconnected hinged steel floaters*, as shown in Fig. 1.

The cage with circular plastic floaters has been widely investigated; thus a detailed description is given here. The cage typically comprises a *floating collar*, a *net cage*, a *sinker tube*, and a *mooring system*. The floating collar provides buoyancy for the whole system and is common with two nearly semi-submerged concentric circular tubes. The two tubes are held together by steel brackets. The usage of the net is to protect the fish from predators and provide a suitable habitat. The net is highly flexible, so it will experience deformation when subjected to external loads from waves and current. A bottom weight ring (sinker tube) is often attached to the cage bottom to ensure sufficient volume of the net cage. In reality, there exist multiple cages at real sites with cages arranged in mooring grid in single or double rows. This will have an influence, for instance, on the steady inflow (current) due to the *shadowing effects* from the upstream cages, compared with a single cage system. The mooring system normally used by the circular cages is a square-shaped grid system held on the seabed with an array of mooring lines. The mooring lines include the anchors, ground chains, ropes and related shackles, and buoys (Shen 2018).



Aquaculture Structures: Numerical Methods, Fig. 1 Left: circular plastic cage. Right: hinged square steel cages. (Source: www.akvagroup.com/news/image-gallery)

Novel Fish-Farm Concepts

There exist multiple challenges for the fish-farm industry nowadays. One of the big problems has been the rise of sea lice. It has become increasingly serious as the sea lice have become more resistant to chemicals used to treat them. Escaped fish is another concern. In addition, due to limited nearshore area and increasing impact to the local ecosystem, the aquaculture industry is trying to move the marine fish farms from nearshore to more exposed sea regions where waves and current are stronger. To meet the challenges faced by the fish-farm industry, new fish-farm designs are emerging. *Closed fish-cage concepts* are proposed to have a better control over the water quality and especially to avoid the problem of salmon lice. *New open-cage fish-farm* concepts are also under development to operate in more exposed sea regions or even in open sea, by combining the best of existing technology and solutions from the Norwegian fish farming industry and the offshore oil and gas sector (Shen 2018).

Historical Development (Traditional Fish Farms)

Many numerical and experimental investigations have been performed during the past decades to examine the behavior of traditional aquaculture fish farms under waves and current. The fact that the netting may have ten million meshes limits the implementation of computational fluid dynamics (CFD) and complete structural modeling. Therefore, rational simplifications are often needed. Important literatures for the modeling of the

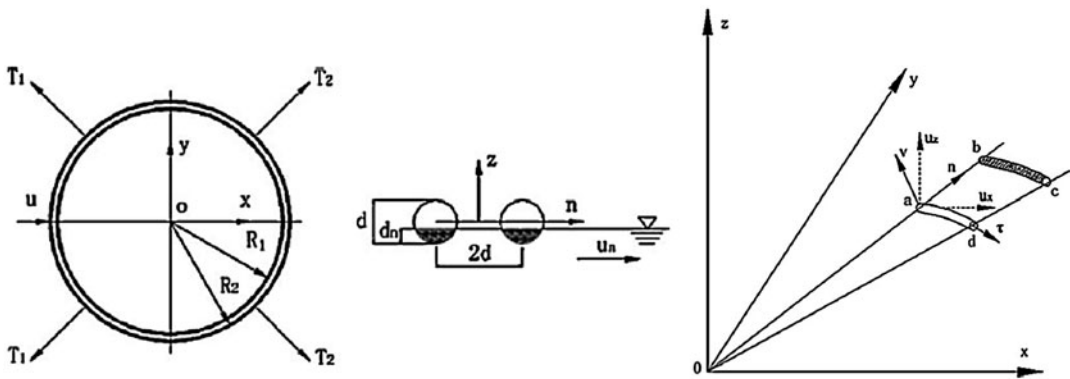
different components of a traditional fish-farm system will be introduced. The focus will be on the *structural* and *hydrodynamic* modeling of the floating collar and the net cage.

The Floating Collar

The floating collar of a sea-cage is typically composed of two concentric elastic pipes interconnected by brackets and a handrail. The structure is often modeled into a single or double column pipe without the handrail. When estimating the hydrodynamic loads on the floater, *rigid body* is commonly assumed and *strip theory* is adopted, disregarding important three-dimensional (3D) flow, frequency-dependency hydrodynamic coefficients, as well as nonlinear effects in steep waves. *Modified Morison's equation* (Brebba and Walker 1979) was widely employed to calculate the load on each collar segment (see Fig. 2), for example, by Huang et al. (2006), Zhao et al. (2007a), Dong et al. (2010a), and Cifuentes and Kim (2017).

Dong et al. (2010b) tried to incorporate the *elasticity* of the floater into the modeling and proposed an analytical method to analyze the elastic deformations of a circular ring in waves. The governing equations of six degree-of-freedom rigid-body motions and elastic deformations were obtained according to Euler's laws and curved beam theory.

Faltinsen (2011) pointed out that the diffraction and radiation effects may matter for the response of the floater. He derived an analytical formula for the added mass for different modal of an elastic floater based on a slender body theory



Aquaculture Structures: Numerical Methods, Fig. 2 Left: a sketch of the float-collar system. Right: a sketch of the float collar mini-segment (Dong et al. 2010a)

and rigid free-surface condition. Li and Faltinsen (2012) calculated the vertical added mass, damping and wave excitation loads on an elastic semi-submersible torus based on low-frequency slender-body theory. They found that three-dimensional (3D) flow, hydroelasticity, and strong hydrodynamic frequency dependency matter on the scale of the torus. The structural response of the floater was modeled using the principle of modal superposition. The numerical model from Li and Faltinsen (2012) was further adopted to investigate vertical accelerations of a moored floating elastic torus without netting in regular waves (Li et al. 2016, 2018b), an aquaculture net cage with floater in regular waves and current (Kristiansen and Faltinsen 2015), and a realistic marine fish farm in waves (both regular and irregular) and current (Shen et al. 2018).

Commercial structural software could also serve as an alternative tool to investigate the deformation of the floating collar. For example, Fu et al. (2007) used the extended three-dimensional (3D) hydroelasticity theory to predict the hydroelastic response of flexible floating interconnected structures, including displacement and bending moments under various conditions. The finite element method (FEM) solver ABAQUS was used as the eigen-solver. Fu and Moan (2012) applied the same theory in the frequency domain to predict the dynamic response of the 5-by-2 floating fish-farm collars in regular waves. Liu et al. (2019) evaluated the structural

strength and failure for floating collar of a single-point mooring fish cage based on FEM solver ANSYS.

Computational fluid dynamics (CFD) calculations were also performed to calculate the hydrodynamic loads on the floater. For instance, Kristiansen (2010) presented model test results in two-dimensional (2D) flow conditions of a semi-submerged floater without netting in regular waves and validated a CIP (Constrained Interpolation Profile)-based finite difference method that solved Navier-Stokes equations for laminar flow. Similar approach was implemented by Bardestani (2017) to investigate the snap loads in the netting in two-dimensional conditions in waves.

The Sinker Tube

The sinker tube can be modeled in a similar way as the floating collar. For instance, the structure can be described by the curved beam equations, and the hydrodynamic loads can be estimated by the modified Morison's equation (Shen et al. 2018).

The Net Cage

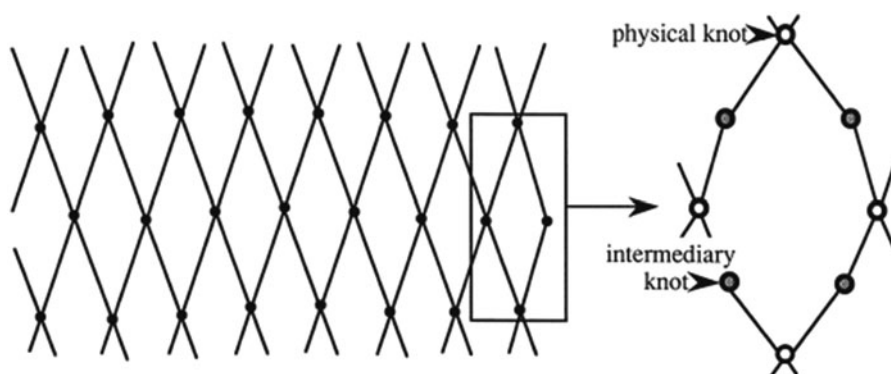
In order to analyze a net in current and waves, one needs a *structural model* and a *hydrodynamic force model*. In *structural analysis* of nets, it is difficult to fully simulate the strong fluid-structure interaction between moving seawater and the highly flexible net. To reduce complexity, the nets are often represented by some *equivalent*

structural elements, e.g., truss, beam, cable, or spring elements. To guarantee that equivalent twine has the same drag force, buoyancy, gravity, and stiffness as the original netting, the equivalent element is given a certain length and diameter such that it has the same projected area as the physical net it represents. In this way, the number of twines can be reduced to a great extent.

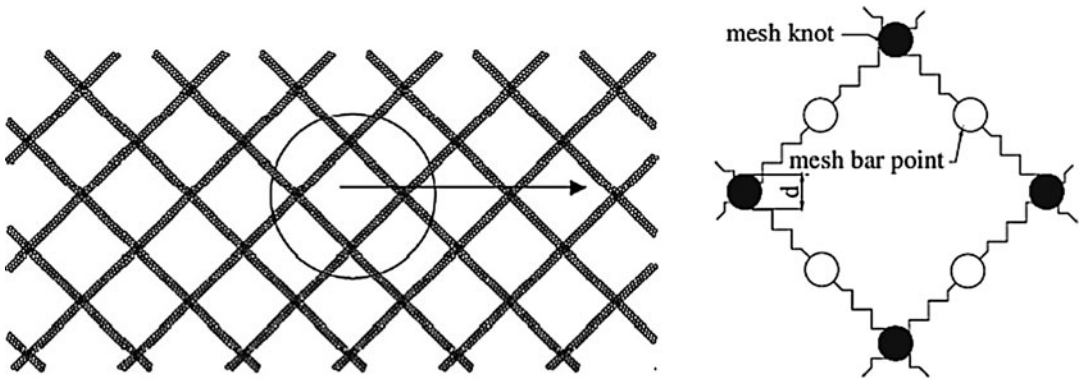
For the hydrodynamic loads acting on the net, although computational power has increased significantly, it is still not practical to apply CFD approach to simulate the full nets system with order of ten million. Two main types of hydrodynamic-force models are typically applied to predict viscous loads on nets, i.e., *Morison model* and *screen model*. The first method is to model the net as individual knots and twines, calculating the total drag on the net as sum of the drag contributions of individual elements, without considering the interaction between twines. *Morison's equation* is used to calculate the cross-flow force on each twine based on cross-flow principle. There are two limitations for the Morison model: (1) a drag model based on the cross-flow principle cannot be justified when the inflow angle is larger than about 45° and (2) the interaction between the twines is not accounted for. The second method is to assume that the net is made of surface elements with properties accounting for the underlying twine and knot structure. The hydrodynamic force is decomposed into a normal and a tangential force component on the net panel (Kristiansen and Faltinsen 2012).

The Morison-force model: The literatures described next applied *Morison-force model* with different structural modeling. Theret (1993) proposed an original 3D numerical model for modeling fishing nets in steady current using rigid cylindrical bar/truss elements. Bessonneau and Marichal (1998) generalized the method proposed by Theret (1993) to investigate the steady shape of submerged trawl nets. The rigid net twines were further divided into two sub-elements, allowing the twines to buckle/loose when subjected to compression (see Fig. 3). The *mesh grouping method* was proposed to reduce the number of unknowns in which several actual meshes were grouped into a fictitious equivalent mesh having the same specific mass, apparent weight, and approximately the same drag resistance.

Li et al. (2006) developed a numerical model, based on a *mass-spring method* (see Fig. 4) to investigate the hydrodynamic behavior of submerged plane nets in current. The method is similar to Bessonneau and Marichal (1998), except considering also the netting elasticity. The fishing nets were modeled as a series of lumped point masses that were interconnected with springs without mass. The model was validated and applied to several other problems, such as a three-dimensional cage with rigid floater in current (Zhao et al. 2007a), in combined regular waves and current (Zhao et al. 2007b) and in irregular waves (Dong et al. 2010a). The method was also implemented to the analysis of



Aquaculture Structures: Numerical Methods, Fig. 3 Decomposition of rigid elements of diamond mesh nettings (Bessonneau and Marichal 1998)



Aquaculture Structures: Numerical Methods, Fig. 4 The schematic diagram of the mass-spring model (Li et al. 2006)

hydrodynamic behavior of multiple cages in combined wave-current flow (Xu et al. 2013). Similar strategy was chosen by Lee et al. (2008) to simulate a fish cage system subjected to current and waves.

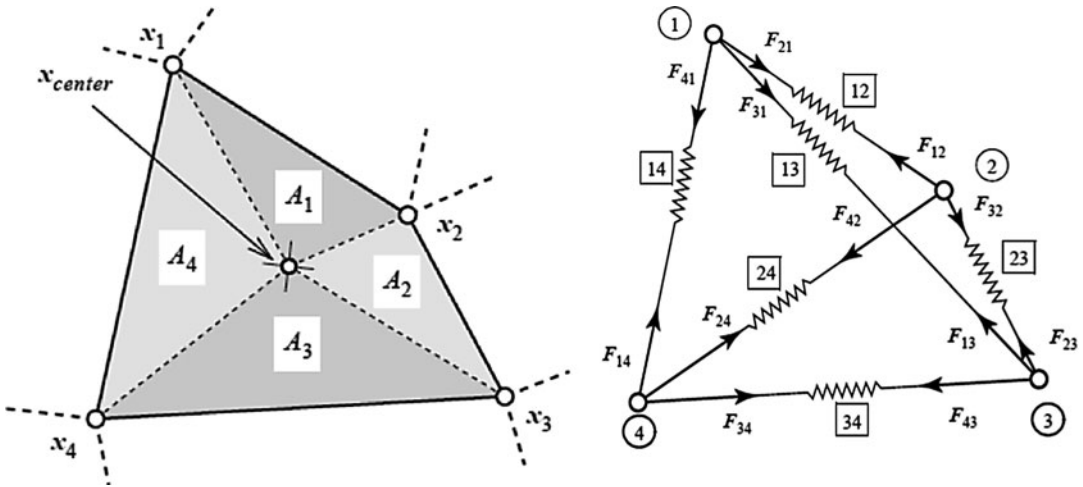
Tsukrov et al. (2003) showed that hydrostatic and hydrodynamic forces exerted on the moving netting, including wave and current-related water motion, cannot be exactly represented by *equivalent structural elements*. They developed a special *consistent net element* to model net panels or their parts. The method was further applied, for instance, to the analysis of a four-cage grid mooring for open ocean aquaculture (Fredriksson et al. 2003, 2004), the prediction of the dynamic response of a single-point moored submersible fish cage under currents (DeCew et al. 2010), and the evaluation the distribution of tension in a two-dimensional net panel (Fredriksson et al. 2014).

Commercial FEM program could also be used to analyze the deformation of the net twines. For example, Gignoux and Messier (1999) proposed an algorithm based on the introduction of the mapping coefficient and successfully applied the model to aquaculture nets using ABAQUS/Aqua beam elements. Moe et al. (2010) used ABAQUS to analyze the deformation of a net cage in currents. ABAQUS was also employed to analyze nonlinear hydroelastic of an aquaculture system, subjected to current, regular, and irregular waves (Li et al. 2013a, b).

The screen-force model: The literatures next describe the development of *screen-force models*, which are gaining increasingly attention. Schubauer et al. (1950) derived theoretical models based on experimental data for the normal and tangential forces acting on screens inclined to steady ambient flow. The examined inclination angle varies between 0° and 45° . Milne (1972) derived an empirical formula for the drag coefficient of plane nets in a steady current as a function of net geometry.

Løland (1991) presented empirical formulas for the drag and lift on planar net panels, known as *Løland's formulas*. The drag and lift coefficients were given as function of solidity ratio and inflow angle for a limited range of *solidity ratio* (0.13–0.317). The *solidity ratio* of a screen is the ratio of the area projected by the screen on a plane parallel to the screen, to the total area contained within the frame of the screen. The *inflow angle* is the angle between current velocity and the normal direction of net panel. The coefficients in the formulas were determined by curve fitting experimental data from Rudi et al. (1988) obtained by towing net panels.

Lader et al. (2003) proposed a dynamic model for 3D net structures exposed to waves and current, based on a *super-element formulation*. In the method, the net structure was divided into four-sided super-elements interconnected in each corner node (see Fig. 5). The hydrodynamic and structural forces were calculated on each super-



Aquaculture Structures: Numerical Methods, Fig. 5 Super-element. Left: element area. Right: the structural model of the element. Node numbers are indicated in circles and spring numbers in squares (Lader et al. 2003)

element, and the total forces were collected in each node where the equations of motion were solved. The hydrodynamic forces on each super-element were calculated based on empirical formulas from Løland (1991). The structural forces were calculated by assuming that each element consists of six nonlinear springs, interconnecting each node to the other three. The super-element method was further utilized to calculate the responses of a flexible net sheet in waves and current (Lader and Fredheim 2006).

Huang et al. (2006) developed a numerical model to simulate the dynamic properties of a net cage system in current based on a screen-force method. In their model the net cage was divided into several plane surface elements to calculate the external forces. The external force was calculated on each element and then equally distributed to its nodes. The drag and lift force coefficient for the net plane was based on the Løland's formulas (Løland 1991).

The screen model proposed by Løland (1991) was further generalized by Kristiansen and Faltinsen (2012), taking into account also the effect of *Reynolds number*, i.e., the force components on a panel are functions of *solidity ratio*, *inflow angle*, and *Reynolds number*. The relevant *Reynolds number* is that based on the physical twine diameter. A uniform turbulent wake was assumed inside the cage based on the theoretical

formula proposed by Løland (1991) for cross-flow past a plane net, leading to a *flow reduction* in the rear part of the net cage. The dynamic elastic truss model by Marichal (2003) was used to describe the net structure. The screen-force model was further adopted to investigate an aquaculture net cage with floater in regular waves and current (Kristiansen and Faltinsen 2015), a realistic marine fish farm in waves (both regular and irregular) and current (Shen et al. 2018), the influence of well boat on aquaculture system (Shen et al. 2019a, b), and the influence of fish in a net cage on mooring loads (He et al. 2018).

Recently, Cheng et al. (2020) presented a systematic review of hydrodynamic models for the net cage, including five Morison models and six screen models. All the models were incorporated into a general finite element (FE) solver for comparison purpose and suggestions for selection of suitable hydrodynamic models were provided.

Computational fluid dynamics (CFD): With the increasing computer resources, growing attention is paid to implement the computational fluid dynamic (CFD) approach to evaluate the flow inside and around a fish cage. During CFD simulations, the net is modeled as a porous media with the influence of the net represented by a resistance term, neglecting the detailed geometry of the net structure. The corresponding resistance coefficients can be obtained from empirical formulas

(*Morison model* and *screen model*) or laboratory experiments. For instance, Shim et al. (2009) investigated the flow through and around a circular cylinder cage using the commercial software Fluent. The net and frame were represented by a porous jump boundary condition, and the pressure drop coefficients were determined through towing tests. Similar approach was adopted by Patursson et al. (2010) and Zhao et al. (2013) to model the flow around a fixed net panel. Attempts have also been made to take the influence of the structural deformation into consideration by coupling the structural model with the porous media model to achieve fluid–structure interaction (FSI) analysis (see Bi et al. (2014a, b) and Yao et al. (2016) for the net under steady current and Chen and Christensen (2017) and Bi et al. (2017) for the net under waves).

The Mooring System

The complete set-up of the mooring system typically comprises ropes and chains, with buoys to support all mooring lines. The buoys are floating circular cylinders and can be modeled by classical seakeeping theory (see Endresen et al. (2014)). Ropes and chains can be treated in a similar way as the net and modeled as elastic trusses/beams with correct diameter, weight, and stiffness. The hydrodynamic forces on the mooring lines can be estimated by a modified Morison's equation based on the cross-flow principle (Shen 2018).

Historical Development (Novel Fish Farms)

As mentioned, the expansion of nearshore fish farming becomes difficult due to limited space and increased environmental impact. This drives the growth of offshore fish farming. Several *new open-cage fish-farm* concepts have been designed and tested to operate in offshore sites. *Closed fish-cage concepts* are also under development to get rid of salmon lice. Important numerical investigations performed recently for these concepts will be described in the following.

Open-Cage Fish Farm

To withstand large wave actions, offshore fish farms are generally with large floating structures,

constructed from steel. These structures can in many ways be compared to platforms from oil and gas industry. Thus, numerical analyses can be conducted through a combination of the same kind of procedures as for offshore platforms and the approach for the fish net.

Bore and Fossan (2015) performed ultimate and fatigue limit state analyses of ocean farming's rigid offshore aquaculture structure. The concept consists of a twelve-sided, cylindrically shaped frame with a net attached to (stretched over) the sides and bottom. This makes up the hull and the fish cage. A Morison model was used for calculation of hydrodynamic forces on all the components. Similar structure was considered by Shi (2019), but with slight modification to operate in Chinese ocean. Morison equation was applied to estimate the viscous loads on slender members. For the pontoon with large dimension, the radiation and diffraction effects may matter; thus the corresponding hydrodynamic forces were evaluated based on a linear potential theory.

Li and Ong (2017) presented a preliminary study of a rigid semi-submersible fish farm for open seas. The farm comprises a lower and upper pontoon with vertical and diagonal bracing in between. The net cage is connected with the lower pontoon and is totally submerged to avoid large wave loads close to the surface. The linear hydrodynamic properties for support structures using different composite models with panel and Morison elements were computed. The nets were simplified as rigid slender elements. This simplification neglects the net deformations and may overestimate the hydrodynamic forces and viscous effects from the nets. Based on the hydrodynamic analysis, linearized frequency-domain and coupled time-domain analysis were performed to predict the extreme motions of the support structure and the extreme tensions in the mooring lines. Similar approach was followed by Li et al. (2018a) to investigate a vessel-shaped offshore fish farm under various wave and current conditions. The effect of current velocity reduction due to shielding effects from upstream cages was also discussed.

Closed-Cage Fish Farm

Closed fish cages can be divided into flexible cages, semi-flexible cages, and rigid. They have typically a vertical symmetry axis at rest. The water inside a closed cage causes statically destabilizing roll and pitch moments. Wave-induced sloshing (interior wave motion) becomes an issue as well-known from many engineering applications. Since marine applications involve relatively large excitation amplitudes, resonant sloshing can involve important nonlinear free-surface effects. The viscous boundary layer damping is very small. However, if breaking waves occur, the associated hydrodynamic damping is not negligible (Faltinsen and Shen 2018).

Linear response of a 2D *closed flexible* fish cage in waves was investigated by Strand and Faltinsen (2019). The motion of the membrane was represented as the sum of rigid body motions and Fourier series with zero displacement at the attachment to the floater. Linear frequency-domain potential flow theory of incompressible water was used both for the interior and external flows. Strong coupling between elastic modes and rigid body motions was observed. The approach was extended to investigate linear wave-induced dynamic structural stress of a 2D *semi-flexible closed* fish cage (Strand and Faltinsen 2020). They compared a quasi-static analysis with a fully coupled hydroelastic analysis to examine the soundness of assuming that the stresses in the structure is quasi-static. They found that the necessity of performing a hydroelastic analysis depends on the stiffness of the structure.

Kristiansen et al. (2018) conducted a numerical and experimental study on the seakeeping behavior of a floating *closed rigid fish cage* with cylindrical shape. The focus was on effects of sloshing on the coupled motions and mooring loads. Dedicated scaled model tests of closed cages in waves were presented and compared with numerical simulations using *linear potential theory* in frequency domain. The results showed that the influence of sloshing on the rigid body motion was significant. Mean wave loads were also affected by sloshing. Similar

cage concept was examined by Tan et al. (2019). The sloshing inside the cage was described by the single-dominant *nonlinear multimodal theory* presented in Faltinsen and Timokha (2009). They demonstrated that the nonlinear swirling waves due to the interactions between different sloshing modes could only be explained by a proper nonlinear theory.

Cross-References

- [Aquaculture Structures: Experimental Techniques](#)
- [Modern Aquaculture Structures](#)
- [Net Structures: Hydrodynamics](#)
- [Traditional Aquaculture Structures](#)

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Arbitrary Lagrangian-Eulerian (ALE)

► Suction Piles

Arctic Pipeline

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Synonyms

[The Arctic Pole pipeline](#); [The North Pole pipeline](#)

Definition

Arctic pipelines refer to pipelines that pass through a permafrost, where the soil or rock remains below 0 °C throughout the year and the formation is sufficiently cooled during the winter to form a layer that persists throughout the summer.

Introduction

The Arctic has enormous resources in permafrost, which need to be developed by being highly aware of the possibility of environmental damage. Permafrost currently accounts for about a quarter of the Earth's land area. Figure 1 shows the distribution of permafrost regions in Alaska, which have large areas of permafrost (Larsen et al. 2008). Arctic pipelines refer to pipelines that pass through a permafrost, where the soil or rock remains below 0 °C throughout the year, and the formation is sufficiently cooled during the winter to form a layer that persists throughout the summer. The main characteristics of arctic pipelines are determined by the climatic conditions pertaining to a given territory, which are described below, and it may be noted that some of these features can be encountered in territories outside of the polar circle, low radiation balance, summer average zero temperature, annual average subzero temperature, large areas of permafrost, wet lands, and an abundance of swamps in plains.

Normally, Arctic pipelines are pipelines transporting gas and oil from production sites to the facilities. What's more, arctic pipeline applications also include unprocessed well fluid and utility lines, processed oil and gas pipelines, combined offshore/overland pipeline systems, subsea field developments, cables, and umbilicals.

The configurations of arctic pipelines include single-walled steel pipe, pipe-in-pipe, flexible pipe and pipe-in-HDPE pipe, etc.

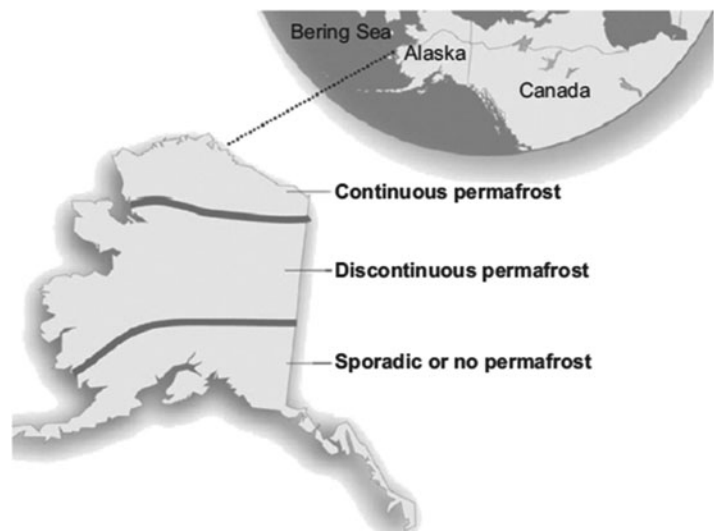
Following BP's discovery of oil in Prudhoe Bay oil field near the Arctic Ocean coast in 1968, the Northstar Pipeline was proposed. This pipeline construction was the first long pipeline construction in Arctic area; prior to this project, only short pipelines were built only in areas such as Siberia and Norman Wells on the Mackenzie River in Canada. With the development of technology and the rise of oil price in the early of the twenty-first century, lots of Arctic pipeline projects were constructed in recent years.

Advancement of Arctic Pipeline Design Technologies

In the Arctic pipeline design, it requires emerging technologies to balance between the efficient and economical design and installation of pipelines in the Arctic. Some early analyses used to evaluate Arctic loads on pipelines apply to those specific projects; but as industry enters more severe Arctic conditions, advancements need to continue to ensure a proper balance between safety and economics. Expanding international knowledge about Arctic conditions, improvements in material behavior, advances in analytical techniques,

Arctic Pipeline,

Fig. 1 Arctic areas of Alaska permafrost coverage (Yong Bai et al. 2014)



and broad acceptance of progressive design concepts, such as strain-based design and the implementation of reliable Arctic operational strategies, enable the consideration and development of other offshore Arctic prospects (Paulin and Caines 2016).

Probabilistic Design Approaches

Based on historical data from a given area and water depth, an ice scouring or structural scour depth statistic assessment can be used to predict extreme ice wash or structural scouring depth at a particular acceptable risk level. However, probabilistic analysis which only considers numerical statistical modeling does not evaluate methods for obtaining data, ice wash, or structural washout depth resolution cutoffs, effects of dynamic environmental activities, pipe and scour direction, scour or flush recurrence rate, pipe length, and other factors. In many instances, gouging is considered the most important loading condition for offshore pipeline design, but it is also considered the most uncertain in terms of predictability (Liberty Energy Project 2017) (Fig. 2).

Finite Element Methods

When a buried pipeline is subjected to a large deformation load, the strain in the pipe wall may be higher than that allowed by conventional design specifications based on linear pipe behavior. In fact, pipeline behavior is nonlinear due to

the potential for large deflections and plastic material properties. Therefore, it is necessary to complete the limit state by including geometric and material nonlinearity, strain-based design. Finite element analysis allows modeling nonlinearities in material, geometry, and tube-soil interactions; it has been used to assess the integrity of pipelines in environmental loading events, such as ice scouring, permafrost melt settlement, frost heave, bulge buckling, and free span of structural scour.

INTECSEA, for example, has developed in-house subroutines for the CEL Advanced Constitutive soil models that more realistically simulate the soil behavior based on critical state soil mechanics theory. These models can address dilation issues and hardening/softening behavior of the soil which results in more accurate estimation of subgouge deformations under ice scour loads. An example of a resultant axial strain due to deformation is shown in Fig. 3.

Industry Regulations, Standards, and Codes

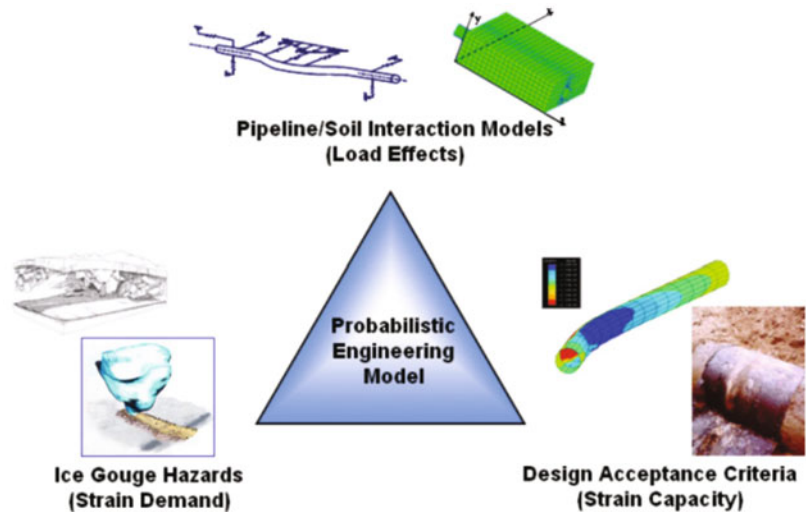
Following regulations, standards, and codes are widely used in offshore Arctic pipeline design.

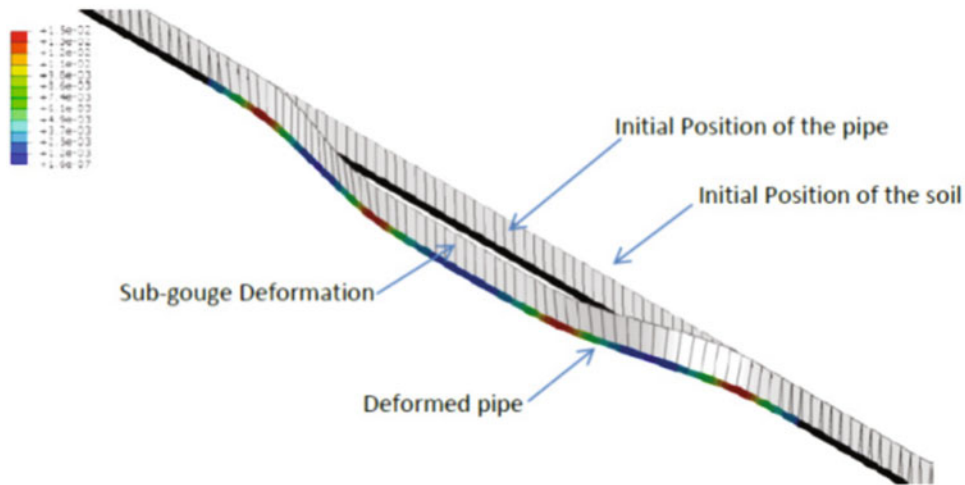
Arctic Specific Regulations

1. ISO 19906 (2010 Edition) Petroleum and Natural Gas Industries-Arctic Offshore Structures

Arctic Pipeline,

Fig. 2 The results of prediction on 1 year





Arctic Pipeline, Fig. 3 Axial strains due to subgouge deformations

2. API RP 2N (2015 Edition) Planning, Designing, and Constructing Structures and Pipelines for Arctic Conditions
3. 30 CFR Parts 250, 254, and 550 Federal Arctic Rule-Oil and Gas and Sulphur Operations on the Outer Continental Shelf-Requirements for Exploratory Drilling on the Arctic Outer Continental Shelf
5. ISO 13623 (2017 Edition) Petroleum and Natural Gas Industries-Pipeline Transportation
6. RMRS 2-020301-005 (2017 Edition) Rules for the Classification and Construction of Subsea Pipelines

US Federal and State Regulations

1. 49 CFR Part 192 (2011 Edition) Transportation of Natural and Other Gas by Pipeline: Minimum Federal Safety Standards;
2. 49 CFR Part 195 (2017 Edition) Transportation of Hazardous Liquids by Pipeline

General Pipeline Design Regulations

1. API RP 1111 (2015 Edition) Design, Construction, Operation and Maintenance of Offshore Hydrocarbon Pipelines (Limit State Design)
2. CSA Z662-15 (2016 Edition) Oil and Gas Pipeline Systems
3. ASME B31.4 (2016 Edition) Pipeline Transportation Systems for Liquids and Slurries/ ASME 31.8 (2016 Edition) Gas Transmission and Distribution Piping Systems
4. DNVGL-ST-F101 (2017 Edition) Submarine Pipeline Systems

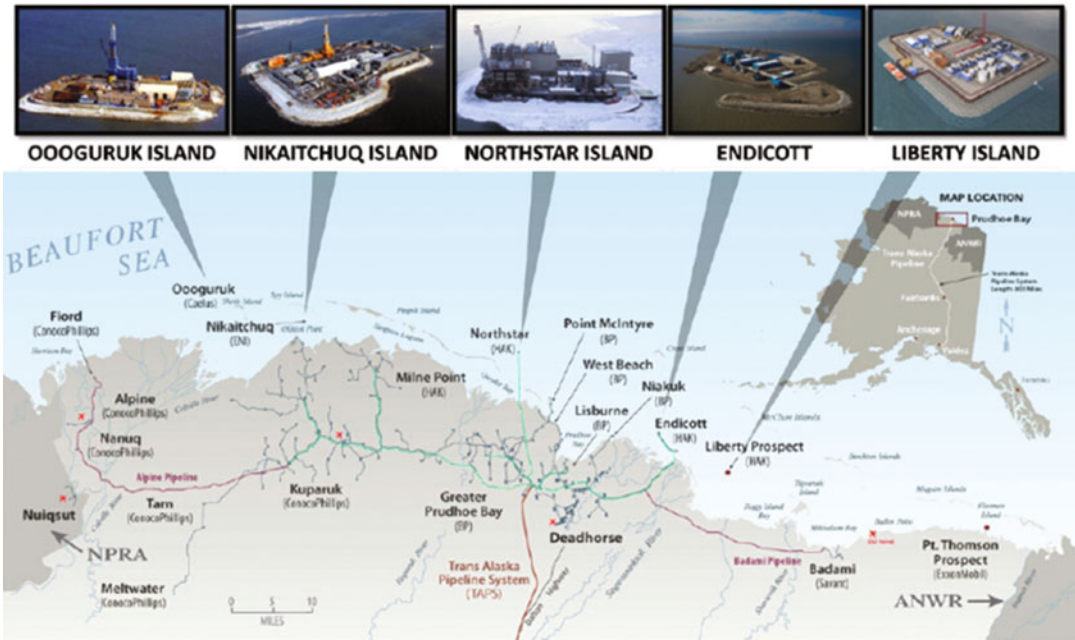
Existing Arctic Pipelines

The existing Arctic pipeline projects include single-walled project: Northstar (Alaska), Varandey Oil Terminal (Russian Pechora Sea), Baydaratskaya Bay Pipeline, Crossing (Russian Kara Sea), Sakhalin 1 (Russian Sea of Okhotsk), Sakhalin 2 (Russian Sea of Okhotsk), Kashagan (Russian North Caspian), and pipe-in-pipe project: Drake Project (Canadian Arctic Archipelago), Oooguruk (Alaska), Nikaitchuq (Alaska), and Liberty Proposal (Alaska). They provide important experience base for designing, installing, and operating future offshore Arctic pipelines. Figure 4 shows the location of these projects.

Figure 5 provides design details for the three operational Alaskan offshore pipelines/flowline bundles.

Northstar

The Northstar pipelines which production commenced in October 2001 were designed for ice



Arctic Pipeline, Fig. 4 Alaskan offshore projects

keel gouging using empirical methods. Limit state strain criteria were developed for design of non-cyclic pipeline displacements settlement, with allowable strain levels established based on the pipe dimensions and material grade.

The pipeline system was made up of a single-walled oil pipeline and a single-walled injection gas pipeline; they were installed in a common trench which the maximum design burial depths reach 7 ft below seabed to top of pipe between, with winter construction using conventional onshore equipment (Paulin et al. 2001) (Fig. 6).

Oooguruk

Natural Resources Alaska Inc. developed the Oooguruk field on the Alaskan North Slope, using a buried three-phase flowline following a pipe-in-pipe design approach from the gravel island to shore, which includes 12 in by 16 in PIP production, 8.625 in OD water injection, 6.625 in OD gas, and 2 in OD AHF line (Lanan et al. 2008). The pipe-in-pipe concept was chosen for the purpose of insulating the production flowline. An added benefit of the vacuum annulus was its

availability as a leak detection system for the offshore production flowline (Fig. 7).

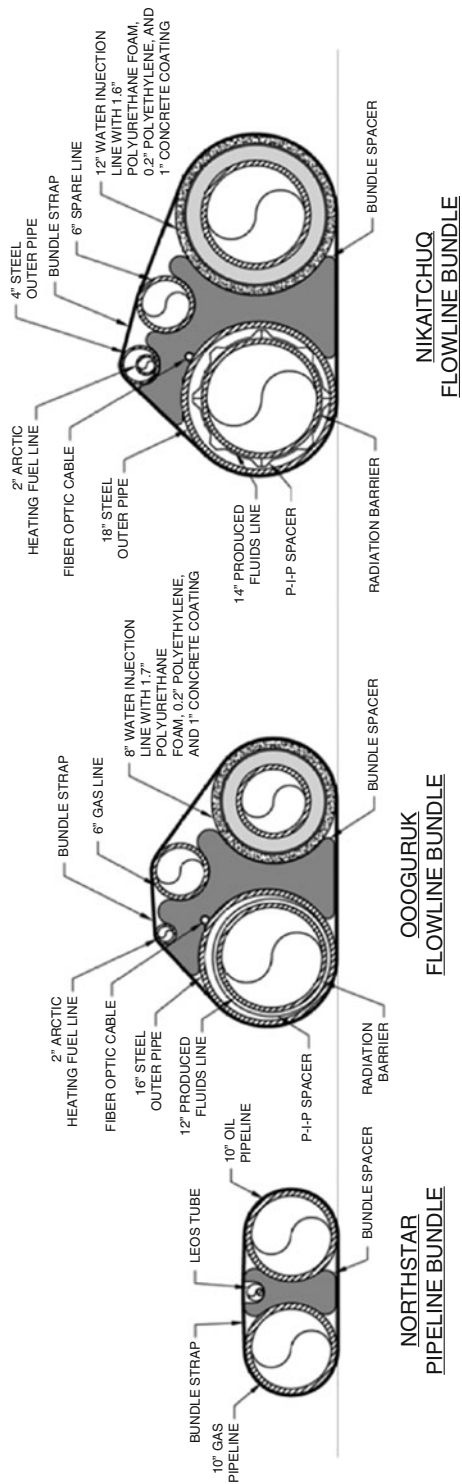
Nikaitchuq

Eni Petroleum Corporation developed the Nikaitchuq field located south of Spy Island in the Beaufort Sea. Produced fluids are transported to shore from the Spy Island drill site via a buried three-phase flowline to a production facility located at Oliktok Point.

The Nikaitchuq flowline bundle is made up of a 14 inch by 18 inch PIP flowline carrying produced fluids, a 12 in OD water injection line, a 6 in OD spare flowline, and a 2 inch by 4 inch PIP Arctic heating fuel line. The PIP concept was mainly chosen for the purpose of insulating the flowline, with the added benefit of allowing annular monitoring for leak detection (Paulin et al. 2001) (Fig. 8).

Arctic Pipeline Design Challenges

Pipeline configurations for Arctic pipeline design, thermal insulation, and trenching requirements

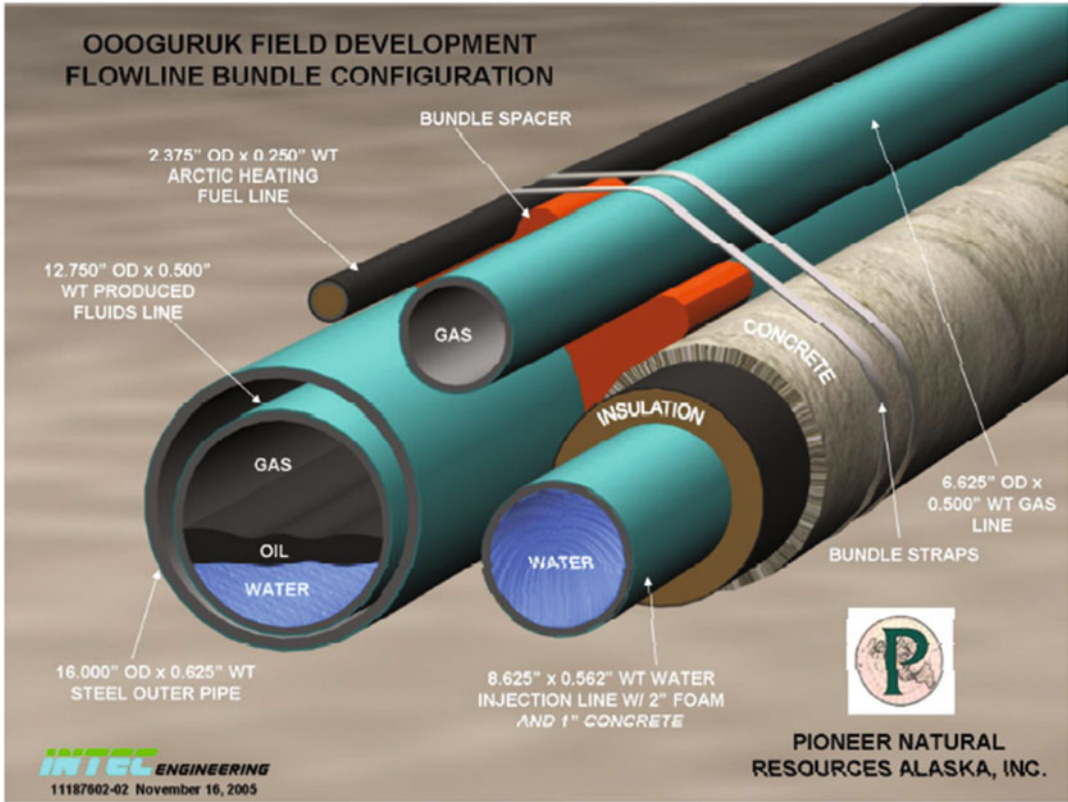


Arctic Pipeline, Fig. 5 Alaskan offshore pipelines/flowline bundles design details (Lanan et al. 2015)

A



Arctic Pipeline, Fig. 6 BP Alaska Northstar Gravel Island (Paulin et al. 2001)



Arctic Pipeline, Fig. 7 Oooguruk bundle details (Lanan et al. 2008)



Arctic Pipeline, Fig. 8 Nikaitchuq Spy Island drill site (Paulin et al. 2001)

are affected by Arctic environmental load conditions. The main features of pipeline design in the Arctic or Arctic environment include pipeline environmental loads and extreme state design under extreme load conditions, resulting from ice scour. Ice scouring is often referred to as the geological term for long and narrow ditches on the sea floor, caused by the grounding of fast ice, fast ice, and pack ice. The differences between arctic pipelines and other conventional pipelines include operating temperature; geotechnical loads, resulting from thaw settlement and frost heave; construction surface disturbance impacts on permafrost terrain; seasonal constraints on construction and maintenance activities; and civil construction techniques in permafrost.

The differences due to the climate conditions and ice coverage require the consideration of certain challenges in the design of arctic pipelines, which include ice scouring, strudel scouring,

permafrost thaw settlement, frost heave, pipe-soil interaction, monitoring and leak detection in the Arctic, and construction and installation techniques. What's more, ice scouring, strudel scouring, and permafrost thaw settlement phenomenon are shown in Figs. 9, 10, and 11, respectively.

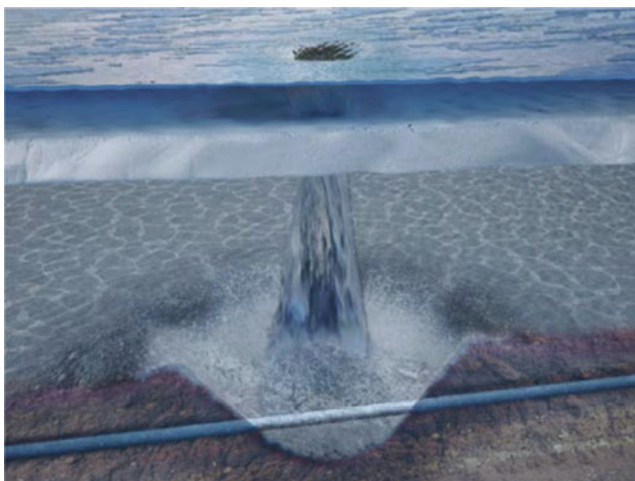
Based on the existing development and research of Arctic pipelines, probabilistic design method and finite element analysis method are used to predict the design principles and layout of Arctic pipelines.

Firstly, the probability design method is used to estimate the reliability range of the design of the Arctic pipeline, and then the finite element analysis method is used to accurately calculate and evaluate the reliability of the design of the Arctic pipeline within the reliability range. Through a lot of calculation and comparative analysis, the design principles and layout of the Arctic pipeline are summarized.

Arctic Pipeline, Fig. 9 Ice keel scouring over a pipeline



Arctic Pipeline, Fig. 10 Strudel scour over a pipeline



Arctic Pipeline, Fig. 11 Thaw settlement and frost heave



The differences due to the climate conditions and ice coverage require the consideration of certain challenges in the design of arctic pipelines, while include:

Ice gouging or icebergs in shallow water
Strudel scour
Frost heave
Permafrost thaw settlement
Upheaval buckling

The main concerning limit states to be investigated during the design of arctic pipelines are two types of limit states. One is the limit state of the rupture of a pipe wall that causes leakage of hydrocarbons, in which the ultimate limit state is thresholds beyond which pressure containment, safety, or the environment is threatened. Another one is the limit state of accident conditions where the pipeline no longer meets one or more design requirements. The serviceability limit states are reached, and the normal operations are restricted, which leads to economic damage to the operator. The goal of the limit state design is to verify the adequacy of the pipeline design against limit states and failure modes relevant to the events. The criteria from DNV OS-F101 and API RP 1111 may be considered. Criteria from these references are based mostly on the risk principles and limit state methodology. The following limit states are to be considered for the design of arctic pipelines: compressive strain limit states for local buckling, tensile strain limit states for strain capacity, and fracture.

Arctic Pipeline Design Procedure

The arctic pipeline design for frost heave and thaw settlement may be carried out through an iterative process in the following steps:

1. Hydraulic analysis for establishing the relationships among the pipe size, material grade, wall thickness, operating temperature, and pressure profiles.
2. Geothermal analysis to predict the potential for frost heave and thaw settlement by using an operating temperature profile. A range of route conditions based on the statistics in terrain analysis are considered as well.
3. Structural modeling to predict pipeline strain and displacement distributions resulting from predicted frost heave and thaw settlement effects over time.
4. Determination of the pipe's capability to sustain the strain and displacement through the analysis of finite element modeling and full-scale test results.
5. Comparison between the strain demands over time with the strain capacity to ensure the pipeline integrity.
6. Testing and checking the heave frost and thaw settlement displacements and freeze bulb growth to assess the environmental impacts.
7. Evaluation of the following factors to the design and maintenance of the arctic pipeline, such as temperature limits, material grade of pipeline, pipeline size, and wall thickness.
8. Repeating Step 1 until Step 7 as required. Arctic pipeline designs also need to include parameters such as ice impact, gouging by icebergs or ice keels, assessment of uncertainty related to ice scour events, and ice-soil-pipe interaction.

Future Arctic pipeline design principles and layout can be predicted on the existing Arctic pipeline development and research.

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Areas Beyond National Jurisdiction (ABNJ)

- [Marine Protected Areas in Areas Beyond National Jurisdiction](#)

Armor Layer

- [Umbilical Cable](#)

Artificial Air Flow

- [Wind Tunnel Test](#)

Assistant Icebreaking Systems

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Atmospheric Diving Suit (ADS)

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Synonyms

[Atmospheric diving system \(ADS\)](#); [Human occupied vehicle \(HOV\)](#); [Manned submersible \(MS\)](#)

Definition

An **Atmospheric Diving Suit (ADS)**, also known as **Atmospheric Diving System**, is a humanoid type deep-sea manned operation system, which can maintain atmospheric pressure when diving. The diver equipped with the suit can directly reach the operation site for observation, navigating by the underwater thruster, or landing and shifting by its own strength. At the same time, with the help of rotary joints, divers can control the gripper of both hands and special tools under underwater operations. The suit is connected to the Tether Management System (TMS) through the neutral cable to obtain energy and communicate. The video image of the operation process is also transmitted to the surface ship for the personnel to make a decision for reference (Jiang et al. 2013c).

Scientific Fundamentals

Historical Development

As a common underwater operation equipment, ADS has been progressed with the development of human technology. Since the simple “diving tub” developed by John Lethbridge in 1715 for the salvage of underwater wreck, to the Hardsuit2000 equipment for submarine rescue, it has experienced 300 years of history (Thornton 2000). In recent years, with the development of offshore oil industry, the ADS in various countries is developing rapidly. Its submergence ability can descend to depth steadily from more than 10 m

to 600 m; from single observation ability to complicated operational ability with professional tools; ADS provide a guarantee in wide application fields in deep-sea oil and gas resources and salvaging of sunken ships.

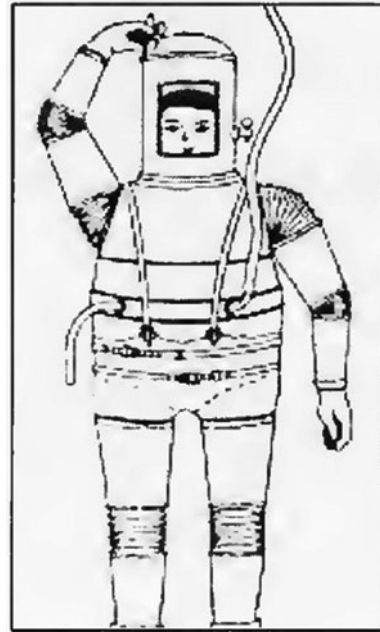
Early Development of ADS

The birth of the first kind of diving suit 1715 to 1882 *Carmagnolle* can be thought of as the early development of ADS. At this stage, the development was mainly driven by salvaging sunken wreck treasure, while technical level at a very low level. Shallow operation depth, no life support systems, or just through the air pipe connecting to the surface, no communication system, no propulsion system, small size and flat glass-viewing window are the typical characters. However, it was during this period, the concept of atmospheric diving for underwater operations, and gradually from the initial bucket structure development to the humanoid type structure, from the development of wooden materials to metal materials, emerged from without limbs joint to the primary of the joints. In this period, *Lethbridge*, *Taylor*, *Phillips*, *Carmagnolle* (Yang 2015) successively appeared.

In 1715, John Lethbridge sought a way to acquire wealth by building a device to recover some of the treasures from the sunken ships (Aylmer 1996). The *Lethbridge*, which he calls “a submersible that is not connected to air,” is widely considered to be the early embryo of ADS. In fact, it is only a wooden bucket that is suspended into the water, through the wall of the bucket with both hands and set up sealing measures to carry out the normal pressure diving.

In 1838, the first diving suit equipped with joints, *Taylor*, was designed. See Fig. 1. Divers in *Taylor* connected with the water supply through a hose, and the joint is similar to the steel structure of accordion type, leather material helping water-tight (Harris 1985). These types of joints have only limited ability in a small range of activities and shallow water depth.

Phillips, designed in 1856, was the first fully enclosed diving suit (Davis 1951) and ball-hole joint was first used. There is a buoyancy chamber on the back, a single-hole viewing window,



Atmospheric Diving Suit (ADS), Fig. 1 Taylor atmospheric diving suit design, 1838 (Harris 1985)

a manhole on the top, and a simple clamping device on the upper extremity, all of which are standard modern fixtures. There is no record of the *Phillips* being built, but many of its features have been applied to successful installations over the next century.

Carmagnolle is the most famous ADS of this stage. In 1882, the Carmagnolle brothers applied for the patent of armored diving equipment in France. The joint of the device is tightly assembled from several concentric hemispheres and is attached to each other by a water-tight cloth with a certain folding so that it can slide (Davis 1951). This is the first truly constructed humanoid assembly, and it is the first time to introduce the Ergonomics consideration in the design.

Mid-Term Development of ADS

After the development of the previous stage, ADS is already applied in wider fields, such as underwater operation of marine oil and gas industry, and aid to the navy submarine rescue. The ADS system also tends to be improved gradually, equipped with independent life support system, communication system, propulsion system, and

operational tools. In this period, there were two important types of joints – oil filled spherical joint and oil filled cylindrical joint, bringing leap-forward development. With materials of steel, magnesium alloy, aluminum alloy, and glass fiber reinforced plastic applied, the operation depth of ADS has been developed from several meters to hundreds of meters. With the enhancement of the underwater working ability of the suits, it is gradually recognized by the industry. Some types have also developed from a single prototype in the early stage to an industrial production status. During this period, many different types emerged, such as *Bowdoin*, *Neufeldt & Kuhnke*, *Galeazzi*, *Campos*, *Peress*, and *JIM* (Yang 2015).

In 1915, American Bowdin obtained a patent for a new type of oil-filled rotary joint. The joint is cylindrical and has a small catheter that equates the internal and external pressure. However, without lasting lubrication, the joints can easily get stuck (Harris 1985). Although this device was not manufactured, it introduced a very important concept of oil filled pressure balance and played an important role in the development of the joint. Then in 1922, Victor Campos applied for a patent for a similar type of oiled joint and made *Campos*. The suit was reported to dive to depth of 600 m. Despite diving 600 m, there is no perceptible movement of the joints. It is worth noting that the *Campos* joint set fail-safe design for the first time. If the joint failure happens, it can automatically seal and not let water into the suit, which has a great significance for the future design of ADS.

In 1917, German company Neufeldt & Kuhnke built two models of ADS based on the patent of ball-hole joint and used the ball to bear and transfer water pressure. In 1924, the German navy tested its second generation, diving to a depth of 161 m, which was very difficult to move the upper limbs and the joints had no fail-safe design (Scott 1931a). In 1931, with the help of this type of equipment, a total of 10 tons of silver COINS and 5 tons of gold COINS were recovered from a sinking ship in Egypt. Through this successful salvage, the suit gained a good reputation (Scott 1931b). In 1930, the suit was licensed to be built and sell. A developed type of *Galeazzi* with many

improvements to the original *Neufeldt & Kuhnke* was manufactured and sold more than 50 sets (Harris 1985).

In 1922, the British inventor Joseph Salim Peress applied for a spherical joint of oil filled and tried to make a type of ADS in 1925 but did not succeed. Then he developed the second type of ADS in 1932, named “*Tritonia*,” which is now often referred to as “*JIMI*.” There were only 8 ball joints, and equipped with simple tools (Barton 1973). *JIM I* was successfully used in *Lusitania*’s wreck salvage, with a maximum depth of 313 m. In 1937, it successfully completed the royal navy sea trial (Loftas 1973). The ball joint of this kind of ADS used hydraulic compensation for the first time, which laid a good foundation for the *JIM* used widely later. Early 1960s, a new type of *JIM* was designed, using the material of cast magnesium, which has high specific strength. See Fig. 2. The front of *JIM* is equipped with ballast, which allows the diver release in an emergency, then the *JIM* can rise to the surface at a speed of about 30 m per min. In 1971, the first set of *JIM* was carried



Atmospheric Diving Suit (ADS), Fig. 2 A JIM suit used by NOAA is recovered from the water. (Wikipedia, photo from NOAA)

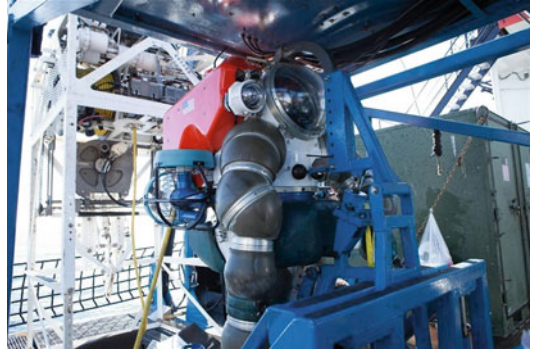
out sea trials by the British naval ship *Reclaim*, and two divers arrived to the depth of 120 m. In 1981, the number of *JIM* has reached 19 sets (Earls et al. 1979), which is widely used in all kinds of underwater operations. *JIM* also became the basis of later developed ADSs.

The *WASP* was completed and put into use in 1978. It follows *JIM*'s design in many aspects but is designed as a GRP column below the waist to replace the lower body. The technology of propeller is adopted for the first time. It is equipped with a small multi-directional propeller, which is controlled by the pedal control board inside the cylinder, thus improving the maneuverability. A new generation of *WASP-III* put into production in 2001, to be able to descend to 760 m depth. The new *WASP* has a stronger vector propulsion system, lateral propulsion capability, and two camera systems.

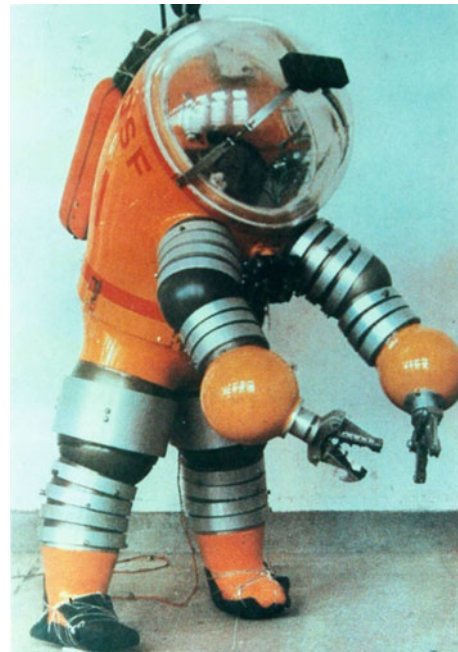
Modern ADS

Phil Nuytten, developed the *NEWSUIT* based on the 1984 patent for rotating joints (Nuytten 1985), which is a truly somatological ADS. It has a wide range of activity allowing it to enter areas previously accessible only to divers. *NEWSUIT* is equipped with 16 patented hydraulic compensated rotary column joints that allow divers to drive the upper and lower limbs by themselves. The innovative joint greatly improves the working depth and flexibility of ADS and is a milestone in the history. The *NEWSUIT* life support system can guarantee 6–8 h of normal operation and 48 h of support in case of emergency.

In order to meet the needs of the army, the US navy cooperated with OceanWorks Company developed a series of *HARDSUIT1000* (305 m), *HARDSUIT1200* (365 m), and *HARDSUIT2000* (610 m) on the basis of *NEWSUIT*. See Fig. 3. The first two types adopt cast aluminum manufacturing, the other aluminum forging manufacturing. Meanwhile, *HARDSUITs* have become the standard equipment for French and Italian naval submarine rescue projects. In addition, the commercial model *HARDSUIT 2500* of *HARDSUIT 2000* will be applied in the industrial field, with a diving depth of 760 m.



Atmospheric Diving Suit (ADS), Fig. 3 U.S. Navy's ADS 2000 At the Ready. (Photo from OceanWorks Company)



Atmospheric Diving Suit (ADS), Fig. 4 QSZ-I atmospheric diving suit. (Photo from CSSRC)

China Ship Scientific Research Center (CSSRC) started research of ADS since 1980, and the first set of QSZ-I was developed in 1986. See Fig. 4. QSZ-I adopts spherical joints to connect limbs, and the maximum diving depth is 300 m. Sea trials with and without diver were carried out in the South China Sea. On the basis of QSZ-I, QSZ-II was improved in the 1990s. Four propellers were increased for depth control, height

control, and cruise control. The computer control system is improved to make divers' underwater activities more convenient and faster. Its biggest characteristic is that underwater movement has dual functions – it can choose to walk on the ground on the knuckles of the legs, and it can also rely on the power of the propeller to propel and cruise in the water. It can be controlled by the diver as well as the deck remote control. It is not only a manned submersible but also an unmanned observation submersible with remote control.

Since 2011, CSSRC began to develop a new type of ADS with the depth of 500 m, purposed to support marine oil and gas production safety maintenance and guarantee. The ADS has completed sea trials in 2015. The TMS, Launch and Recovery System (LARS), independent life support system, high-definition cameras, imaging sonar, two LED lights, emergency ballast, neutral line cutting, stroboscopic lamp, emergency pinger lamp, etc. equipped with the ADS forming a complete set.

After 2000, Phil Nuytten designed the latest *EXOSUIT*, which will work 305 m deep and reach 610 m in the future. The two salient features of the *EXOSUIT* are its lightweight and cable-free nature, allowing divers to swim within the *EXOSUIT* by swinging their lower limbs to drive their flippers, indicating that their joints are already highly flexible. It has 22 highly flexible rotary joints, and the foot swimming fins, the air (excluding divers) weighs only 72 kg. *EXOSUIT* has the ability of 48 h life support and is equipped with the latest underwater communications. It can be decomposed and loaded into a cylindrical vessel with a diameter of 0.4 m and a height of 0.6 m, which is convenient for storage in the submarine cabin. Traditional underwater operations, from compressed air to communication, rely on umbilical cables, so the effect of uncabled underwater operations needs to be further verified.

Principles of Modern Atmospheric Diving Suit

The arms and legs of modern ADS are connected by joints. The limbs are human-powered. Propulsion devices are equipped to maintain mobility in the water.

Joint is one of the key components of ADS. Hydraulic support column joint based on the patent of Phil Nuytten in 1984 is widely used now (Nuytten 1990). This type of joint has light weight, small rotation torque, large rotation angle, and reliable sealing. At present, the joint technical difficulties include: (1) underwater sealing technology – the research of sealing form and interior design parameters with high reliable seal and low rotating friction under deep water pressure; (2) rotary positioning technology – the research of rotary positioning between cylinder and piston with reliable radial and axial accuracy but not additional joint rotation torque and no appearance of stuck under high pressure; (3) hydraulic support technology – reliable design of hydraulic support to resist water pressure and reduce the rotational resistance under deep water pressure, also make sure that the joints can automatically compensate the amount of oil loss in the process of using; (4) pressure-resistant structure design – design of joints with lightest weight, most reasonable deformation control and uniform stress distribution meeting the requirements of working depth, and deformation under high pressure will not affect the sealing mechanism at the same time; (5) safety self-locking technique – when complete oil loss or sealing failure appears, the joints can lock themselves so as not to allow seawater to enter the cavity (Jiang et.al 2013a). It is a worldwide technical problem to develop high depth ADS that can meet the urgent requirements of high depth atmospheric diving suit.

The torso of ADS is somatological and the specific size is usually referred to ergonomic standards. Wrought aluminum milling forming is the popular process using at the time to ensure the density and intensity of the material. In addition, machining precision can be controlled strictly, make the thickness of the shell more reasonable, and the overall ADS weight lighter. There are four large openings in the upper body of the torso, including window opening, left and right upper limb opening, and waist opening, respectively. See Fig. 5. The same three openings in the lower body, respectively in the waist and the left and right lower limbs. It is necessary to carry out relevant theoretical analysis and experimental



Atmospheric Diving Suit (ADS), Fig. 5 The upper body of ADS torso with hemispherical viewing window. (Photo from CSSRC)

research on such non-regular, multi-channel, and large-opening structures. At present, the waist opening is usually adopted, which make divers in and out of ADS convenient. A bonus is the torso height can be adjusted by adding or removing torso sections to adapt to the needs of different height of the divers (Jiang et.al 2013b).

Hemispherical viewing window technology is also important. The *JIM* was originally designed with four viewing windows on the top. Later it was improved to a fully transparent Plexiglas helmet, providing a wider view for the operator. However, under the action of high pressure seawater, organic glass will undergo continuous creep deformation. This puts forward new requirements for design strength and safety. In addition, the manufacturing process of hemispherical viewing window is rather complicated. It is important to pay attention to the installation form of a hemispherical viewing window with torso connection. The study found that big bearing stress will occur during the progress of hemispherical glass shell inlaid into metal shell (Taylor and Lawson 2009).

Propellers are equipped on various types of ADS after the development of *JIM*, which are controlled by the pedal control board inside the ADS. Thus, the movement capability is

greatly improved. Two 2.25 hp constant speed adjustable pitch propellers are used on most existing ADS to help divers “fly” underwater or stay in the same site when they flow slightly down (Shen et al. 2015). The developing electronic ring propeller (ERP) is expected to reduce wear and maintenance and is the development direction of future propellers.

Neutral cable connects the ADS and TMS, transmitting the underwater power source, image information, status information and communication link between the diver and the deck commander (Liu 2009). When ADS is close to zero underwater buoyancy or a suspended state, the tensile force inside the neutral cable is very small, leading to several circle of cable looseness appears at the same time under fast cable releasing rapid. Therefore, the synchronous transmission mechanism consists of a set of counter-rotating active wheels and passive wheels will be necessary for cable smooth and orderly arrangement. The wheels provide the same direction traction to help send neutral cable out of TMS, while appropriate inverse hysteresis force to keep neutral cable tightly aligned back on the winch (Zhao et al. 2014).

Advantages

ADS is able to send divers directly to the underwater site, carrying out more effective operation. Smaller and lighter weight make it easier get into some limited space, engaging complex diving that only wet divers can do before. ADS eliminates or alleviates the physiological risks brought by ordinary wet diving. Divers do not need to decompress or pressurize for a long time, thus extending the underwater residence time and improving the operating efficiency. ADS can be equipped with various operation tools and adjust operation depth above rated work depth by propeller. When hovering over the water, ADS also has the ability of resistance to the current at the same time. With these advantages, ADS has a good military and civilian generality, increasingly wide application in all kinds of underwater operation, and it is gradually forming industry market. The advantages of low economic cost, high efficiency, and sufficient security also gradually revealed.

Key Applications

Application range of Atmospheric Diving Suit

ADS has become an effective way of underwater operation. In ocean engineering, ADS is currently used in deep-sea oil and gas resources development, realizing many functions such as offshore platform jacket installation, pipeline and cable laying, oil facilities and structure inspection support, underwater platform cleaning, rope hanging, life support POD delivering, and so on. It can also be used in offshore industries, such as underwater industrial operations. In the aspect of submarine rescue, ADS has successfully participated in the Deep Submergence Rescue Program of various countries and carried out the task of underwater rapid assessment and rescue. ADS can be widely used in the exploitation, operation, and application of deep-sea resources.

Important Events of Atmospheric Diving Suit

In 1975, the *JIM* won the privilege of petroleum engineering maintenance, carrying out oil well operations for 5 h 59 min at a depth of 300 m under water. *JIM* created the longest diving record in 150 m+depth work at the time (Curley and Bachrach 1982).

The *WASP* completed the repair of an oil pipeline 20 cm in diameter at a depth of 650 m under water in conjunction with a Remote Operated Vehicle (ROV), setting a record for the depth of the submarine pipeline maintenance work at that time (McCabe et al. 2000).

At the beginning of 1988, the *QSZ-I* conducted a 2500 square meters census of Yingxiu bay reservoir in Aba Tibetan autonomous prefecture, Sichuan province, inspecting concrete erosion and damage. In less than 10 days, the position of more than 2000 m² of underwater scouring, grinding, and silting was basically found out, which provided a technical basis for the reinforcement of the bottom plate of the dam gate of Yingxiu bay reservoir (Lin et al. 1990).

In the summer of 1988, the *QSZ-I* conducted dam crack detection for Beijing Zhuwo reservoir, scanning and recording five dam sections. According to the accurately specified position, underwater repair was successfully carried out.

The *NEWTsuit* first participated in the rescue of the wrecked ship *SS Edmund Fitzgerald* in 1995 (Jiang et al. 2013a).

In September 1998, the OceanWorks Company (OWC) completed a 2000-foot-deep manned submersible test for ADS2000 system in NEDU, Panama City. After several times of function demonstration experiments, ADS2000 system was officially delivered to the US navy (Gibson and English 2000).

Cross-References

- [Human Occupied Vehicle \(HOV\)](#)
- [Submarine](#)

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Atmospheric Diving System (ADS)

- [Atmospheric Diving Suit \(ADS\)](#)

ATOT, Arctic Tandem Offloading Terminal

- [Moored Ship in Ice](#)

Automatic Control

- [Intelligent Control Algorithms in Underwater Vehicles](#)

Automatic Control System

- [AUV/ROV/HOV Control Systems](#)

Automatic Identification System

- [Impact of Maritime Transport](#)
- [Ship Propulsion System](#)

Autonomous and Remotely Operated Vehicle (ARV)

- [Hybrid Remotely Operated Vehicle \(HROV\)/Autonomous and Remotely Operated Vehicle \(ARV\)](#)
- [Submersible](#)

Autonomous Lagrangian Circulation Explorer (ALACE)

- [Profiling Float](#)

Autonomous Underwater Glider (AUG)

- [Glider](#)

Autonomous Underwater Vehicle (AUV)

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Synonyms

Doppler velocity log (DVL); Long baseline (LBL); Ultrashort baseline (USBL); Unmanned underwater vehicle (UUV)

Definition

Autonomous underwater vehicles (AUVs) are unmanned underwater vehicles (UUVs) without tethers that are powered by onboard energy sources. They are intended to accomplish predefined tasks with little or no human supervision. They can be fully or largely autonomous, communicating intermittently with operators using acoustic or radio links.

Gliders form a distinct subclass of AUVs. They move through the water column, translating the vertical forces of positive or negative buoyancy into a horizontal motion using wings. Whereas propeller-driven AUVs have endurance measured in hours or days (tens or hundreds of miles), glider endurance is measured in weeks or months (thousands of miles).

Brief History of AUV Development

AUV development began in the late 1950s (Blidberg 2001; Von Alt 2003). A few AUVs were built mostly to focus on data gathering. SPURV I (Widditsch 1973), the first “true” AUV developed by the Applied Physics Laboratory (APL) of the University of Washington in the early 1960s, displaced 480 kg and could operate at 2.2 m/s for 5.5 hours at depths to 3000 m. It was acoustically controlled from the surface and could autonomously run at a constant depth, seesaw between two depths, or climb and dive at up to 50°.

During the 1970s, a number of test beds were developed. The University of Washington (APL) developed the Unmanned Arctic Research Submersible (UARS) (Francois and Nodland 1972) and SPURV II (Nodland et al. 1981) (modified from SPURV I) to gather data from the Arctic regions. IFREMER’s Epulard was designed in 1976, assembled by 1978, and fully operational by 1980. Epulard was capable of diving to 6,000 m water depth and was operated remotely by an acoustic link. Also at this time, the development of the SKAT vehicles started under the AUV program of Institute of Marine Technology Problems, Russian Academy of Sciences.

In the 1980s a number of technological advances greatly affected AUV development. Small, low-power computers and memory offered the potential of implementing complex navigation and control algorithms on autonomous platforms. Advances in software systems and engineering made it possible to develop complex software systems. The Advanced Unmanned Search System (AUSS), developed by the Naval Oceans System Center (now SPAWAR) of the United States, was launched in 1983. AUSS displaced 907 kg, carried 20 kw hours of energy in silver-zinc batteries, and was rated to 6000 m. It had an acoustic communication system that transmitted video images through the water.

During 1990–2000, AUVs grew from proof-of-concept test beds into operational systems able to accomplish specified tasks. A number of organizations around the world undertook development efforts focused on various operational tasks. Examples include Odyssey of MIT Sea Grant AUV Lab, Autonomous Benthic Explorer (ABE) and REMUS of Woods Hole Oceanographic Institution (WHOI) (Yoerger et al. 1991; Allen et al. 1997), Autosub of Southampton Oceanographic Centre (Brierley et al. 2002), The-seus of International Submarine Engineering (Butler and den Hertog 1993), and Hugin of Kongsberg Simrad of Norway (Storkersen et al. 1998).

Since 2000, AUV technology extended from the academic and research environment into the commercial mainstream of the ocean industry. Examples include Nereus (Bowen et al. 2008; Bowen et al. 2009) and Sentry (Yoerger et al. 2006) of WHOI, Bluefin of (Panish et al. 2011), r2D4 of the University of Tokyo (Ura et al. 2004), Tethys long-range AUV of Monterey Bay Aquarium Research Institute (MBARI) (Hobson et al. 2012).

AUV Subsystems and Technologies

Pressure and Hydrodynamic Hulls

By providing enclosed chambers for the onboard equipment that needs to work in dry, atmospheric environment, pressure hulls withstand the

ambient pressure. As the depth rating increases, the proportion of vehicle weight allocated to pressure hull increases, thus reducing the proportion of weight allocated to the other systems including energy source.

Depending on depth ratings, pressure hulls can have either cylindrical or spherical shapes. The most common materials for pressure hulls are aluminum or titanium. However, the use of ceramics as a pressure hull material appears promising and has been verified in WHOI Nereus (Bowen et al. 2008).

Hydrodynamic hulls are intended to reduce flow drag as AUVs move through water to maximize speed and endurance. Streamlined hulls are preferable as they generate less turbulent flows around vehicle body. However, the hydrodynamic performance of the laminar flow body degrades significantly when the hull shape is changed due to mission configuration. Therefore, torpedo body that has a nose cap followed by a parallel midsection and a tapered tail section is applied as an alternative and thus leads to an additional drag of approximately 30%.

Ballast

Although traditional AUVs are designed to have slightly positive buoyancy and passive stability (the ability to return to the upright position), it is still desirable to adopt ballast to aid ascent and descent. Emergency drop weights, also part of ballast systems, are released in the event of hardware failure so that the vehicle returns to the surface.

Energy and Power Management

Most power systems for AUVs rely on batteries that supply limited energy. Lead-acid batteries became common AUV power sources in the 1980s. However, such batteries provide relatively little energy given their weight. Silver-zinc batteries replaced lead-acid batteries but proved expensive and vulnerable to failure after relatively few cycles. Alternatively, vehicles have used lithium primary batteries, which are not rechargeable. With advance in the development of more efficient and safer batteries for electric cars, the rechargeable lithium-ion battery has been widely used as energy source for current AUVs.

A number of battery cells are connected in series to form battery banks which are then connected in parallel to provide sufficient current draw by onboard electrical devices. To ensure uniform battery drain and to handle ground faults, power management system is needed. Power management system monitors the main bus voltage and the current drawn from each battery bank. Based on this data, the energy used is calculated and the energy remaining can be estimated.

Propulsion and Maneuvering Systems

Propulsion for torpedolike AUVs is generally provided by electric motors that drive propellers. As the amount of energy AUVs that can carry onboard is limited, the efficiency of the propulsion system is critical to minimize the amount of electrical power consumed. Due to their advantages over brushed motors in efficiency, reliability, and power density, brushless direct-current (DC) motors are applied increasingly for AUV propulsion systems. For a propulsion system to achieve high efficiency, it may be necessary to match the optimal rotational speeds of motors and propellers by reduction gears. Transmission loss needs to be considered as well.

Most AUVs maneuver by deflecting control surfaces or vectored propulsors. In missions that require hovering, vehicles can maneuver to face against currents. Alternatively, multiple thrusters can be applied to allow vehicles to hover with demanded heading. Gliders can maneuver by shifting weights along longitudinal axes for pitching and rotating weight (such as batteries) around longitudinal axes for rolling and yawing. To achieve desirable sawtooth pattern, changing buoyancy and shifting internal weights have to be coordinated.

Navigation and Positioning Systems

Navigation and positioning systems, on their own, reckon AUVs' positions from various types of motion sensors by fusion algorithms, e.g., Kalman filter or particle filters (Stutters et al. 2008). Positional information is the indispensable part of the survey data collected by AUVs. By working together with control systems, navigation and positioning systems enable AUVs to move along

pre-defined tracks which are specified by path/trajectory (motion) planners.

For low-cost AUVs, primary onboard motion sensors include global positioning system (GPS) receiver (valid only when AUVs are on surface), strap-down microelectromechanical systems (MEMS), inertial navigation system (INS), depth sensor, and magnetic compass. INS integrates acceleration and angular rate from inertial measurement unit (IMU) to estimate motion states, which include translational and rotational speeds, position, attitude, and heading. Magnetic compass determines vehicle's heading relative to the north direction. Depth sensor measures vehicle's depth in terms of the ambient hydrostatic pressure.

To reduce the growing rate of error in position estimate, MEMS-based INS is replaced by fiber optic gyro (FOG)- or ring laser gyro (RLG)-based INS. Moreover, Doppler velocity log (DVL) that transmits four separate beams downward from the vehicle to measure the speed relative to the bottom is required (Kinsey and Whitcomb 2004).

Even equipped with FOG-/RLG-based INS and DVL, the positional error involved in navigation and positioning system grows with time. For some missions, external position fixes are usually needed to assist navigation and positioning system. Underwater position fixes are often provided by acoustic positioning systems, e.g., ultrashort baseline (USBL) system and long baseline (LBL) systems. With only one transducer installed on the ship hull, USBL is portable and consequently preferred when a variety of surface ships may be used. However, careful shipboard calibration is required to provide adequate positional accuracy. Traditional LBL requires both the deployment of equipment on the seafloor and careful calibration. Inverted LBL, where GPS receivers are used on the ocean surface, is promising as shipboard calibration is not required.

Control Systems

Generally, control systems can be divided into motion and equipment control systems. While the latter mainly deals with control of onboard devices, e.g., switching of power for equipment and payload sensors, the former enables the

vehicle to move with demanded pattern that matches the pre-defined tasks.

From the viewpoint of implementation, equipment control is not difficult as simple logic-based algorithms are sufficient. However, motion control is a challenge as both complex vehicle dynamics and various disturbances have to be dealt with (Yuh 2000).

As roll and pitch modes are often left uncontrolled due to their passive stability, primary motion control system enables AUVs to move with demanded forward speed, heading, and depth. Advanced motion control system requires AUVs to move along pre-defined tracks with specified forward speed, heading, and depth.

With motion controller as its "brain," motion control system generates maneuvering commands for propulsion and maneuvering system, which in turn drive thrusters and/or control surfaces, in terms of the deviation between the set and feedback values of interested motion states, e.g., forward speed, heading, and depth.

It is the motion controller that needs to deal with the complex vehicle dynamics and various disturbances. Dynamics of underwater vehicles, including hydrodynamic parameter uncertainties, are highly nonlinear, coupled, and time varying. Disturbances come from ocean currents, varying buoyancy due to the density variation of seawater at different depths and near-surface wave-suction forces. Typical control schemes include sliding mode control (Yoerger and Slotine 1985; Healey and Lienard 1993), nonlinear control (Nakamura and Savant 1992), and adaptive control (Zhao et al. 2004).

Communications

Bidirectional acoustic, radio-frequency, and satellite-based communications systems enable human operators to supervise and redirect AUV missions from a ship or from land. However, the duty cycle of the telemetry system can have a significant impact on AUV energy requirements or endurance.

Compared to radio-frequency communications, high-speed communications in the underwater acoustic channel has been more challenging because of limited bandwidth, extended

multipath, severe fading, rapid time variation and Doppler shifts (Chitre et al. 2008). While robust low-complexity incoherent techniques for low data rate acoustic communications have been employed widely, coherent techniques are promising.

Motion Planning

To a great extent, the autonomy of AUVs is equivalent to their ability of motion planning, that is, generating collision-free, dynamically feasible, and low-cost trajectories which enable AUVs to accomplish their missions.

Path planner is the essential part of a motion planner. Traditional path planner often imposes a discretization over the operational area and searches an optimal path over the discretization. The discretization is obtained by imposing a regular grid represent the collision-free areas. Since vehicle dynamics is not taken into account, the planned paths are not necessarily dynamically feasible (Button et al. 2009).

Therefore, it is desirable for AUVs to plan their trajectories online in response to vehicle motion states while deliberating on how best to follow the paths planned in advance. In addition, motion planners will have to reason about the high-level aspects that are related to the dynamic interaction between AUVs and their surroundings. There is a growing need to reason about both the high-level and low-level aspects simultaneously (McMahon and Plaku 2016).

Payloads

Payloads on-board AUVs mainly include acoustic, magnetic, electromagnetic, optical, and CTD (conductivity, temperature, and depth) sensors. Mounted on AUVs that move near the bottom with constant altitudes, the data connected by these payload sensors have higher and more consistent spatial resolution than their shipboard compartments.

Acoustic Sensor

Typical acoustic sensors include multi-beam sonar, side-scan sonar, and sub-bottom profiler.

Multi-beam sonars consist of a Mills Cross transducer array that produces multiple across-

track beams as the intersection of one transmit beam (fan shaped across the survey track) with numerous receive beams (each fan shaped along the track). This allows the sounder to provide continuous swath bathymetry over a 120–150° sector.

Side-scan sonars are one of the most accurate sensors for imaging large areas of the ocean floor by transmitting beams of acoustic energy from the sides of the vehicle (across the track). Unlike a ship, the sonar transducers of side-scan sonar are mounted on the vehicle hull rather than towed.

Sub-bottom profilers are a shallow geophysical sonar designed to provide higher-resolution profiles below the seabed but with less penetration than lower-frequency seismic equipment such as air guns and sparkers. Sub-bottom profiler images are made up of backscatter from the seafloor strata. The first boundary is at the seabed, between the water and the seafloor itself. As layers of clay, sand, and various other sediments succeed each other, they reflect sound at their interfaces. It is this reflected energy that the system uses to build the image.

A recent trend in active sonars is synthetic aperture sonar (SAS). Whereas simple active sonars emit individual pings and process them individually, SAS systems on moving AUVs assemble images from multiple pings and thus give sonar designers the means to increase sonar aperture and improve resolution by an order of magnitude.

Magnetic Sensor

Magnetic sensors include magnetic compass and magnetometer. While the former is used for AUV navigation (heading), the latter is applicable to inspect pipelines/cables and to locate ferrous objects on the seabed.

Optical Sensor

Imaging optical sensors include still and video cameras augmented by lighting systems. The performance of imaging optical sensors for missions such as inspection can also be enhanced using simple dual-laser scaling devices that emit parallel beams of light separated by a known distance.

Pairs of dots projected by these devices provide a scale helpful in visually identifying objects.

Non-imaging optical sensors are used for basic tasks, such as measuring water turbidity.

CTD Sensor

CTD sensors are used to collect oceanographic data and predict and improve the performance of onboard sonars. The conductivity of seawater is closely linked to its salinity. Salinity, temperature, and depth are, in turn, the predominant factors used to predict undersea sound velocity. Therefore, CTD sensors can be thought of as calibrating tools for the active sonars.

Key Applications

With advances in AUV technologies, e.g., high-density energy source, underwater navigation and communication, and sensing, AUVs have been increasingly applied in ocean exploration and exploitation. Most AUVs have been developed to accomplish science, commercial, and military missions.

Science Mission

Typical science missions include:

- Locating, mapping, and photographing hydrothermal vents by WHOI ABE (German et al. 2008).
- High-quality acoustic imaging of seabed by the University of Tokyo r2D4 (Ura et al. 2004).
- Conducting swath sonar measurements under sea ice by the University of Southampton Autosub II (Wadhams et al. 2006).
- Tracking a coastal upwelling front by MBARI's Tethys long-range AUV (Zhang et al. 2012).
- Capturing water samples at chlorophyll fluorescence peaks in a thin phytoplankton layer by MBARI dorado (Zhang et al. 2010).
- Assessing the Deepwater horizon oil spill with WHOI's sentry (Kinsey et al. 2011)
- Oceanographic research with commercial gliders Seaglider (Eriksen et al. 2001), Slocum (Webb et al. 2001), and spray (Sherman et al. 2001).

Commercial Mission

Typical commercial missions include:

- Survey of gas transport pipeline route with Kongsberg Hugin 1000 (Marthiniussen et al. 2004)
- Offshore surveying with Kongsberg Hugin 3000 (Marthiniussen et al. 2004)
- Undersea-cable deployment and inspection with ISE Theseus (Butler and den Hertog 1993)
- Searching wreckages of missing airplanes, e.g., REMUS 6000 for Air France Flight 447 and Bluefin-21 for Malaysia Airlines Flight 370

Military Mission

Typical military mission includes autonomous mine reconnaissance survey with Kongsberg Hugin (Marthiniussen et al. 2004).

Cross-References

- ▶ [AUV/ROV/HOV Propulsion System](#)
- ▶ [AUV/ROV/HOV Resistance](#)
- ▶ [AUV/ROV/HOV Stability](#)
- ▶ [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)
- ▶ [Glider](#)
- ▶ [Remotely Operated Vehicle \(ROV\) in Subsea Engineering](#)
- ▶ [Structural Design](#)
- ▶ [Submersible](#)
- ▶ [Underwater Information Sensing Technology](#)

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Autonomous Underwater Vehicles (AUVs)

- [Environmental Perception for Underwater Vehicles](#)
- [Hydrodynamics for Subsea Systems](#)

Autopilot Control System

- [Semisubmersible Vehicle Autopilot Control System](#)

AUV/ROV/HOV Control Systems

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Synonyms

[Automatic control system](#); [Control approach](#); [Control device](#); [Control module](#); [Control unit](#); [Controller](#); [Motion control system](#)

Definition

Control system is a kind of active regulation system designed to improve or optimize specific performance or indicators for economic, social, or physical systems. AUV/ROV/HOV belongs to a multi-degree-of-freedom underwater platform system with high redundancy, coupling, and

model nonlinearity. It works underactuated in complex underwater environment, and the body must maintain autonomous stability in the process of operation. Therefore, the design of appropriate control system to ensure the stability of the underwater platform during the operational process is of great significance to the successful and accurate completion of the underwater mission of AUV/ROV/HOV.

Scientific Fundamentals

Difficulties in the Design of Control System

Underwater vehicles have a large variety of types, and they are widely involved in undersea surveillance, inspection, and survey missions. Typically, gliders are common with a torpedo shape for long range missions, and human-occupied vehicles (HOVs) as well as remote operating vehicles (ROVs) are generally of a cubic shape used for hovering tasks. For some specific applications, undersea pipeline inspection, offshore infrastructure surveillance, and large vessel maintenance, AUV is preferred.

Regardless of modeling issues, the value of a model-based control approach depends on how robust and efficient the control scheme can adopt the hydrodynamic model. Potential trends of current methods focus on faster controllers to assist the pilot or the autopilot with better accuracy. Optimal controllers can reduce propelling actions to save the battery power as well as to increase the propeller lifespan. Moreover, numerous uncertainties should be considered, including parameter variations, nonlinear hydrodynamic damping effects, sensor transmit delays, and ocean current disturbances (Clement et al. 2016).

Each underwater vehicle is designed to perform specific functions. Even for the same type of underwater vehicle, their shapes may be very different. Therefore, the mathematical models of underwater vehicles will be much different. Each underwater vehicle pursues unique performance indicators. This makes it difficult to solve all control system design problems with a single theory or model system.

A decade of operational experience by numerous research groups has demonstrated ROVs to be ideal platforms for “rapid prototyping” and “rapid field deployment” of UUV subsystems. Recent examples include sonar imaging and survey, optical imaging and survey, navigation, control, oceanographic sampling, and subsea manipulation. Numerous research results first pioneered with ROVs, and towed vehicles are now commonly employed for autonomous underwater vehicles (AUV, HOV) and seafloor observatories (Smallwood et al. 1999).

Structure of Underwater Vehicle System

In the following chapters, the typical structure of the underwater vehicle system will be given. These components are the integral part for underwater vehicle to complete basal motion.

The submersible control system has two components, surface ship control system and underwater vehicle control system. The surface ship control system includes system GMT clock, ship dynamic positioning system, ship LBL navigation system, GPS satellite navigation system, real-time data logging system, science payload interface, instruments, video distribution, and recording subsystem. The underwater vehicle control system includes vehicle control computer system, surface telemetry, pilot’s user interface (such as joystick, keyboard instruction), engineer user interface, manipulator user interface, work package interface, and pilot video display. Vehicle control computer system includes control process, positioning navigation process, LBL-Doppler navigation process, and power management and hotel process, shown in Fig. 1.

Surface Systems

The surface control system is the central “brain” of the ROV control system. It is comprised of a “core” system of safety-critical systems that are essential for safety and control of the ROV and an “extended” system providing non-safety-critical systems such as data logging and video recording. The safety-critical core system is comprised of the vehicle control computer and user interfaces for the vehicle pilot and engineer. The pilot station provides real-time video and navigation

instruments, has joystick controls for closed-loop control of the vehicle reference trajectories and navigation waypoints, and has controls for the vehicle’s manipulator arms. The engineer station has a more comprehensive set of real-time vehicle status indicators and enables the engineer to control all vehicle subsystems. The modules are depicted in Fig. 1.

Concentrating the vehicle intelligence in the ship-board control computer dramatically simplifies and accelerates development – reprogramming an on-ship computer is significantly easier than reprogramming an embedded vehicle control computer.

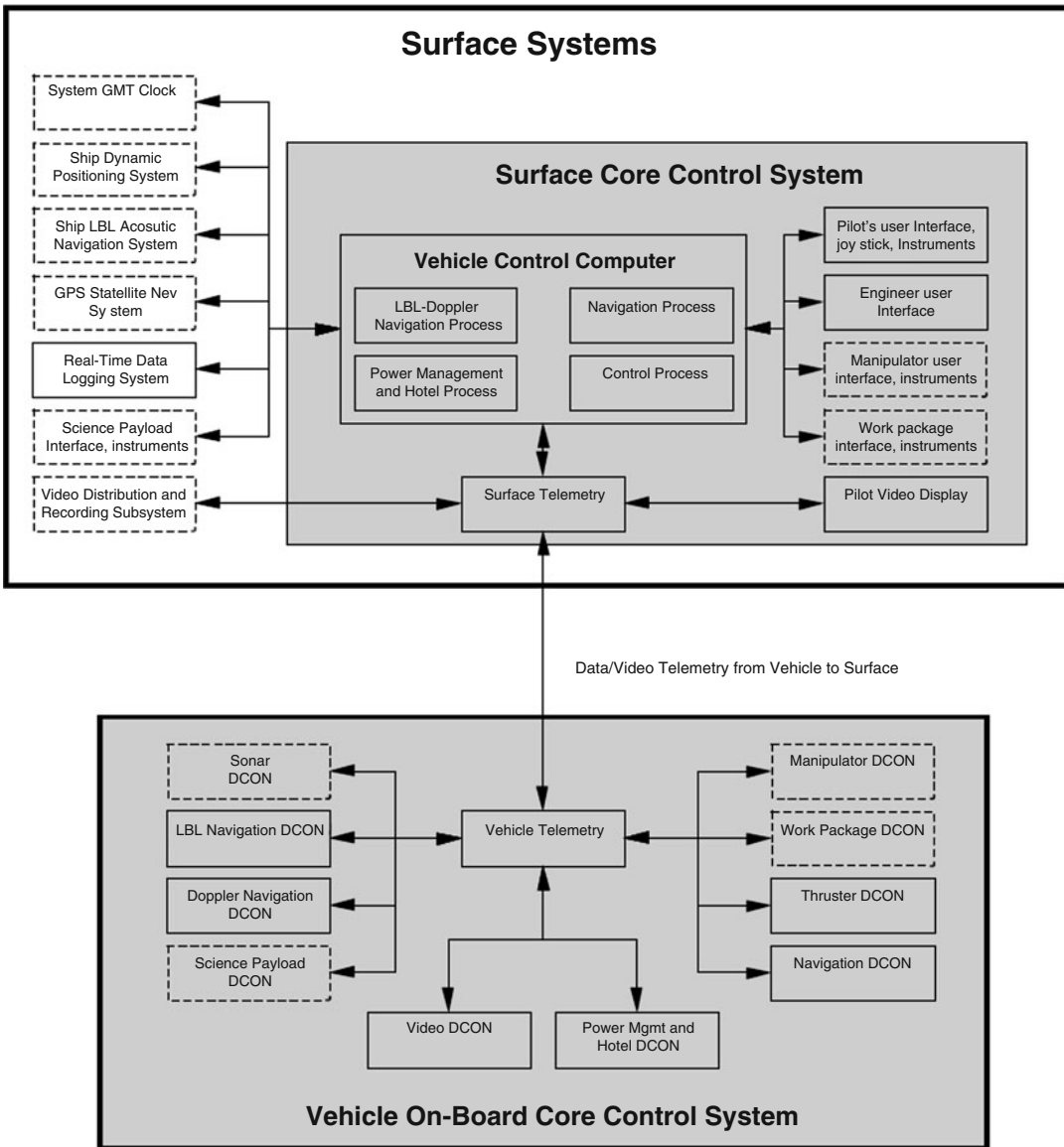
Vehicle on-Board Core Control System

The on-board vehicle control system controls and monitors all vehicle sensors and actuators in response to real-time commands from a surface control system. We will adopt a “data concentrator” (DCON) type of architecture. Each data concentrator (DCON) module will independently control the power and data telemetry for an entire vehicle subsystem or scientific payload. The DCONs will receive commands from the surface control computer, monitor the status, and report data from on-board vehicle subsystems and instruments. Each data concentrator operates asynchronously and will communicate to the surface control computer via a high bandwidth fiber-optic telemetry link.

The data concentrator vehicle control system architecture employs relatively simple on-vehicle computer systems. We anticipate employing commercial off-the-shelf (COTS) embedded computers for the data concentrators. The simplicity of the data concentrator design will render them both highly reliable and easily reconfigurable (Smallwood et al. 1999).

Realization of Vehicle Control System

Generally, the underwater vehicle is equipped with scanning sonars and a color CCD camera, together with depth sensors and a fiber-optic gyroscope. This device is intended primarily as a research platform upon which to test novel



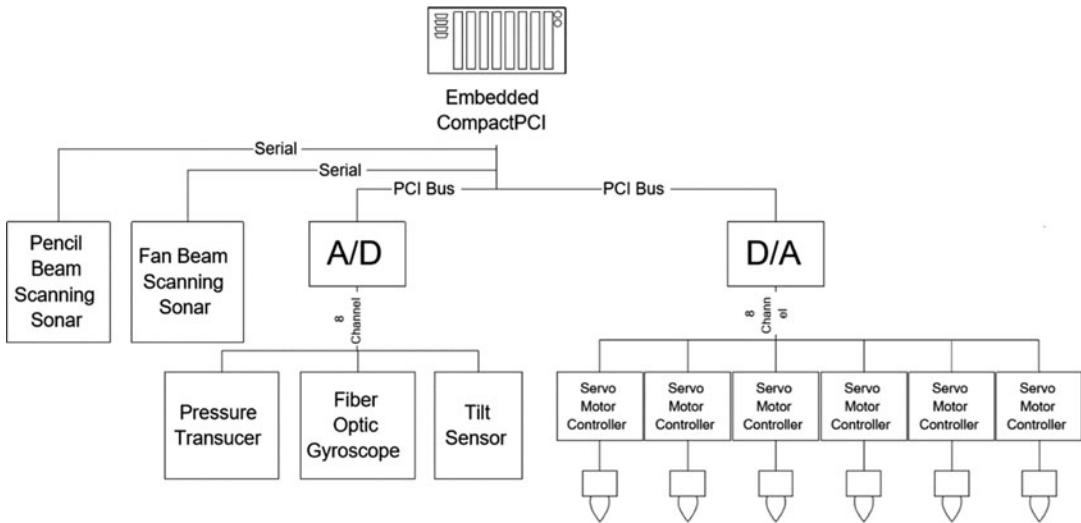
AUV/ROV/HOV Control Systems, Fig. 1 Underwater vehicle system structure (Smallwood et al. 1999)

sensing strategies and control methods. Autonomous navigation using the information provided by the vehicle's on-board sensors represents one of the ultimate goals of the project (Williams et al. 2006).

Embedded Controller

At the heart of the robot control system is an embedded controller. Figure 2 shows a schematic

diagram of the vehicle sensors and their connections. The vehicle uses a CompactPCI system running Windows CE that interfaces directly to the hardware and is used to control the motion of the robot and to acquire sensor data. While the Windows operating system doesn't support hard real-time performance, it is suitable for soft real-time applications, and the wide range of development and debugging tools make it an ideal



AUV/ROV/HOV Control Systems, Fig. 2 Vehicle control system diagram system (Williams et al. 2006)

environment in which to test new navigation algorithms. Time-critical operations, such as sampling of the analog to digital converters, are performed on the hardware devices themselves and use manufacturer-supplied device drivers to transfer the data to the appropriate processes.

The sensor data is collated and sent to the surface using an Ethernet connection where a network of computers are used for further data processing and data logging and to provide the user with feedback about the state of the sub. Communications between the computers at the surface and the sub are via a tether. This tether also provides power to the robot, a coaxial cable for transmitting video data, and a leak detection circuit designed to shut off power to the vehicle in case of a leak. While some effort might have been spent on eliminating the tether, it was felt that the development of the navigational techniques was of more immediate interest.

Sonar

Sonar is the primary sensor of interest on the vehicle. There are currently two sonars on the robot. A sonar unit has been mounted at the front of the vehicle. It is positioned such that its scanning head can be used as a forward and downward looking beam. This enables the altitude above the seafloor as well as the proximity of

obstacles to be determined using the wide of the angle beam of the sonar.

The second sonar is an imaging sonar and has a dual frequency narrow beam sonar head that is mounted on top of the sub and is used to scan the environment in which the sub is operating. It can achieve 360° scan rates on the order of 0.25 Hz. The information returned from this sonar is used to build and maintain a feature map of the environment.

Internal Sensors

A laser Gyro has been included in the robot to allow the robot's orientation to be determined. This sensor provides yaw rate information and is used to control the heading of the sub. We currently estimate the bias in the gyroscope reading prior to a mission. The bias compensated yaw rate is then integrated to provide an estimate of vehicle heading. Because the yaw rate signal will inevitably be noisy, the integration of this signal will cause the estimated heading to drift with time. At present, missions do not typically run for longer than 30 minutes, and yaw drift does not pose a significant problem on this time frame. In the future, a compass will be added to the system to allow us to periodically reset the heading for longer missions. This will also allow us to detect changes in the yaw rate bias that typically

occur as a result of changes in the internal temperature of the unit.

A pressure sensor measures the external pressure experienced by the vehicle. This sensor provides a voltage signal proportional to the pressure and is sampled by an analogue to digital converter on the embedded controller. The pressure can then be converted to a depth below the surface of the ocean. Feedback from this sensor is used to control the depth of the sub.

Camera

A Panasonic camera in an underwater housing is mounted externally on the vehicle. It is used to provide video feedback of the underwater scenes in which the robot operates. This is a color camera and sends the video signal to the surface via the tether. A video acquisition card is then used to acquire the video signal for further image processing.

Thrusters

There are currently six thrusters on the vehicle. Four of these are oriented in the vertical direction, while the remaining two are directed horizontally. This gives the vehicle the ability to move itself up and down, control its yaw, pitch and roll and move forward and backward, and move lateral motion.

Vehicle Control Method

Control system of a mobile robot in six-dimensional space in an unstructured, dynamic environment that is found underwater can be a daunting and computationally intensive endeavor. Navigation and control both present difficult challenges in the subsea domain. We have developed a distributed, decoupled control architecture to help simplify the controller design.

The vehicle control architecture currently running on the sub is based on the Distributed Architecture for Mobile Navigation (DAMN) as proposed in Rosenblatt (1997). This behavior-based control architecture uses a centralized arbiter to combine votes from various behaviors running in the system in order to determine the optimal course of action to pursue. A behavior

encapsulates the perception, planning, and task execution capabilities necessary to achieve one specific aspect of robot control and receives only that data specifically required for that task (Brooks 1986).

Decoupled Control

By decoupling the control problem, individual controller design is greatly simplified. In the case of the typical vehicle, vertical motion is controlled independently of its lateral motion using two separate PID controllers. These controllers are then tuned to provide the required performance in each case.

This division of control has been selected as it fits in with many of the anticipated missions to be undertaken by the vehicle. A typical mission might see the vehicle performing a survey of an area of the Great Barrier Reef while maintaining a fixed height above the seafloor (Williams et al. 1999). The task of performing the survey can then be made independent of maintaining the vehicle altitude. This also allows us to optimize the performance of the depth controller prior to deploying the navigation algorithms used for mapping of the environment to ensure that the vehicle has little risk of hitting the sea floor.

The behaviors that run on the vehicle are also divided into horizontal and vertical behaviors, and there are consequently two arbiters currently present. One is responsible for setting the desired depth of the vehicle, while the other sets the desired yaw and forward offset to achieve horizontal motion.

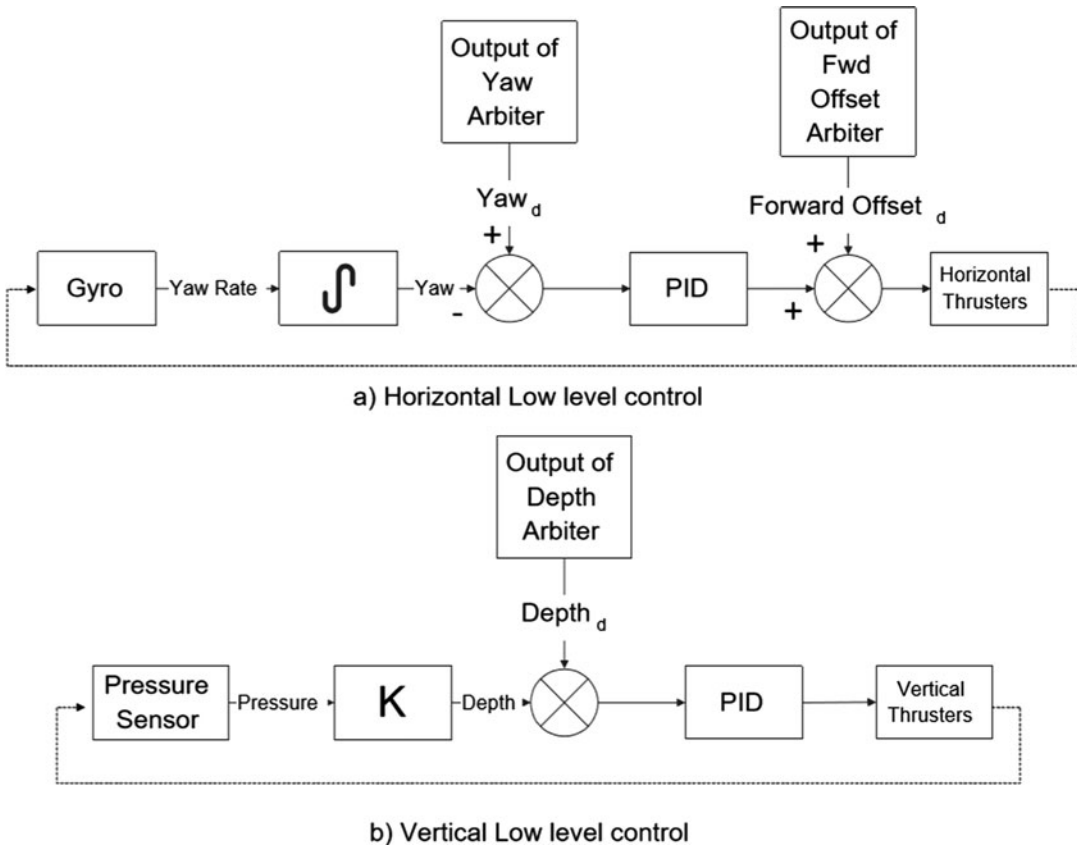
For a survey mission, the vertical behaviors would typically be responsible for keeping the sub from colliding with the seafloor. The vertical behaviors that run on the vehicle include maintain minimum depth, maintain minimum altitude, maintain depth, and maintain altitude. The combination of the outputs of these behaviors determines the depth at which the vehicle will operate. A large negative vote by the maintain minimum altitude behavior will keep the vehicle at a minimum distance from the seafloor. The horizontal behaviors that would run on a survey mission include follow line (sonar and/or vision), avoid

obstacles, move to location, and perform survey. The combination of the outputs of these behaviors determines the orientation maintained by the vehicle as well as the forward offset applied to the two horizontal thrusters. This allows the vehicle to move forward while maintaining its heading.

Low- and High-Level Control

The control of the sub is further decoupled into low level- and high-level control processes. The low-level processes run on the embedded controller and are used to interface directly with the hardware (see Fig. 3). In addition, the PID controllers have been implemented at this level. This allows the PID controllers to respond quickly to changes in the state of the sub without being affected by the data processing and high-level control algorithms running at the surface.

The high-level processes run on a pair of Pentium machines at the surface control station. Information from the sub's sensors is fed to a series of processes running on these machines. These processes use the data supplied by the sensors to determine the desired set points for the low-level control processes. The raw sensor data is preprocessed to produce virtual sensor information – information that is of interest to multiple behaviors in the system. The behaviors register their interest in the data being generated by the virtual sensors and send votes to the arbiters to specify their desired course of action. A task-level mission planner is used to enable and disable behaviors in the system depending on the current state of the mission and its desired objectives. The arbiter combines the votes from the behaviors and selects the optimal action to satisfy the goals of the



AUV/ROV/HOV Control Systems, Fig. 3 The low-level control processes that run on the vehicle controller. These processes include processes for sampling the internal

sensor readings, computing the PID control outputs, and driving the thrusters (Williams et al. 2006)

system. A schematic representation of the data flow within the system is shown in Fig. 4. This control strategy relies on the ability of the sub to gather information about its environment and reason about its desired actions. By continuously monitoring the state of the environment, the sub is able to respond to changes as they occur.

Distributed Control

A number of processes have been developed to accomplish the tasks of gathering data from the robot's sensors, processing this data, and reasoning about the course of action to be taken by the robot. These processes are distributed across a network of computers and communicate asynchronously via a TCP/IP socket-based interface using a message passing protocol developed at the center.

A central communications hub is responsible for routing messages between the distributed processes running on the vehicle and on the command station. Processes register their interest in messages being sent by other processes in the system, and the hub routes the messages when they arrive. While this communications structure has some drawbacks, such as potential communications bottlenecks and reliance on the performance of the central hub, it does provide some interesting possibilities for flexible configuration, especially during the development cycle of the system.

The above introduction mostly comes from Smallwood et al. (1999) and Williams (2006). More technical details or design method can be found in their related literature.

Key Applications

Over the past 20 years, with the widespread development of large-scale integrated circuit in the world, which plays an important role in the field of marine technology, many new submarine vehicles have been created to solve a wide range of problems. These devices have already demonstrated their effectiveness in performing emergency rescue, ocean survey, environment monitoring, inspections and operation, research, and other types of work.

Because the marine environment is unstructured and hazardous, the development of underwater vehicles will face many challenging scientific and engineering problems, and researchers have made great efforts to overcome these problems so far (Paull et al. 2014). The progress in new materials, sensor technology, computer technology, and advanced algorithms has greatly contributed to research and development activities in the underwater vehicle community.

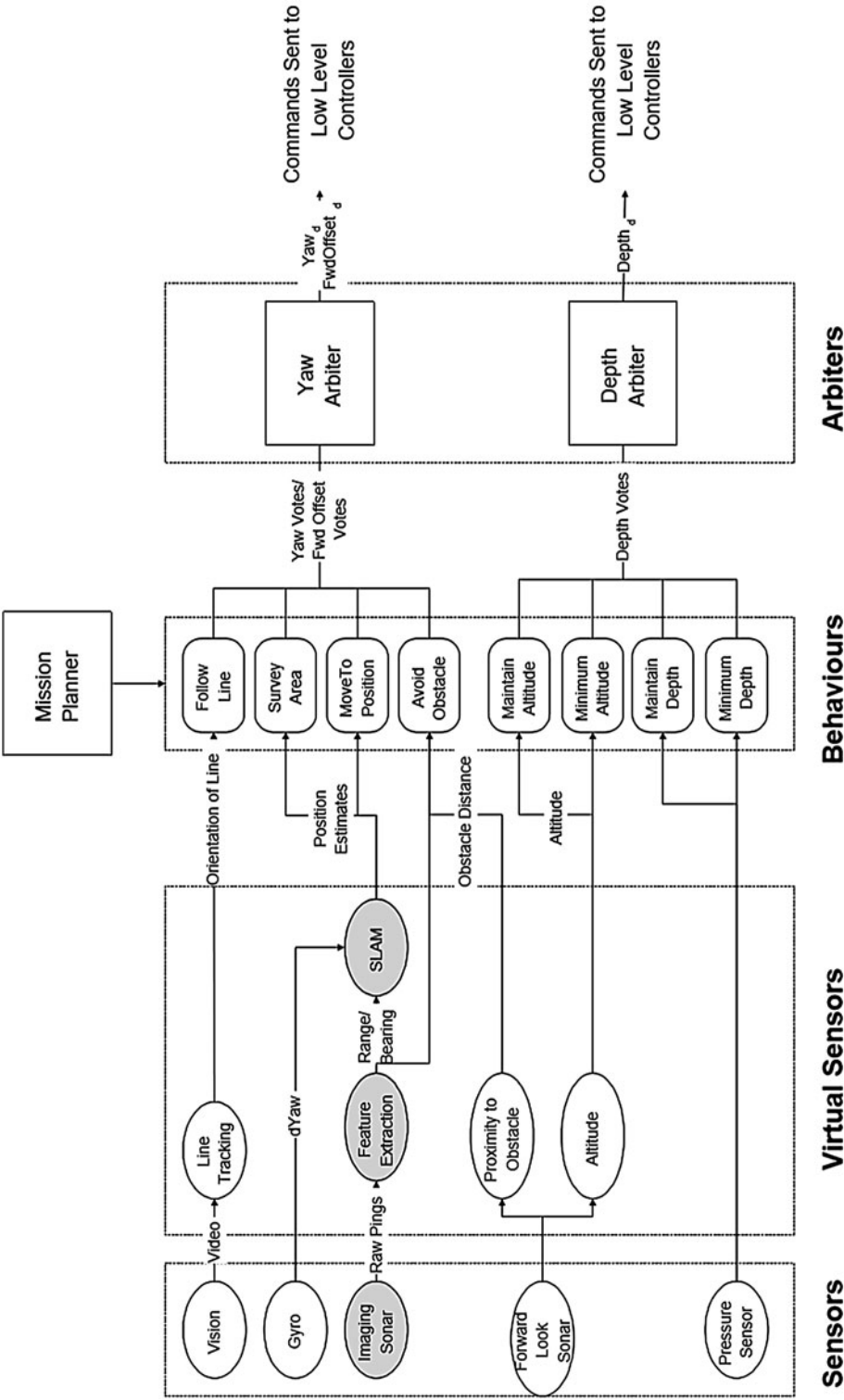
With technological advances, underwater vehicles have the advanced control technology and intelligent control system that can efficiently perform scheduled tasks with its own decision-making and control capabilities.

Advanced Control Technology

The AUV control methodologies that have been proposed in the literature include linear control (see Yildiz et al. (2009)), sliding mode control (see Yang et al. (2013) and Kim et al. (2015)), and fuzzy and neural network control (see Khodayari and Balochian (2015) and Lakhekar et al. (2015)).

One simple and effective nonlinear control design approach that has been implemented in marine applications is dynamic inversion (DI), in which the control law is formulated to eliminate system nonlinearities by means of feedback. The DI control technique in turn allows to incorporate well-established linear control techniques. In spite of its flexibility and simplicity, standard DI lacks robustness to model uncertainties. Several modifications were added to the basic DI control structure to improve its robustness characteristics; see Steinicke and Michalka (2002) and Wang et al. (2012). Regardless of these attributes, DI has several shortcomings and limitations, including blind nonlinearity cancellation, large control effort, and numerical singularity configurations of square matrix inversion.

A new inversion-based control design methodology is Generalized Dynamic Inversion (GDI); see and Bajodah (2009). The methodology is of the left inversion type, and hence it does not involve deriving inverse equations of motion for the plant. In GDI, dynamic constraints are prescribed in the form of differential equation that encapsulates the control objective and is



AUV/ROV/HOV Control Systems, Fig. 4 The high-level process and behaviors that run the vehicle. The sensor data is preprocessed to produce virtual sensor information available to the behaviors. The behaviors receive the virtual sensor information and send votes to the arbiters who send control signals to the low-level controllers (Williams et al. 2006)

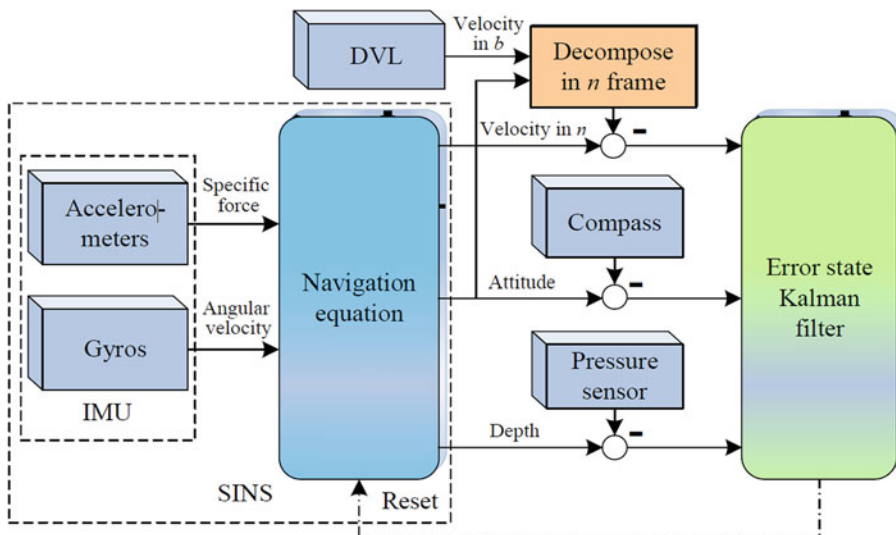
generalized inverted using the Moore-Penrose Generalized Inverse (MPGI)-based Greville formula. The GDI control technique has been an effective tool for several engineering problems; see Bajodah (2009), Bajodah (2010), Hameduddin and Bajodah (2012), and Gui et al. (2013). The Robust Generalized Dynamic Inversion (RGDI) control system is obtained by Ansari and Bajodah (2017).

Location and Navigation Technology

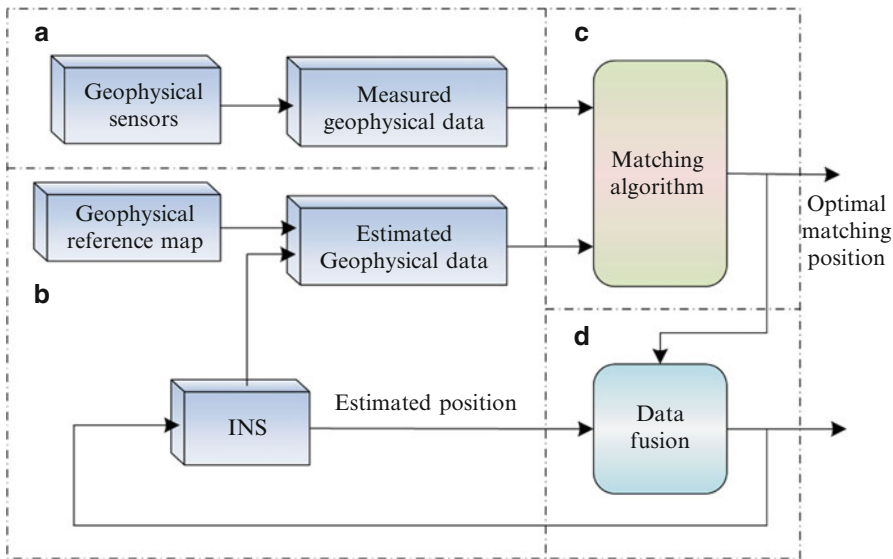
Navigation is one of the key AUV technologies because the localization, path tracking, and control of the vehicle are all based on precise navigation parameters. Some navigation methods commonly used for land and air are not suitable for underwater because of the attenuation effect of water on electromagnetic signals, and underwater navigation has become a challenging issue in AUV research (Paull et al. 2014). Among the many underwater navigation systems available, the inertial navigation system (INS) using inertial sensors typically acts as the central navigation system of AUVs because of its autonomy (Stutters et al. 2008).

Generally, the INS contains an inertial measurement unit (IMU), which consists of accelerometers measuring linear acceleration and gyroscopes measuring angular velocity; the

accelerometers and the gyroscopes are usually made up of three mutually perpendicular accelerometers and three mutually perpendicular gyroscopes, respectively. For inertial navigation, the instantaneous speed and position of the vehicle are obtained by integrating the measured values of the accelerometers and gyroscopes. The errors of the IMU increase with increasing elapsed time due to the drift of accelerometers and gyroscopes. Theoretically, the velocity and heading errors accumulate linearly over time, and the position error accumulates exponentially over time (Ali and Mirza 2010). Therefore, the INS can provide relatively accurate navigation information within a short time, but it is physically impossible for a pure inertial navigation system to maintain the high-precision level throughout a mission. Aiding the INS with external information or measurements is an effective means of improving navigation accuracy and has been widely used. In AUV navigation, auxiliary sensors or other navigation systems, such as a Doppler Velocity Log (DVL), compass, pressure sensor, Global Positioning System (GPS), acoustic positioning system (APS), or geophysical navigation system, are usually combined with the INS to form an integrated navigation system (Kussat et al. 2005 and Vasilijevic et al. 2012) (Figs. 5 and 6).



AUV/ROV/HOV Control Systems, Fig. 5 SINS/DVL integrated navigation structure (Bao et al. 2019)



AUV/ROV/HOV Control Systems, Fig. 6 Principle diagram of geophysical navigation-assisted inertial navigation (Bao et al. 2019)

AUVs typically use the INS as their main navigation system; however, due to inherent error accumulation of inertial sensors over time, the pure INS has difficulty obtaining long-range precision navigation. Therefore, INS is often combined with other navigation systems or devices such as GPS, DVL, acoustic positioning systems, or geophysical navigation systems to improve the AUV navigation accuracy (Bao et al. 2019).

Cross-References

- ▶ [Automatic Control System](#)
- ▶ [Control Approach](#)
- ▶ [Control Device](#)
- ▶ [Control Module](#)
- ▶ [Control Unit](#)
- ▶ [Controller](#)
- ▶ [Motion Control System](#)

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AUV/ROV/HOV Hydrostatics

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Synonyms

Buoyancy; Density; Displacement volume; Fixed weight; Hydrostatics pressure; Salinity; Submerged depth; Temperature; Variable weight; Weight

Definition

Hydrostatics is a kind of ability to enable an AUV/ROV/HOV afloat in the certain diving depths. The relationship between buoyancy and weight results in two primary flotation conditions: surfaced and submerged. Variation in weight must be supported by the corresponding buoyancy for each condition. Many configurations, such as the suitable location of the main ballast tanks (MBTs), solid droppable ballast, and buoyancy material, may be adopted to control hydrostatics of submersibles for different purposes.

Scientific Fundamentals

As with any object in a fluid, a submerged AUV/ROV/HOV must conform to Archimedes Principle, which is applied on by two opposite direction forces: upward buoyancy (due to the displacement of water by its hull) and downward weight (Renilson 2015). The AUV/ROV/HOV is sensitive to variation in both buoyancy and weight. An exact equality of these two forces is termed “neutral” buoyancy, which is a hover condition (to keep a given depth while stationary); if weight exceeds buoyancy, known as “negative” buoyancy, the submersible will drop and deeply submerge to the seabed. Conversely, if buoyancy exceeds weight, known as “positive” buoyancy, the submersible will rise to float on the water surface (Burcher and Rydill 1994).

An AUV/ROV/HOV on the surface has to satisfy the same hydrostatic principles. However, it has a smaller watertight volume above water and relatively lower water-plane characteristics than that of a surface ship. With the flooding water into the ballast tanks, the weight of AUV/ROV/HOV increases and the submersible transitions from floating on the surface to being fully submerged. Correspondingly, the immersed volume, termed “displacement,” is also enlarged until the integral structure volume (maximization).

Buoyancy Force

The buoyancy force of submersible is given by the immersed volume, partly or fully, multiplied by the water density and gravity acceleration, i.e.,

$F = \rho gV$, where ρ is the density of seawater, g is the acceleration due to gravity, and V is the immersed volume of the submersible.

Hydrostatic Pressure

The force acting on the submerged AUV/ROV/HOV is the weight of the seawater column above it. Since seawater can be considered incompressible and the variation of g can be neglected (the height h of fluid column is much smaller than the radius of Earth), hydrostatic pressure can be calculated according to the following formula: $p - p_0 = \rho gH$, where p is the hydrostatic pressure, p_0 is the atmospheric pressure on sea surface, and H is the height from the surface to the submerged depth.

Displacement Volume

The changes in the geometry of the AUV/ROV/HOV will be caused by hydrostatic pressure loading with depth. The hydrostatic pressure is increased with the greater submerged depth, which results in a reduction in displacement volume. Therefore, the buoyancy will be reduced due to the compression as diving depth increases, especially for the deep submersible. On the other hand, as submerging, the temperature around the AUV/ROV/HOV is different, which has also an effect on the immersed volume, especially for the volume of solid buoyancy material. That is to say, the displacement volume is the function of hydrostatic pressure and the temperature (Tie dong 2010). The changes in displacement volume due to the variation in these two parameters result in different buoyancy subsequently.

Density

Buoyancy is not only dependent on the volume but also on the density of the water. And the density is varied if the AUV/ROV/HOV is operated in different areas. Seawater specific gravity (SG) can range from 1.00 near the mouths of rivers or in fjords where the water is virtually fresh due to melting glaciers, to about 1.03 where there is high salinity. A standard value of 1.0275 toward the higher end of the density range is averagely encountered in a submersible operation. When the AUV/ROV/

HOV approaches a coast or where there are substantial changes with depth (layers) of temperature and salinity, a quite large change may occur rapidly, and a variation of 3% in buoyancy due to seawater density variation would be adopted in the design (Burcher and Rydill 1994).

As well as salinity effecting to the density of displacing seawater around submersible, the hydrostatic pressure due to the submerged depth from the sea surface and the temperature also affect the density. With the increase of the submerged depth, hydrostatic pressure will be larger and the temperature will be lower. For the same salinity, the lower temperature results in a higher density of water; while for the same temperature, the higher salinity leads to higher density. Therefore, the density is determined by the salinity, the temperature, and the hydrostatic pressure. Then any changes of density will result in the variation of buoyancy.

Weight

The determination of the weight of an AUV/ROV/HOV is more complicated than that of buoyancy. Whereas buoyancy is primarily related to gross volumes of geometrically simple forms and the density of displacing water; weight is the summation of every element (fixed weight and variable weight) that goes into a submersible. The fixed weight consists of large components, such as pressure hull structure, main propulsion plant, batteries, etc., and some minor structures and systems, which are permanently installed components. The variable weight associates with crew, their effects, stores, and the fluid contents of many tanks.

TRIM

The mass and longitudinal center of gravity of a submersible will change during the operations. In addition, changes in seawater density and hull compressibility will all result in small changes to buoyancy and position of the center of buoyancy. Therefore, it cannot be kept in upright position any more, maybe in longitudinal inclination equilibrium because of these changes.

Key Applications

Unlike for a surface ship, in the case of a deeply submerged AUV/ROV/HOV, the immersed volume cannot be increased by increasing the vessel's draught. Thus, for equilibrium in the vertical plane, the mass must be balanced exactly by the buoyancy force. The state of equilibrium in a given submergence depth tends to be unstable. A slight upward or downward movement from this position will result in the boat moving away from this initial position. Further, variation in buoyancy will occur continuously due to small changes in seawater density in the vertical and horizontal planes, while variation in weight will mainly be due to the consumption of stores, ballast, or collection samples. These will also have a significant influence on the ability to control the submersible in the vertical plane.

Most submersibles have an operational requirement to stop or move at very low speed (hover condition). It can be contrived to keep weight and buoyancy as nearly equal as possible, and means or special control devices for doing so are provided. Another reason for neutral buoyancy is related to safety. If the submersible is in power failure and not in neutral buoyancy, it will lose depth control. If lighter than buoyancy, it will rise uncontrolled to the surface. It must, however, be a safety design criterion for HOV. The opposite situation, i.e., heavier than buoyancy, is potential disaster when the submarine will plunge beyond its collapse depth or hit the sea bed.

Main Ballast Tanks (MBTs)

To keep the depth control, the main ballast tanks (MBTs) are fitted in a suitable location for the submersible where it is flooded or discharged with seawater, either wholly or partially. When it submerges, the submersible must either increase its weight or reduce its buoyancy. Conversely, an increase in buoyancy or a reduction in weight leads it to rise from submerged to surface.

Solid Droppable Ballast

For deep submersible, which is likely to be for exploration, some solid droppable ballast will be

added. The initial weight is larger than buoyancy, which leads the submersible to sink quickly. When the given depth arrives, part of the solid ballast will be abandoned. Then the submersible will be in the hover condition. After cruising is finished, the rest of the solid ballast will be thrown away. The weight is much less than the buoyancy, and the submersible will be back to the surface quickly.

Buoyancy Material

Adjustment of the relationship of weight and buoyancy is operated by flooding or discharging the seawater in MBTs. They can be readily enough flooded from the sea through the tanks valves, but discharging water from the tanks is not so easy. The water can be blown out to sea using high-pressure air, but that is noisy and subsequent venting of the tanks raises atmospheric pressure inside the boat. Alternatively, a pumping system can also be used to discharge to sea, and all the piping is required to withstand diving pressure. It may be useless for the deep submersible.

Some solid buoyancy material will be adopted for the deep submersible. The MBTs are fully flooded, and most of the buoyancy will be provided by this material when it is submerged in deep sea. The MBTs are only discharged when the submersible is almost back to surface, which help to float on the surface better.

Compensating Tanks

Compensating tanks are used for correcting the trim of a submerged AUV/ROV/HOV, which usually is composed of two tanks, one forward and another afterward. To correct the longitudinal balance, one tank will be emptied and another filled. The amount of correcting depends on the current situation of the submerged submersible.

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AUV/ROV/HOV Propulsion System

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Synonyms

Autonomous underwater vehicle (AUV); Brushless DC motor (BLDCM); Human-occupied vehicle (HOV); Magnetohydrodynamics (MHD); Remote operated vehicle (ROV); Tandem propulsion system (TPS)

Definition

The AUV/ROV/HOV propulsion system is an underwater energy conversion device that converts other forms of energy into kinetic energy of AUV/ROV/HOV. It consists of several thrusters that work together to propel the AUV/ROV/HOV to complete various complex movements underwater, such as cruise, search, hover, and other complex sports. Due to different missions, different underwater vehicles have different requirements for their own propulsion systems. In general, the propulsion system of an underwater vehicle should have good maneuverability, efficient propulsion efficiency, and low weight.

Scientific Fundamentals

Fundamentals of Propulsion System

Theory of Ideal Thruster

A disk surface of diameter D having no thickness has the function of absorbing external power and pushing water to obtain an axial induction speed so that the ideal disk surface is called ideal thruster, also called agitating plate. The ideal thruster makes the following simplifying assumption: an ideal and incompressible fluid; thruster operating in an

unbounded inflow; an actuator disk with no thickness and the water is accelerated only in the axial direction when passing through the disk; velocity and pressure distributions over the disk are uniform. As shown in Fig. 1, when passing through the ideal thruster, the velocity and pressure of the control volume will change.

Thrust and induction velocity of ideal thruster:

1. Fluid quality flowing through the agitating plate per unit time:

$$m = \rho A_0 (V_A + \mu_{a1})$$

A_0 – disk area, $A_0 = \pi D^2/4$

ρ – density of water

V_A – around the fluid velocity of disk

2. The momentum of inflow:

$$mV_A$$

The momentum of outflow:

$$m(V_A + \mu_a)$$

The momentum of increment:

$$m(V_A + \mu_a) - mV_A = m\mu_a = \rho A_0 (V_A + \mu_{a1}) \mu_a$$

3. According to the momentum theorem: the force acting on the fluid is equal to the increment of momentum per unit time (ideal thrust):

$$T_i = \rho A_0 (V_A + \mu_{a1}) \mu_a \quad (1)$$

4. Application of Bernoulli's equation

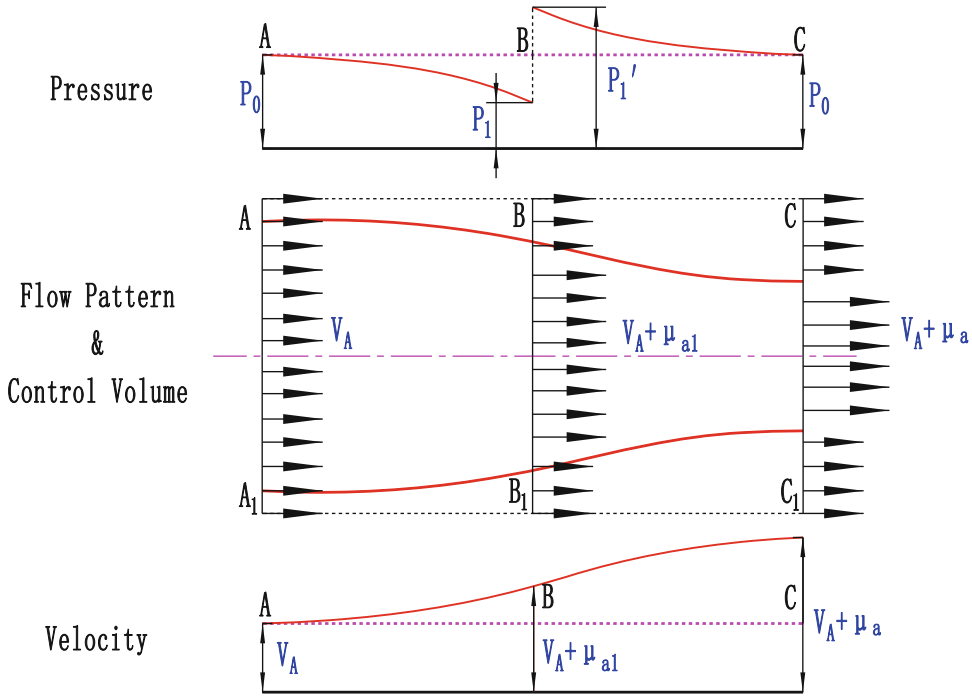
Far upstream to the front side of the disk:

$$P_0 + \frac{1}{2} \rho V_A^2 = P_1 + \frac{1}{2} \rho (V_A + \mu_{a1})^2 \quad (2)$$

The rear side of the disk to far downstream:

$$P'_1 + \frac{1}{2} \rho (V_A + \mu_{a1})^2 = P_0 + \frac{1}{2} \rho (V_A + \mu_a)^2 \quad (3)$$

Subtracting one equation from another:



AUV/ROV/HOV Propulsion System, Fig. 1 Pressure and velocity of ideal thruster

$$P_i = P'_1 - P_1$$

The disk thrust is equal to the pressure difference:

$$T_i = A_0(P'_1 - P_1) = \rho A_0 \left(V_A + \frac{1}{2} \mu_a \right) \mu_a \quad (4)$$

5. By comparing Eqs. (1) and (4), the induction velocity at the disk surface can be obtained:

$$\mu_{a1} = \frac{1}{2} \mu_a \quad (5)$$

μ_{a1} – axial induction velocity of the disk

μ_a – axial induction velocity of far downstream

Propeller Propulsion Theory

The movement of underwater vehicles is usually achieved by propellers, which are simple and widely used thrusters. The function of a propeller is to convert the torque to thrust. Thrust is another name for force, and a basic axiom of

mechanics tells us that force (thrust) is equal to mass multiplied by acceleration. In other words, if we apply acceleration to a large amount of water, we will generate thrust or push that will accelerate the underwater vehicle forward when a large amount of water moves in the opposite direction. As shown in Fig. 2, the blades of the propeller and the shaft are supported on the propeller hub at a certain angle, and the propeller hub is driven by the propeller shaft. When the propeller rotates one revolution, the forward distance in the axial direction is called the process, and the difference between the pitch and the process is called slippage, and the ratio of slippage to pitch becomes the slip ratio, which can be expressed as:

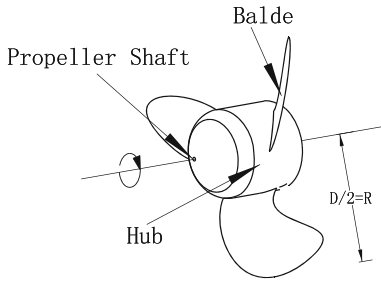
$$S_a = \frac{H - v/n}{H} \quad (6)$$

S_a – slip ratio

H – slip, m

n – propeller rotate speed, rpm

v – the speed of underwater vehicles, m/S_o



AUV/ROV/HOV Propulsion System, Fig. 2 Propeller

In order to generate thrust, the propeller must accelerate or move a large amount of water astern. Now, when the propeller rotates, it looks like a screw and slides over the water without moving any water. If this happened, the propeller would advance an amount equal to the propeller pitch, which is called zero slip. But in order to produce thrust, we must accelerate or move water, and therefore, it is apparent that the propeller will not advance the full amount of pitch in each revolution. In order to generate thrust, we must have slip, and the amount of slip will be proportional to the thrust required by the vehicle. The propeller thrust T can be expressed as:

$$T = \rho n^2 D^4 K_T \quad (7)$$

ρ – the density of water, kg/m^3
 n – propeller rotation speed, rps
 D – propeller diameter, m
 K_T – thrust coefficient

In order to generate this thrust, a torque must be applied to the propeller to rotate the propeller and the water is pushed out by the blades. The torque Q can be expressed as:

$$Q = \rho n^2 D^4 K_Q \quad (8)$$

K_Q – torque coefficient

Thrust coefficient and torque coefficient are both functions of propeller geometric parameters.

Thruster Type

The core component of the propulsion system is the thruster. Different types of thrusters have been



AUV/ROV/HOV Propulsion System, Fig. 3 Electric thruster

developed according to the type of underwater vehicle and the environment in which it is used. At present, the thrusters commonly used in AUV/ROV/HOV propulsion systems mainly include electric thrusters, hydraulic thrusters, water-jet thrusters, hubless rim-driven thrusters, tandem thruster system, magnetohydrodynamic drive thruster, bionic thrusters, etc.

Electric Thruster

Most battery-powered underwater vehicles use motors to directly drive the propeller. DC motors and AC motors can be used. The DC motor has low cost, the speed control and the control system are simple, and the AC motor needs the inverter to convert the DC into AC, which has high cost and complicated system. In particular, underwater vehicles that use battery packs as a power source use DC motors. Brushless DC motor is a new type of DC motor developed with the rapid development of electronic technology in recent years. Its most prominent feature is that there is no complicated commutation mechanism. Usually, permanent magnet is used as rotor and has no excitation loss, no commutation spark, no radio interference, easy operation, and maintenance (Fig. 3).

Hydraulic Thruster

Some large and medium-sized underwater vehicles use hydraulic motors as propulsion power. Compared with the underwater motor, the thruster with hydraulic motor has the following advantages: the hydraulic propulsion system controlled



AUV/ROV/HOV Propulsion System, Fig. 4 Hydraulic thruster of sub-Atlantic



AUV/ROV/HOV Propulsion System, Fig. 5 Hubless rim-driven thruster

by the flow has good speed regulation performance, and it is easy to realize stepless speed regulation in a wide range; the hydraulic propulsion system usually has hydraulic compensation, and it is easier to realize the sealing of the system; the safety is good, the reliability is high, the installation position is flexible, and the cost is low. As shown in Fig. 4, the hydraulic thrusters manufactured by Sub-Atlantic are used in many ROVs.

Water-Jet Thruster

The water-jet thruster is a propulsion device that uses the reaction of the high-speed water jet ejected from the jet pipe to provide thrust. It is similar to the aviation jet thruster, except that water instead of air combustion mixed gas is used as the working medium. After the water is obtained at a high speed by a large-flow high-pressure water pump installed on the underwater vehicle, the jet is ejected to generate thrust. The advantage of the water-jet propulsion device is that almost all accessories on the underwater vehicle are eliminated; the loss of the shafting and transmission and the loss of the thruster's cavitation are eliminated; the underwater vehicle can be easily manipulated.

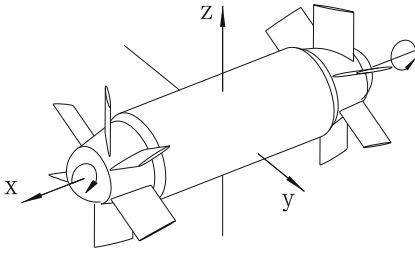
Hubless Rim-Driven Thruster

In recent years, a propulsion device integrating the structure of a powerful motor and a propeller has appeared, which is called hubless rim-driven

thruster (Yan et al. 2017). As shown in Fig. 5, the hubless rim-driven thruster is mainly composed of ducts, propeller blades, armature, rare-earth magnetic steel, front and rear sliding bearings, bearing supports, and other components. The hubless rim-driven thruster integrates the rotor of the motor with the propeller blades of the thruster. The stator of the motor is integrated into the duct of the thruster. When the motor is running, the rotor of the motor drives the propeller blades to rotate, generating thrust and driving the vehicle to sail. The propeller blades without hub are supported by two sliding bearings that are fixed to the front and rear ducts. Without the rear guide vanes, the space occupied by the propeller hub can be fully utilized, and the blades can be protected from external impurities damage. Due to its compact structure, high efficiency, low vibration and noise, good cooling conditions, no need for dynamic sealing, modular design and disassembly, it has become a hot research topic in the field of underwater vehicles (Song et al. 2015).

Tandem Propulsion System (TPS)

In 1961, American engineer F. R. Haselton proposed the concept of tandem propulsion system (TPS) (Nanba and Shimizu 1988). After more than 20 years of efforts, in 1985, the engineers of Ametek designed and manufactured an underwater vehicle with TPS. The TPS requires two propellers that are oriented in the longitudinal



AUV/ROV/HOV Propulsion System, Fig. 6 Tandem thruster system

direction of the underwater vehicle (usually mounted on the bow and stern respectively), allowing for flexible control of the underwater vehicle in 6 degrees of freedom, as shown in Fig. 6. This type of propulsion system actually places the stator of the electric machine in the underwater vehicle and supports the rotor on the outer casing of the underwater vehicle, and the blade is directly mounted on the rotor. It differs from a conventional underwater vehicle propeller in that the pitch of the blade of a conventional propeller is usually unchanged, while the blade of the omnidirectional propeller changes at every moment, and the pitch variation of each blade is not according to the same law.

Compared with other thrusters, TPS is simple in construction, easy to manufacture, and maintain, especially, when the diameter of the vehicle is limited, TPS has higher efficiency than conventional thrusters, thus saving energy and reducing ROV cable diameter and cable resistance. For HOV and AUV, the weight and volume of the power supply can be reduced. TPS can better absorb external load changes and reduce the impact on underwater vehicles, and because its power and thrust are borne by two sets of blades, TPS has better tail vibration performance than conventional thrusters. In addition, the tail shaft of the TPS system is long, the weight is large, the price is high, the seal is easy to leak, and the control system is complicated and requires computer-aided control. Therefore, the TPS system has not been put into practical use yet, but it represents one of the developments of the underwater vehicle propulsion system, and it will be widely used in future.

Magnetohydrodynamic Drive Thruster

A magnetohydrodynamic drive thruster or MHD accelerator is a method for propelling underwater vehicles using only electric and magnetic fields with no moving parts, accelerating an electrically conductive propellant (liquid or gas) with magnetohydrodynamics (Takeda et al. 2005). The fluid is directed to the rear and as a reaction, the underwater vehicle accelerates forward.

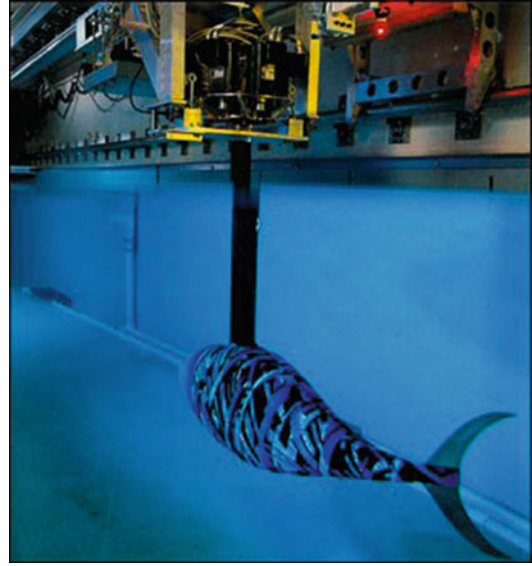
The working principle involves the acceleration of an electrically conductive fluid by the Lorentz force, resulting from the cross product of an electric current (motion of charge carriers accelerated by an electric field applied between two electrodes) with a perpendicular magnetic field. The Lorentz force accelerates all charged particles (positive and negative species) in the same direction whatever their sign, and the whole fluid is dragged through collisions. As a reaction, the vehicle is put in motion in the opposite direction. This is the same working principle as an electric motor (more exactly a linear motor) except that in an MHD drive thruster, the solid moving rotor is replaced by the fluid acting directly as the propellant (Bui et al. 2010).

MHD thrusters are classified into two categories according to the way the electromagnetic fields operate: Conduction devices and induction devices. Conduction devices when direct current flows in the fluid due to an applied voltage between pairs of electrodes, the magnetic field being steady. Inductive devices when the alternating current is caused by a rapidly changing magnetic field, such as eddy currents. In this case, no electrode is required.

MHD is attractive for underwater vehicle applications because it has no moving parts, which means that a good design might be silent, stealthy, reliable, and efficient. Additionally, the MHD design eliminates many of the wear and friction pieces of the drivetrain with a directly driven propeller by an engine. The major problem with MHD is that with current technologies, it is more expensive, and much slower than a propeller driven by an engine. In 1991, the world's first full-size prototype Yamato 1 was completed in Japan after 6 years of R&D by the Ship & Ocean Foundation (Fig. 7). The ship successfully carried a



AUV/ROV/HOV Propulsion System, Fig. 7 MHD Thruster of Yamato I



AUV/ROV/HOV Propulsion System, Fig. 8 RoboTuna II

crew of 10 plus passengers at speeds of up to 15 km/h in Kobe Harbor in June 1992.

Bionic Thruster

The traditional underwater propulsion system is propeller propulsion, but the efficiency is relatively low. Modern bionic technology provides new ideas for the development of new water propulsion systems. Fish and other marine life have evolved in the ocean for millions of years, and their perfection is beyond imagination. The data shows that fish in the ocean by the fins and body twist to provide forward momentum, jellyfish through the squeeze and shrink umbrella to provide forward momentum, its advancement is almost perfect, and the efficiency is as high as 98% (Naomi 2000).

At present, many countries are actively developing bionic underwater vehicles, such as Charlie I and RoboTuna II of Massachusetts Institute of Technology, as shown in Fig. 8, UPF-2001 and PF-700 of Japan's National Marine Research Center, Bionic Robotic Eel, and SPC of Beijing University of Aeronautics and Astronautics.

Because the performance of materials is far from the advanced level of marine biological muscles, the biomimetic vehicle test platform or prototype developed by scientists in various countries at present is a simple mechanical transmission to simulate the promotion of marine organism.

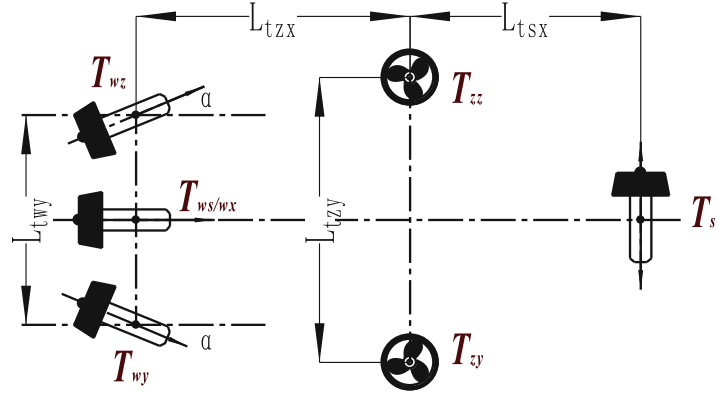
However, the test results of various countries show that the propulsion efficiency of these bionic thrusters is higher than that of ordinary propeller propulsion. Therefore, with the deepening of research and breakthroughs in key technologies, the bionic thruster will be more widely used.

Control Allocation of Thrust

Control allocation of thrust is the core of the control system of underwater vehicles (Liu et al. 2018). By properly arranging the number and position of the thrusters on the surface and inside of the vehicle, and properly setting the thrust of each thruster, the complex underwater movement of the submersible is completed (Sordalen 1997), as shown in Fig. 9. The underwater vehicle is a highly nonlinear coupled system with high nonlinearity. It is difficult to obtain accurate hydrodynamic coefficients, so it is difficult to establish accurate maneuvering motion models (Johansen et al. 2004). At the same time, environmental disturbances, such as ocean currents, are difficult to predict (Lindegard and Fossen 2003). Therefore, most propulsion system power distribution does not depend on precise mathematical models. Intelligent control combines the artificial intelligence method with the feedback control theory and has obvious advantages in dealing with

AUV/ROV/HOV Propulsion System,

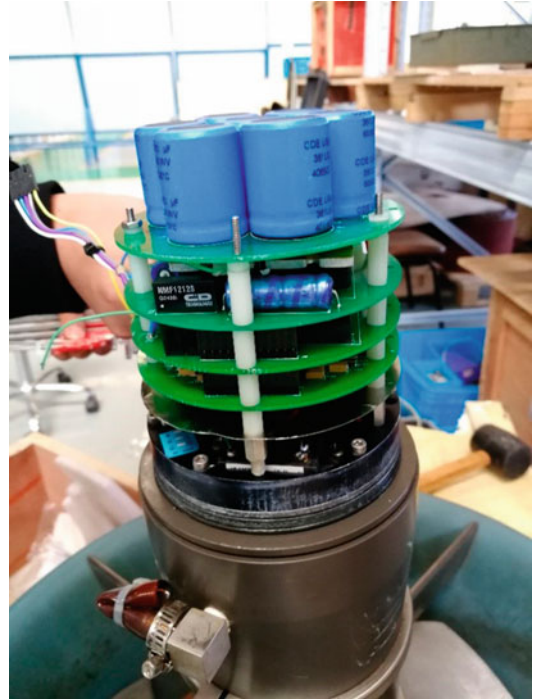
Fig. 9 Thrusters distribution of “Jiaolong” HOV



complexity and uncertainty. Therefore, when the system thrust distribution is promoted, the intelligent control method can obtain sound control effects. At present, the intelligent control methods commonly used in underwater vehicle propulsion systems mainly include classical PID control, fuzzy control, neural network control, synovial variable structure control, and S-plane control method (Cao et al. 2013).

Motor of Thruster

The brushless DC motor (BLDCM) has the advantages of simple structure, high efficiency, reliable operation, low mechanical noise, good speed regulation, etc., and is widely used in underwater thrusters. Conventional brushless DC motors cannot be used directly in seawater high-pressure environments, and motors for underwater thrusters require unique design. First, in order to prevent the brushless DC motor from being damaged by high pressure, the motor housing must be filled with insulated aviation hydraulic oil and connected to a compensator installed inside or outside the thruster to ensure that the oil pressure in the motor housing is always slightly higher than the external seawater, to prevent external seawater from entering the motor housing. Secondly, the waterproof and pressure-resistant design of the motor drive circuit, the housing of motor drive circuit is mainly divided into waterproof electronics housing and wet electronics housing. The waterproof electronics housing is protected against seawater pressure by adding a pressure-proof sealed casing to isolate



AUV/ROV/HOV Propulsion System, Fig. 10 Waterproof electronics housing

seawater and ensure the drive circuit is at operating pressure. Working in the environment, the drive circuit does not need special design, as shown in Fig. 10. In the wet electronics housing, the drive circuit is directly immersed in the oil to isolate the seawater by oil, but the drive circuit components need to withstand the pressure of the surrounding oil, so components that cannot

withstand pressure need to be designed with special protection packages.

Rotary Dynamic Seal of Underwater Thruster

When rotating, there is a gap between the shaft and the motor housing. In order to transfer the torque of the motor to the propeller, the problem of the rotary dynamic seal of the shaft must be solved. If an O-ring is added between the shaft and the motor housing, the O-ring will wear, heat, and deform when the shaft rotates at high speeds, and is liable to leak under high seawater pressure. Current practice has proved that the method of adding O-ring seal is not reliable. For this reason, engineers and technicians have developed a variety of advanced dynamic sealing devices, including mechanical dynamic seals, underwater rotary oil seals, and magnetic coupling isolation seals and so on.

Mechanical Dynamic Seal

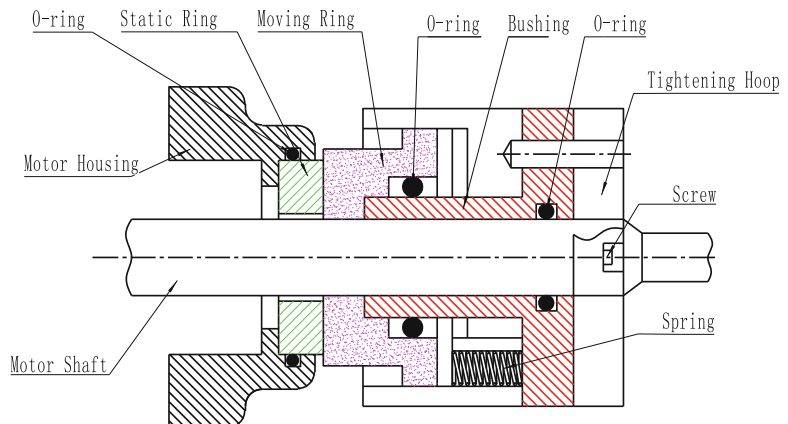
The essence of the mechanical dynamic seal is to convert the clearance between the shaft and the housing into the clearance between the static ring and the moving ring. The static ring is connected with the motor housing and remains stationary. The moving ring is connected with the motor shaft and rotates following with the motor shaft. Therefore, only the contact surface of the moving ring and the static ring has clearance, as shown in Fig. 11. However, the static ring is made of tungsten carbide, known as hard ring, the moving ring is made of impregnated resin graphite, known as

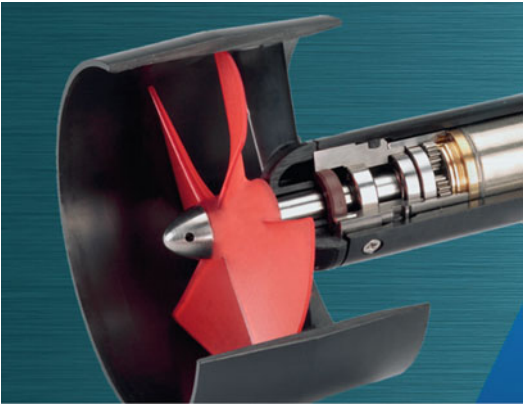
soft ring, and the two mutual contact surfaces are processed into a mirror, moving ring, and static ring tightly contact together by spring pressure. The moving ring, bushing, spring seat, and the tightening hoop are fixed into one. When the tightening hoop and the motor shaft are tightened, the moving ring can rotate with the motor in the same axis. Because the moving ring is soft ring easy to wear, when following the motor shaft rotation, there will form oil film between the moving ring and the soft ring, and the oil film fills the gap between the contact surface of the moving ring and the stationary ring, so as to prevent seawater from flowing through the contact surface. The thrusters of the 300 m underwater vehicle RECON-IV of Shenyang Institute of Automation used mechanical dynamic seal and achieved good application effect. At present, because of the complex structure, such structures cannot guarantee zero leakage, mechanical dynamic seal has been rarely used in underwater environments.

Underwater Rotating Oil Seal

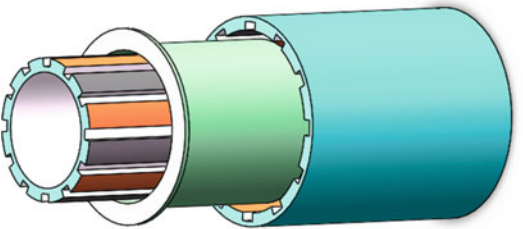
The underwater rotating oil seal can be used for the sealing of pressure compensated thrusters (Li 2005). The oil-filled pressure compensator always maintains motor housing pressure higher than the external seawater pressure of 0.2–0.7 bar, so the inside pressure of the oil seal is slightly higher than the outer side, thus achieving a good sealing effect, as shown in Fig. 12. This sealing method is simple and reliable but does not guarantee zero leakage. The 4500-m “Deep-Sea Warrior” HOV

AUV/ROV/HOV Propulsion System,
Fig. 11 Structure of mechanical dynamic seal





AUV/ROV/HOV Propulsion System, Fig. 12 Oil seal of underwater thruster

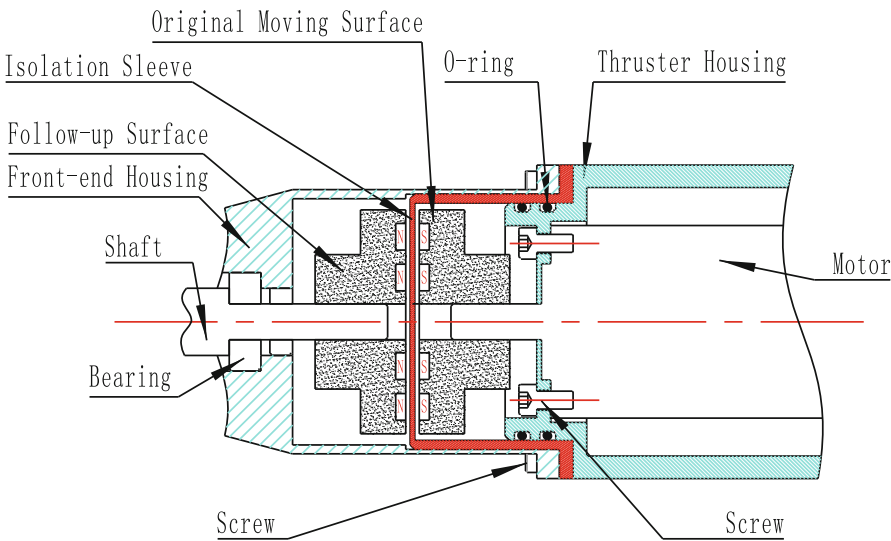


AUV/ROV/HOV Propulsion System, Fig. 13 Cylindrical magnetic coupling isolation seal

developed in China uses this sealing method and has completed more than 100 diving missions with good performance (Liu et al. 2016).

Magnetic Coupling Isolation Seal

The magnetic coupling isolation seal of underwater thruster converts the mechanical energy of the motor into magnetic energy, which is then converted into the mechanical energy of the propeller. According to the structural characteristics, it can be divided into cylindrical structure and planar structure. As shown in Fig. 13, the cylindrical magnetic coupling isolation seal is mainly composed of inner ring, outer ring, and isolating sleeve. The inner and outer rings are filled with high-remnance permanent magnets by U-shaped groove soft iron. The isolation sleeve is made of stainless steel or titanium alloy, and the magnetic permeability is good. The “Jiaolong” HOV thrusters use this sealing method. As shown in Fig. 14, the planar magnetic coupling isolation seal mainly consists of the original moving surface, isolation sleeve, and follow-up surface. Both the original moving surface and follow-up surface are inlaid with high remanence permanent magnets. Under the suction of the magnet, the follow-up surface rotates following the original moving surface and transmits torque. This seal has a large



AUV/ROV/HOV Propulsion System, Fig. 14 Surface magnetic coupling dynamic seal

axial force and is therefore only suitable for low-power thrusters.

Key Applications

Propulsion System of “Qian Long-III” AUV

As one of the representatives of the AUV, the propulsion system of “Qian Long-III” consist of five thrusters, including four Kort-nozzle main thrusters and one horizontal channel thruster, as shown in Fig. 15. These thrusters are designed as oil-filled compensation. The special pressure-resistant package design of the circuit components ensures that the drive circuit can work in a high-pressure environment, while the oil seal ensures that the motor extends out of the shaft and is reliably sealed. Under the control allocation system of thrust, “Qian Long-III” can get the maximum speed of 2.5 kN, and sail more than 30 h at the speed of 2.0 kN. In April 2018, in the South China Sea, the “Qian Long-III” sailed 156.82 km

and 42.8 h in a single dive and created a new record for China’s deep-sea AUV (Li et al. 2018).

Propulsion System of “Hai Long-III” ROV

As one of the representative of ROV, the propulsion system of the “Hai Long-III” HOV is composed of seven thrusters, including four horizontal thrusters and three vertical ones, as shown in Fig. 16. These thrusters are driven by hydraulic motors, and the hydraulic source of the hydraulic motors is provided by a main hydraulic pump station via a seven-way servo hydraulic control valve box. The hydraulic pump station can provide a flow rate of 232 l pm and pressure of 285 bar, the servo hydraulic control valve can adjust the flow rate of pipelines and then adjust the speed of the thrusters. Under the control allocation system of the seven-way servo hydraulic valve, the “Hai Long-III” can get maximum forward and backward speed of 3.2 kN, maximum lateral moving speed of 2.4 kN, and maximum vertical moving speed of 2.0 kN. At present, “Hai Long-III” has carried out several sea tests, and its hydraulic propulsion system works well.

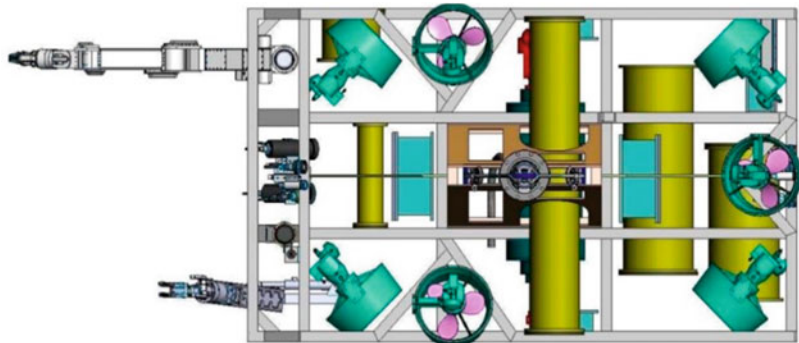


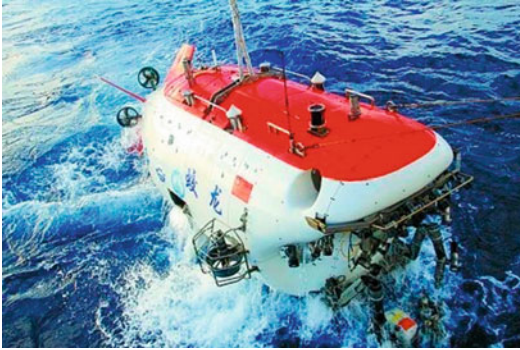
AUV/ROV/HOV Propulsion System, Fig. 15 The “Qian Long-III”

Propulsion System of “Jiaolong” HOV

As one of the representatives of HOV, the propulsion system of “Jiaolong” HOV is composed of seven thrusters, including four Kort-nozzle main thrusters in the tail, one horizontal channel thruster at the bow, and two rotary Kort-nozzle thrusters at the middle, as shown in Fig. 17. These thrusters use dry housing and wet housing isolation design, integrated built-in oil-filled compensator, magnetic coupling isolation seal drive, etc., to ensure reliable functioning for long time in the

AUV/ROV/HOV Propulsion System, Fig. 16 Structure of “Hai Long-III”





AUV/ROV/HOV Propulsion System, Fig. 17 “Jiaolong” HOV

high-pressure seawater environment (Cui 2013). Under the control allocation system of thrust, the “Jiaolong” HOV can get maximum forward speed of 2.5 kN, cruise speed of 1.0 kN, maximum transverse velocity of 0.7 kN, and maximum dynamic vertical transfer velocity of 0.7 kN. So far, “Jiaolong” HOV has made more than 150 dives, and its propulsion system is working well.

Cross-References

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Human Occupied Vehicle \(HOV\)](#)
- [Remote Operated Vehicle \(ROV\)](#)

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AUV/ROV/HOV Resistance

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Synonyms

[Appendage resistance](#); [Form resistance](#); [Frictional resistance](#); [Hull resistance](#); [Resistance](#); [Viscous resistance](#)

Definition

When the underwater vehicles such as AUVs/ROVs/HOVs or submersibles navigate in the

sea, they will inevitably suffer the resistance of the water as they move in the water fluid medium. Therefore, the reaction force of the water on the hull is called the resistance. The resistance of the submersible is usually divided into the resistance of the bare hull and the resistance of appendage (Renilson 2015). Appendage resistance refers to the increase in the resistance value of the appendage structure, such as rudders, stabilizers, manipulators, thrusters, etc., which overhung on the naked hull. From the mechanical point of view, the resistance of the submersible consists of the frictional resistance because of the tangential force along the surface of the hull and the pressure resistance because of normal force which is exerted on the surface of the hull. Unlike the ship hull, since the submersible does not have a free surface, the pressure resistance exists only in the composition of the viscous pressure resistance, and there is no wave-making resistance.

Scientific Fundamentals

Method for Studying the Submersible Resistance

Methods for studying the resistance of submersibles include theoretical analysis, experimental research, and numerical calculation (Sheng 2003). Theoretical analysis can only be applied to simple problems and qualitative analysis because of the simplification of the object. Experimental method has the characteristics of strong authenticity and high reliability relative to theoretical research; but due to the cost, project duration, and other constraints, experimental method is generally only used to validate the results of numerical methods. The towing tank test and wind tunnel test are the mostly used experimental methods currently. In recent years, with the rapid development of computer technology, computational fluid of dynamics (CFD) method in the hydrodynamic performance of submersibles are applied in more and more applications. Compared to traditional theoretical and experimental methods, CFD methods have the advantages of short time consumption and low cost.

Classification of Submersible Resistance

There are many appendage structures on the surface of the submersible, including the rudder, manipulator, thruster, underwater camera, underwater lighting, sampling basket, etc., so the submersible resistance can usually be divided into the resistance of the bare hull and the resistance of appendage according to the structure of the submersible.

From mechanical point of view, the main source of resistance of the submersible under water is the tangential force along the surface of the hull and the normal force which is perpendicular to the surface of the submersible, which are the frictional resistance and the viscous pressure resistance respectively.

According to the boundary layer theory, a boundary layer is formed on the surface when the viscous water flows through the submersible, and the rate of change of the fluid velocity in the boundary layer is prominent. According to the Newton internal friction law, the frictional shear stress on the surface of the submersible is great, so the resultant force of these frictional shear forces is the frictional resistance of the submersible. In scientific research, the frictional resistance can be thought of as the sum of the smooth plate frictional resistance with the same wetted surface, the same length and speed, and the additional frictional resistance after considering the tortuosity and roughness. In the theoretical study, scholars often use the formula to estimate the frictional resistance. After comparing with the experimental data, it can be proved that these formulas have appropriate precision within limits.

The viscous pressure resistance is another component of the resistance of the submersible, in essence, due to the viscosity of the fluid. It is known from the theory of fluid mechanics that a lot of vortices appear in the rear part of the hull when boundary-layer separation occurs, causing a difference in fore and aft pressure, which causes resistance. This resistance is called viscosity pressure resistance. Sometimes, it is also referred as vortex resistance. There are various methods for measuring the viscosity pressure resistance, and the most commonly used method is the wake pressure measurement method. In the actual

research, in order to avoid the tediousness of the experiments and to obtain the value of the viscosity pressure resistance, the Froude two-dimension method and the three-dimensional method which is proposed in the 1950s can be used.

Factors Affecting the Resistance of the Submersible

The hull form is the most important factor which is affecting the resistance performance of the submersible. The following two parameters are the most sensitive parts of the resistance performance of the submersible (Renilson 2015).

1. L/D : Aspect ratio. Where L is the submarine length, D is the diameter.
2. $C_p = \frac{\nabla}{A_M L}$: Prismatic coefficient. Where ∇ is the volume of the submarine and A_M is midships cross-sectional area.

In addition, the fore body form, the parallel middle body length and shape, and the aft body form all affect the resistance performance of the submersible.

Key Applications

Studying the resistance performance of the submersible is of great significance for improving the cruising speed for AUV/ROV/HOV.

Methods on Measuring Submersible Resistance

There are generally three methods to measure the drag of submersible: empirical equation estimation, model test method, and computational fluid dynamics (CFD) method. In different design phases of submersible, different methods are used.

Empirical Equation Estimation (Cui et al. 2018)

For submersibles moving underwater, if the depth exceeds one coxswain, the total resistance can be expressed as:

$$R_t = R_f + R_{pv} + R_{ap}$$

$$= \frac{1}{2} \rho V^2 (C_f + \Delta C_f + C_{pv} + C_{ap}) \quad (1)$$

In the formula, " R_t " is the total resistance.

" R_f " is the frictional resistance.

" R_{pv} " is the viscous pressure resistance.

" R_{ap} " is the appendage resistance.

" ρ " is the fluid density.

" V " is the submersible speed.

" C_f " is the frictional resistance coefficient.

" ΔC_f " is the roughness coefficient.

" C_{pv} " is the viscous pressure resistance coefficient.

" C_{ap} " is appendage resistance coefficient.

- (1) Empirical formula for calculating frictional resistance

The following three formulas can be used to calculate the friction resistance of submersible.

1. Schoenherr formula

$$\frac{0.242}{\sqrt{C_f}} = \lg(R_e C_f) \quad (2)$$

When $R_e > 10^6 - 10^9$, the formula is equal to:

$$C_f = \frac{0.4631}{(\lg R_e)^{2.6}} \quad (3)$$

2. ITTC formula

$$C_f = \frac{0.075}{(\lg R_e - 2)^2} \quad (4)$$

3. Prandtl-Schlichting formula

$$C_f = \frac{0.455}{(\lg R_e)^{2.58}} \quad (5)$$

- (2) Empirical formula for calculating viscous pressure resistance

$$C_{pv} = C_\phi \cdot K \cdot \frac{A}{S} \quad (6)$$

In the formula, $K = f(B/H)$.

" S " is the wetted surface area.

" C_ϕ " can be expressed as follow:

$$C_\phi = f\left(L_A / A^{\frac{1}{2}}, \phi\right) \quad (7)$$

“ L_A ” is the length of outlet section;

“ ϕ ” is the longitudinal prismatic coefficient, $\phi = \nabla/L \cdot A$.

“ A ” is the midship section area.

- (3) Empirical formula for calculating appendage resistance

The appendage resistance can be expressed as follows:

$$C_{ap} = \sum C_{scf} \cdot \frac{S_{SC}}{S} + K_{SC} \cdot C_{pv} \quad (8)$$

In the formula, “ C_{scf} ” is frictional resistance coefficient on control surface.

“ S_{SC} ” is the wetted surface area on control surfaces.

“ K_{SC} ” is the empirical coefficient.

“ C_{pv} ” is viscous pressure resistance coefficient on naked bodies.

The Towing Tank Test

The towing tank test is one of the most used experimental methods to obtain the resistance of underwater vehicles, ships, etc. Due to the limitation of facilities of any towing tanks, scaled-sized models are generally used instead of real size models. In the towing test, the scaled model is usually mounted on the trailer through the support rod, and the resistance of the model is measured in all directions through the movement of the trailer, which simulates the movement of

the submersible. The real body resistance can be converted by the model resistance through the similarity laws. Froude similarity theory is often used for resistance conversion in engineering experiments. A towing tank test of a full ocean depth submersible is shown in Fig. 1 (Jiang 2016).

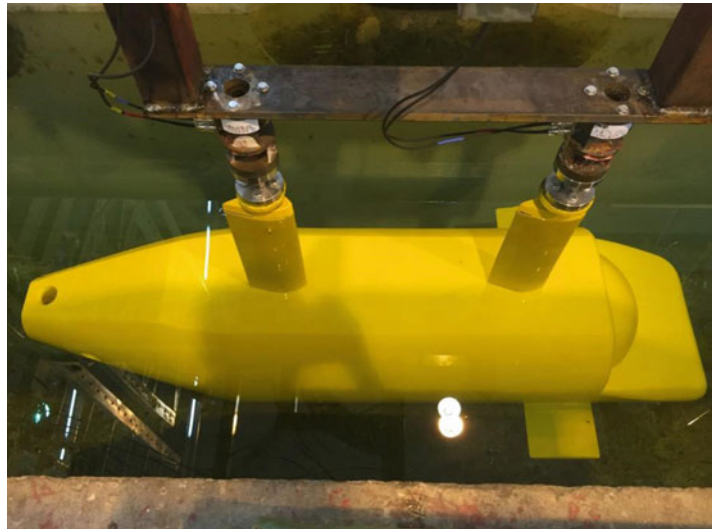
Resistance Analysis Using Computational Fluid Dynamics (CFD) Method

Empirical equation estimation has poor adaptability varying with different shapes and characteristics of different types of submersibles. Towing tank model test is costly and time consuming. Compared to experimental method, using CFD analysis in the resistance analysis will be much faster and require relatively low cost. In addition, in principle it is possible to use CFD to obtain results at full-scale Reynolds numbers, something which is not possible using model experiments. On the other hand, the analysis object can be modeled according to the shape or molded lines of it, and even appendages or local contour can be simulated in a CFD analysis; the results obtained through CFD analysis will be more accurate and reliable compared to using empirical equation estimation. Therefore, the CFD method is usually adopted to conduct flow field simulation, especially in the early stage of submersibles' design, and numerical calculation model is built in CFD software to obtain submersibles' resistance.

AUV/ROV/HOV

Resistance,

Fig. 1 Resistance model tests



Cross-References

- [AUV/ROV/HOV Propulsion System](#)
- [Towing Tank Test](#)

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AUV/ROV/HOV Stability

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Synonyms

[Intact stability](#); [Longitudinal stability](#); [Metacentric height](#); [Restoring moment](#); [Stability at large angles](#); [Stable equilibrium](#); [Transverse stability](#)

Definition

Stability is a kind of ability to enable an AUV/ROV/HOV to restore the original position after deviating from the equilibrium position under disturbing force. The distance, BG, between the center of gravity (G) and the center of buoyancy (B), governs the stability characteristics of the submerged submersible. For a surfaced boat, the measure of the stability is given by the distance GM (metacentric height). Thus, to be in stable equilibrium when the *surfaced boat* upright, the vertical position of the

center of gravity (G) should be sufficiently below that of the center of buoyancy (B) or metacenter (M).

Scientific Fundamentals

For the submerged AUV/ROV/HOV, only intact stability needs to be considered because there is no reserve of buoyancy. Hydrostatic stability, both transverse and longitudinal, i.e., in heel and pitch, is described by the position relationship between the center of gravity (G) and the center of buoyancy (B). The distance, BG, governs the stability characteristics of the submersible, which determines the restoring moment when it experiences an angle of heel or an angle of pitch.

The criteria for the magnitude of BG are in relation to several circumstances: (Burcher and Rydill 1994).

- (a) Static: It has sufficient BG that moving weights or men on board athwartships or fore and aft does not cause a large change of attitude.
- (b) Dynamic: The value of BG affects the behavior of the submersible when proceeding at speed submerged. If it changes course at speed, there is a tendency to cause a heeling moment and to result in a very large heel angle. During a turn, BG provides a hydrostatic moment to the hull resisting heeling and pitching motion. At very slow speeds, the hydrostatic moment results in the depth control surfaces no longer cause the boat to dive, but instead cause it to angle down by the bow while experiencing an upward rising motion.
- (c) Transition: The transverse metacenter moves from the center of buoyancy to the position of metacenter appropriate to the surface waterline in the transition from the submerged to the surfaced condition. The center of gravity also moves owing to the removal of water from the MBTs, which gives a free surface effect. It is necessary to ensure that during transition the migrating center of gravity is not above the metacenter.

Transverse Stability

For a submerged AUV/ROV/HOV to be stable in roll, known as transverse stability, the center of buoyancy (B) must be above the center of gravity (G), termed as positive BG value, as shown in Fig. 1a. In this case, if the boat is heeled to a small angle, as shown in Fig. 1b, the hydrostatic moment on it will cause it to return to the upright. On the other hand, if G is above B, at a small angle heeling under an external moment, the hydrostatic moment will cause it to continue to heel, as shown in Fig. 1c. (Martin Renilson 2015).

When the submersible is floating on the surface, the situation is different. In this case, the center of buoyancy (B) moves transversely as a function of heel angle. For small angles, the upward force, through the center of buoyancy (B), always acts through the metacenter, designated as “M” in (d). Thus, for a surfaced boat to be in stable equilibrium when upright, the position of metacenter (M) has to be above the center of gravity (G), and measure of the stability is given by the distance GM (metacentric height).

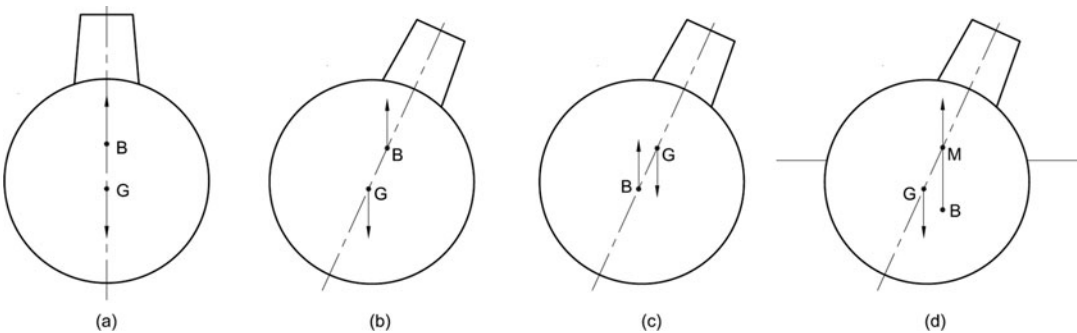
The formula of the vertical distance between the center of buoyancy (B) and the metacenter (M) is given as follows: $BM = I / \nabla$, where I is the second moment of area of the waterplane around the longitudinal axis and ∇ is the immersed volume. When the submarine is submerged, I will be equal to zero, and hence the position of metacenter (M) will be the same as the position of the buoyancy center (B).

Longitudinal Stability

Similar as transverse stability, the same principles apply to a submerged AUV/ROV/HOV as to a floating surface ship. However, the lack of a waterplane results in a very small restoring moment in the longitudinal direction if the submersible is in the pitch motion. Thus, it is essential to have the longitudinal position of the center of gravity (G) lined up with the longitudinal position of the center of buoyancy (B). As the longitudinal position of the center of gravity moves during a voyage due to use of consumables, sample collection, etc., it is necessary to be able to adjust this by use of ballast tanks or other suitable methods.

Stability at Large Angles

As well as designing an AUV/ROV/HOV to have positive surface flotation at a reasonably level trim, there is also the need to satisfy the heel stability criterion by having a positive GM (metacentric height). A submersible on the surface has very small reserves of buoyancy compared with surface ships. On the other hand, with its small profile, the surfaced submersible does not face as much in the way of rolling excitation from wind and waves as most surface ships. The configuration of a submersible makes it difficult to provide the boat with large heel stability. Motion on the surface is usually characterized by slow large angle rolling, sometimes to alarming angles, whereas, fully submerged, little motion occurs.



AUV/ROV/HOV Stability, Fig. 1 When the submersible is floating on the surface, the situation is different. In this case, the center of buoyancy (B) moves transversely as a function of heel angle. For small angles, the upward force, through the center of buoyancy (B), always acts through

the metacenter, designated as “M” in (d). Thus, for a surfaced boat to be in stable equilibrium when upright, the position of metacenter (M) has to be above the center of gravity (G), and measure of the stability is given by the distance GM (metacentric height)

Hence, the surface is not the place for a submarine in heavy weather.

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Key Applications

The stability characteristics of an AUV/ROV/HOV are governed by the distance of BG and GM for the submerged and surfaced condition, respectively. Therefore, enough attention must be paid to keep the vertical position of the center of gravity (G) sufficiently below that of the center of buoyancy (B) or metacenter (M), which leads to an adequate separation for the purposes of transverse and longitudinal hydrostatic stability.

Main Ballast Tanks (MBTs)

The purpose of the MBTs is to allow major adjustment of the submersible mass to enable it to operate submerged as well as on the water surface. Water and air enter and leave the MBTs through flooding holes at the bottom and vents at the top of the tanks. The size of the vents and the flooding holes will have a direct effect on how long it will take for the MBTs to fill, and small flooding holes may cause problems with overpressure and stability issues on the surface.

To improve the initial stability, the waterline areas should be as large as possible. The MBTs may be located at sides in transverse and at bow and stern in longitudinal, which gains the higher metacenter characteristic on the surface.

Compensating Tanks

Compensating tanks is usually composed of two tanks, one forward and another afterward. When the submersible submerged, adjustment ability of MSTs is lost, because it is fully flooded; compensating tanks can be used to correct the longitudinal balance by flooding or discharging of these two tanks separately, which hence effect on the stability of submersible.

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Auxiliary Icebreaking Methods

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Synonyms

[Assistant icebreaking systems](#); [Auxiliary icebreaking technologies](#)

Definition

Auxiliary icebreaking methods collectively refer to the application of auxiliary systems and/or technologies on the icebreaker to improve its icebreaking ability. They are usually used for the high-level icebreaker which works in high ice conditions. In contrast, conventional icebreaking method usually refers to that an icebreaker breaks ice by its own momentum and/or gravity, in either continuous or crash icebreaking ways (Ettema et al. 1987).

Scientific Fundamentals

As mentioned above, there are usually two traditional ways to break ice for an icebreaker: continuous or crash icebreaking ways. The former is to crush the ice by icebreaker's gravity, and the latter is to crash the ice by icebreaker's power mainly. As the thickness of ice increases, both methods become difficult, and the capacity of icebreaker will be limited. Under extreme conditions, when the ice thickness exceeds the icebreaking capacity of the icebreaker, the ship is possible to be blocked or trapped and even broken. At this time, auxiliary

icebreaking methods to improve the icebreaking capacity of the icebreaker become important, which will help the icebreaker get rid of trapped. Beyond that, fast and effective auxiliary icebreaking methods are also proposed to improve the performance of the ice breaker in normal working conditions. One can see various auxiliary icebreaking methods booming. To make it simple, we broadly classify auxiliary icebreaking methods into the following categories according to the working principles:

- Mechanical methods, which crush ice by accessory mechanical devices, such as mechanical ice cutting, Alex bow barge, mechanical saws, Archimedes screw vehicle, stem knives, bow ramp, mechanical impact devices, etc.
- Auxiliary ship movement methods, such as high-speed heeling system and pitching systems
- Auxiliary medium methods, which help to reduce ice resistance by adding auxiliary medium on the ship, such as water hull lubrication systems, bubbler systems, water jets, and low friction hull coatings
- Additional floating-body methods, which help to break ice by introducing additional floating bodies whose icebreaking principles can be various, such as explosive icebreaking barge, air cushion vehicles, upper Mississippi icebreaker, etc.

Mechanical Methods

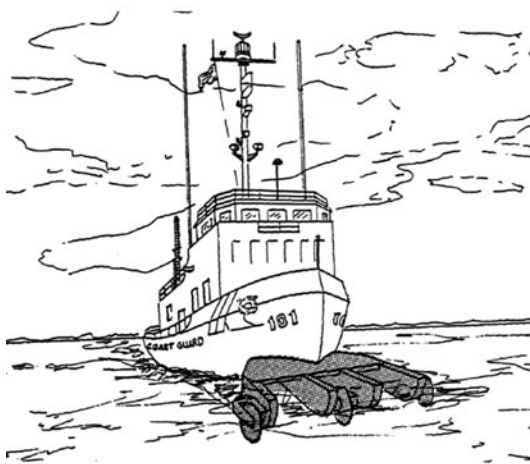
Just as its name implies, mechanical methods enhances the icebreaking capability of the icebreaker by installing additional mechanical devices. These accessory mechanical devices could help to break ice directly, but they need to be installed on the ship rather than work independently. As shown in Fig. 1, these devices can be various. Based on the installation positions, they can be further classified into three types roughly. One is devices installed around the bow, including saws, planning tools, plow-form devices, etc. as shown in Fig. 1a and b. This kind

is most common, as bow is the principal part during icebreaking. The second one is devices suspended from the hull to the ice surface by suspenders, such as pneumatic actuators and pile hammers, as shown in Fig. 1c. Others are placed in the middle of the hull, such as two large counterrotating screws, as shown in Fig. 1d.

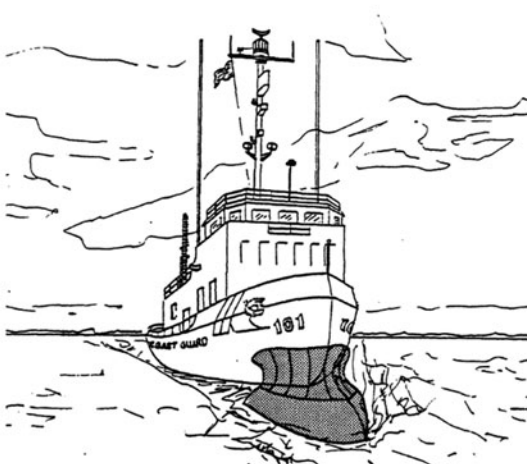
Under suitable conditions, mechanical devices can improve the efficiency of icebreaking and increase the safety of navigation in ice-covered areas. However, most of them are limited to research fields rather than actual engineering applications, because they change the shapes of the hull severely and induce a lot of problems when the ship moves in open water. Here we choose mechanical saws as the example to introduce its working principle and development process.

The mechanical saw is usually installed on the bow and has corresponding control devices to adjust its installation angle. Cut by the mechanical saw, the ice can be weakened in advance and even broken directly. The weakened ice or crushed ice will be further broken and pushed away by the hull. As a result, the channel is opened up. Various toothed blades, which can cut through the ice layer, can be adopted, such as direct saw, slitting saw, cross cut saw, chain saw, electric bow saw, etc. Stephens designed a semisubmersible oil tanker icebreaker with three steel ice saws mounted on its narrow superstructure, which could undercut the ice (Stephens 1973). In addition, some mechanical saws are equipped with thermal installations. The hot water discharged along the saw tooth reduces the blockage of the ice during icebreaking effectively. The Russians have extensively studied mechanical saws and report that model tests show devices using saws on channel clearing devices requiring 17–28% of the force to move a model of a typical icebreaker. However, there is no literature on full-scale devices in operation (Smith et al. 1981). Americans have also conducted extensive researches on mechanical saws and analyzed the application of such devices to meet the requirements for Coast Guard support in various domestic waterways (Lecourt and Voelker 1974). However, just mentioned above, this technology

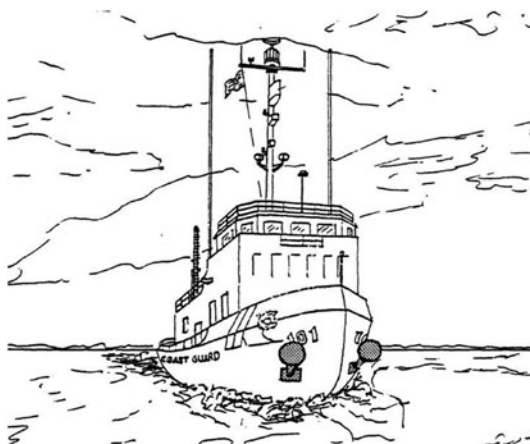
Mechanical methods



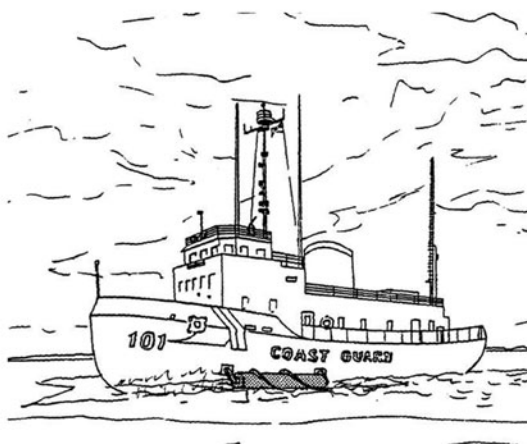
(a) Mechanical saws



(b) Alex bow barge



(c) Mechanical impact device



(d) Archimedes screw

Auxiliary Icebreaking Methods, Fig. 1 Sketches of some typical mechanical auxiliary icebreaking methods (Smith et al. 1981)

has been limited to the research field, and no use in actual engineering has been found until now.

Auxiliary Ship Movement Methods

Auxiliary ship movement methods aim to enhance the motion of the icebreaker in one or two degrees of freedom, usually the roll and pitch motions, in order to get rid of trapped rapidly. Correspondingly the high-speed rolling and pitching systems are developed. Sometimes

the former is also known as high-speed heeling systems. Both could also collectively be named as tilting technology (Harbron 1983). Most common methods to achieve this goal are to adjust the ballast water.

The method by adjusting ballast water to achieve pitch motion is widely used for the continuous icebreaking mode of icebreaker. The bow of the icebreaker is designed into a special shape, and ballast tanks are equipped at the stem and stern, respectively. Ballast water is firstly adjusted to stern so that the icebreaker trims by

stern and the bow rides on the ice. Usually the weight of ship bow can crush ice. If not, ballast water is adjusted to stem to add the weight of ship on the ice until the ice is crushed.

High-speed heeling system has similar working principle to pitching system. The only different thing may be that the former is usually adopted to prevent the hull from being trapped or frozen, while the latter for continuous icebreaking mode. The working principle of high-speed heeling system is plotted in Fig. 2, where two water tanks are arranged on both sides of the hull. Ballast water loads are continuously transferred between both sides of the tanks, so the hull swings back and forth continuously as a result. In this way, the ship can get rid of the frictional obstacles of the ice on both sides of the hull, thus getting out of trouble. The high-speed heeling system, also known as the active shake system, can be seen as a kind of special anti-heeling system. Different from the general ballast water transfer system, the purpose of this system is to be unbalanced in a faster speed. Therefore, compared to the general system, the pump flow is much larger and the piping and valves are bigger. There are many ballast water transfer technologies, including gravity type, water pump-driven type, and pneumatic type. Each has its own advantages and disadvantages, and they are often used in combination. In the design stage, one needs to consider comprehensively and selects one or several combinations according to the actual demand, thus to achieve the best effect of ballast water adjustment.

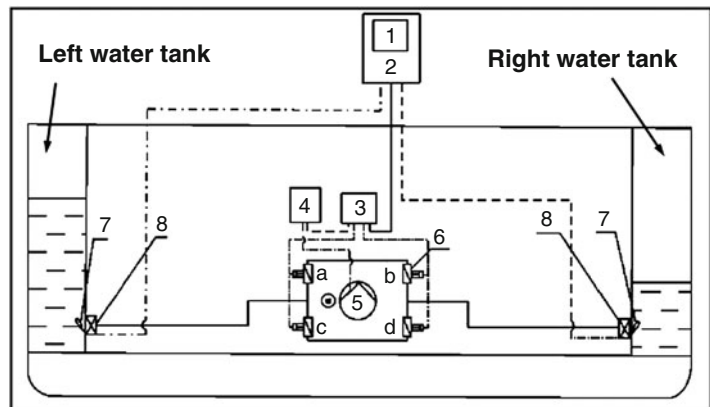
Auxiliary Medium Methods

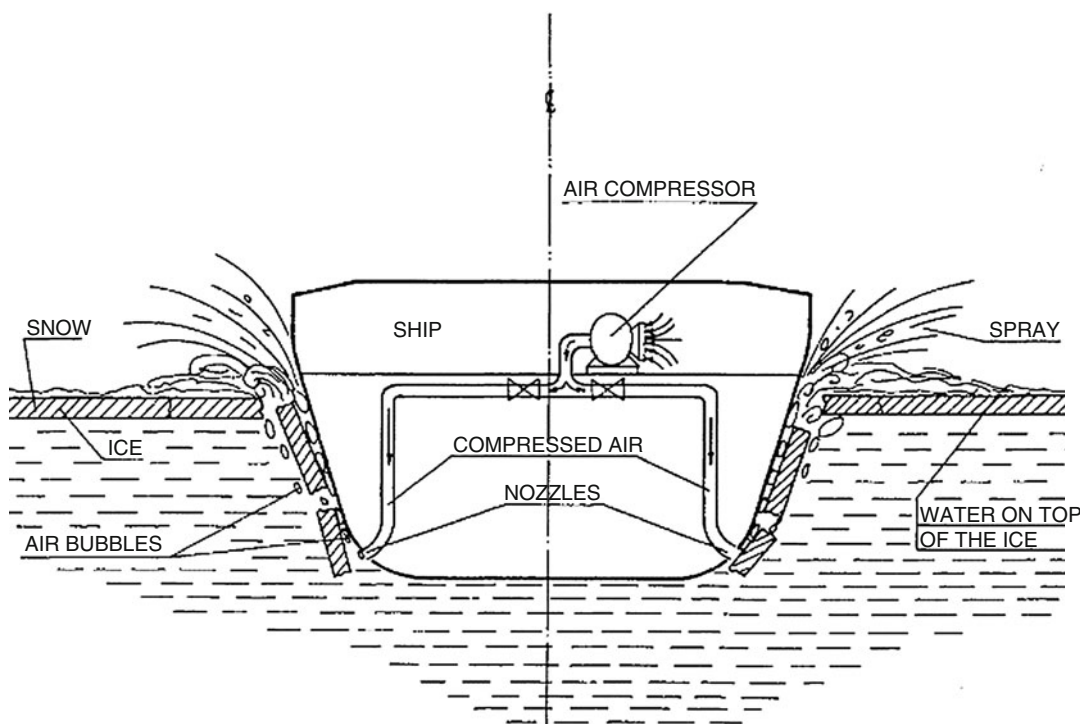
The auxiliary medium icebreaking method is to decrease the resistance between the hull and the ice by adding extra substances between them. In this way the icebreaker can break ice more efficiently. The media can be low friction coatings, which reduces the friction between the hull and the ice directly. The most successful low friction coatings until now may be unsolvated polyurethane and epoxy (Calabrses et al. 1976; Alliston 1985). On the other hand, the media can also be high-pressure water flow, bubbles, and/or steam water mixture, which is the auxiliary medium method we usually refer to. High-pressure compressed water, gas, or water vapor mixture is injected through a number of nozzles arranged on the hull below the waterline. In this way, a low-resistant film is formed between the hull and the ice, which reduces the friction force on the hull. Although the medium differs, the principles are similar. Here we take bubbler system as the example.

Because of the low density of air, bubble drag reduction on ships in open water has been proposed and studied for a long time (Madavan 1985). Similarly, this idea is also used and extended for icebreaker. One of the most representative systems was invented by Wartsila in 1967 and was widely applied to icebreakers in the 1970s, which was called Wartsila Air Bubbling System (WABS) (Parksvangen et al. 1971; Segercrantz 1989), as

Auxiliary Icebreaking

Methods, Fig. 2 The principle of the high-speed heeling system (Yang and Ma 2010)





Auxiliary Icebreaking Methods, Fig. 3 The principle of the air bubble system (Segercrantz 1989)

shown in Fig. 3. The main principle of the system is that the compressed air is ejected from the hole on the lower part of the hull. The rising air bubbles push the broken ice away, thus reducing the friction between the hull and the ice and improving the icebreaking performance of the ship. The icebreaking performance of the ship equipped with the bubble-assisted icebreaking system was studied by using both the scale model and the prototype icebreaker. The results showed that the bubble-assisted icebreaking system could save the power of the icebreaker in the ice-covered zone efficiently and the maximum energy-saving efficiency could reach about 30% (Major 1977; Goodwin 1980; Vance 1980). This technology was also applied to the nuclear-powered heavy icebreaker 50 Let Pobedy, as shown in Fig. 4, which was commissioned in Russia in 2007. The bottom of the ship was equipped with a bubble-assisted icebreaking system that produced $24 \text{ m}^3/\text{s}$ of air bubbles in a region of 9 m deep underwater to assist in icebreaking operations (Zhang et al. 2017).

Additional Floating-Body Methods

Compared with the methods above, additional floating-body technologies can be quite complex. It usually has two distinct characteristics: one is that an auxiliary equipment needs to be installed on a floating body outside the hull and the other is the additional floating body has independent function relative to the main hull. For example, the explosive icebreaking method requires a barge to be equipped with an explosive tank with a mixture of compressed air and hydrocarbon fuel. The barge will be placed under the ice. When the mixture explodes, the open-cell structure on the box will instantaneously release the high-pressure gas and destroy the ice above, thus improving the icebreaking efficiency.

Another typical example is air cushion icebreaking platform or vehicle, which is not limited by draft and can break ice in both deep and shallow waters. At present, countries such as the United States, Canada, Russia, and Finland have



Auxiliary Icebreaking Methods, Fig. 4 Nuclear-powered icebreaker: 50 Let Pobedy (from https://en.wikipedia.org/wiki/50_Let_Pobedy#/media/File:50letPob_pole.JPG)

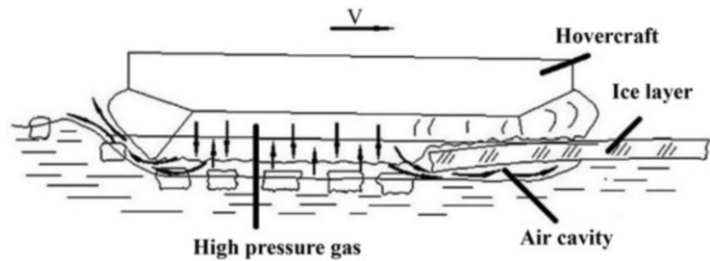


Auxiliary Icebreaking Methods, Fig. 5 Icebreaking ship of the Canadian Coast Guard: CCGS Sipu Muin (from <https://inter-j01.dfo-mpo.gc.ca/fdat/vessels/98>)

used air cushion platforms and hovercrafts in icebreaking engineering applications, and the icebreaking effect is satisfactory (Hinchey and Colbourne 1995; Whitten et al. 1986). Figure 5 provides the picture of an air cushion icebreaking

vehicle CCGS Sipu Muin of Canada. The icebreaking principles of hovercraft can be divided into low-speed mode and high-speed mode. In low-speed operation conditions, the air cushion pressure causes the water surface below

Auxiliary Icebreaking Methods, Fig. 6 Sketch of air cushion vehicle icebreaking



the ice layer to depress, as shown in Fig. 6. The simplified force model of the ice layer can be seen as a cantilever beam model as a result. When the stress of the ice layer is larger than its allowable stress, the ice will be fractured (Muller 1979; Mellor 1980). In the case of high speed, the hovercraft relies on the wave making on the ice surface to break ice. When the hovercraft moves at a critical speed, the energy of the air cushion vessel exerted to the ice cannot be radiated out. Therefore, under the continuous action of the air cushion vessel, the energy accumulates incessantly, and the amplitude of the fluctuating deformation of the ice layer increases, which eventually leads to the fracture of the ice layer due to the internal bending stress over its yield limit (Buck and Pritchett 1978; Buck et al. 1978).

In conclusion, auxiliary icebreaking methods or technologies have been reviewed and classified roughly. Here we need to clarify that auxiliary icebreaking technology is developing rapidly, so the classification will be advanced with time.

Cross-References

- [Ice Breaking Vessel](#)
- [Ice Management in Offshore Operations](#)
- [Special Marine Vehicle](#)

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Auxiliary Icebreaking Technologies

- [Auxiliary Icebreaking Methods](#)

Auxiliary Navigation

- [Integrated Navigation](#)

Auxiliary Propulsions

- [New Technologies in Auxiliary Propulsions](#)

Awareness

- [Human Factors in the Role of Energy Efficiency](#)

B

Bare Wire

- [Cable](#)

Barges

- [Jack-Up Platforms](#)

BATS (Broadband Acoustic Tracking System)

- [Ultra-short Baseline Underwater Acoustic Location Technology](#)

Beaching

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Definition

Beaching, as a shipbreaking method, generally refers to dismantling ships at grounded condition in intertidal zones. In this method, a ship is

emptied of cargo and ballast, then driven to coast on a high magnitude tide, and stranded on the beach. The workers access the vessel at low tide and start scrapping for recycling materials.

Scientific Fundamentals

History

Until the 1960s, shipbreaking activities mainly took place in industrialized countries and were conducted off the beach (FIDH 2002). The origin of the beaching can be traced to a Greek ship stranded accidentally on the beach of Sitakunda, Chittagong, during a cyclone in 1960. The vessel could not be refloated and so remained there for several years. In 1965, in East Pakistan, the then Chittagong Steel House bought the ship and had it scrapped on beach. From the early 1980s, the global shipbreaking activities started migrating to South Asian to increase profit (FIDH 2002). The industry developed rapidly in India, Bangladesh, and Pakistan where dismantling was exclusively accomplished on beaches (Yujuico 2014). In the recent years, beaching has become the most commonly used method for shipbreaking worldwide. About two-thirds of the end-of-life ships around the globe are dismantled by beaching each year.

The Layout of the Yards

Taking the beach of Chittagong shipbreaking yard as an example, the beach is generally divided up into

several plots about 60 m wide and up to 90 to 160 m deep. Infrastructure is minimal. It may consist of office building, few heavy-duty winches, vehicle-mounted cranes, and some walls between plots if preferred. Some temporary storage facilities and areas on concrete or steel bases usually exist. More sophisticated operations may have extensive offices, storage areas, and concrete areas within the facility. And even large separate secondary operations areas are on the shore behind the facility, which might include asbestos treatment areas, materials handling areas, medical facilities, shower room, labor shelter, and messing. Moreover, they have their own fire service and hospitals as well as a large and useful training facility on the hill overlooking the beach. The surroundings are typically made up of worker's houses as well as shops reselling the items that come from the ship.

The Process of Beaching

The following passages explain the typical process of beaching in Bangladesh (Hossain and Islam 2006). The process in India and Pakistan is about the same (Deshpande et al. 2013; Ahmed and Siddiqui 2013).

1. Firstly, the ship is anchored in international waters off the coast. The ship will then be inspected, checked, and made gas-free. After the administrative formalities are completed, the ship will obtain a permission to enter territorial waters for beaching. The ship has to wait until the tide condition is ideal for performing beaching. Usually no pre-cleaning of the ships is required for entering the shipbreaking area.
2. When the tide is high, the ship will be driven to the beach by own propulsion power. Usually workers would set up a big colored flag for guidance. After the tide drops, the ship will be lying stable on its flat bottom. The ship must be placed in this exact position and come to ground as high up on the beach as possible to facilitate dismantling operations. If the ships don't make it to the beach and are stranded on the mudflats, they are pulled higher with chains or heavy steel wire hawsers at the next suitable tide after being made lighter. Beaching has a crucial impact on the final cost. The time needed for dismantling can be doubled, if the beaching operation is not successful.
3. After the ship is grounded, the workers start removing items and cutting the vessel into large parts using blowtorches. The parts will drop down to the mudflats and be progressively dragged to the dry part of the shore by winches. Though the machinery does the main job, a large number of workers are engaged to help dragging the parts.
4. Another group of cutters, helpers, and workers start cutting the dragged parts of the ship into trackable parts as per order of the purchasers. The steel is often cut up into around 2×4 m pieces (usual practice) and then sold for cold rolling. Although it may also be cut into inch thick square bars using hydraulic shears and used directly for reenforcing concrete or similar materials, those plates, frames, bars, girder, stiffener, longitudinal, etc. are also used in local ship building or repair industry.
5. The unskilled workers carry metal plates, metal bars, or pipes on their heads or shoulders, then pile up metal plates in stack yards, or load them on trucks. The supervisors control the group of workers; the onlooker guides them and helps them in piling up the heavy metal plates in stacks.
6. Heavy equipment like boilers, motors, capstan stocking, etc. are carried to stack yards by moving crane. The valuable components (e.g., small motors and pumps, generator, navigation equipment, life-saving equipment, furniture, electrical cables, utensils, etc.) are dismantled and sold to secondhand market. It needs 5–6 months to dismantle a typical cargo ship.

Hazards of Beaching

End-of-life ships contain a great amount of hazardous materials. Depending on the size and function of the ship, scrapped ships have an unladen weight of between 5000 and 40,000 tons (the average being 13,000+), 95% of which is steel, coated with between 10 and 100 tons of paint

containing lead, cadmium, organotins, arsenic, zinc, and chromium. Ships also contain a wide range of other hazardous wastes, sealants containing PCBs, up to 7.5 tons of various types of asbestos, and thousands of liters of oil (engine oil, bilge oil, hydraulic and lubricant oils, and grease). Tankers additionally hold up to 1000 cubic meters of residual oil. Most of these materials have been defined as hazardous waste under the Basel Convention (Hossain and Islam 2006).

The breaking process would release those materials, threatening the health of the workers and the ambient environment. The problem is more serious when the ships are dismantled on beach other than in the dock, because hardly any protection measures will be taken.

Impact on Intertidal Sediments and Soils

In shipbreaking areas, various refuse and disposable materials are dropped or stacked haphazardly on the seashore. The hazardous materials are accumulated and get mixed in the soil. Those materials are mainly metal fragments and rust (particularly iron). It accelerates the amount of shore erosion and increases the turbidity of seawater and sediments in the area (Hossain and Islam 2006). Trace metals can be found in the sediments. Accumulation levels of Cr, Zn, As, Pd, and Cd are higher than recommended values of unpolluted sediments (Hasan et al. 2013a). There are on average 81 mg of small plastic fragments per kg of sediment. They do not degrade easily and hence pose a threat to the environment and marine ecosystems (Reddy et al. 2006).

Impact on Seawater

Ship scrapping activities pollute the seawater environment in the coastal area. Seawater is strongly polluted by Fe, Pd, and Hg, moderately by Mn and Al, and slightly by Pb and Cd. Groundwater is also polluted, strongly by Fe, Pb, and Hg, moderately by Mn and Al, and slightly by As (Hasan et al. 2013b). Toxic concentration of ammonia and marine organisms found in seawater had an increase in PH levels. Critical concentration of dissolved oxygen (DO) and higher biochemical oxygen demand

(BOD) were found with an abundance of floatable materials (grease balls and oil films) in the seawater (Hossain and Islam 2006).

Impact on Biodiversity

Shipbreaking activities contaminate the coastal soil and seawater environment mainly through the discharge of ammonia, burned oil spillage, floatable grease balls, metal rust (iron), and various other disposable refuse materials together with high turbidity of seawater. The high PH of the seawater and soil observed may be due to the addition of ammonia, oils, and lubricants. High turbidity of water can cause a decrease in the concentration of DO and substantially increase the BOD. Furthermore, oil spilling may cause serious damage by reduction of light intensity, inhibiting the exchange of oxygen and carbon dioxide across the air-seawater interface, and by acute toxicity. As a result, the growth and abundance of marine organisms especially plankton and fishes may seriously be affected. Indiscriminate expansion of shipbreaking activities poses a real threat to the coastal intertidal zone and its habitat (Hossain and Islam 2006).

Worker Rights Violation

In the majority of the shipyards in developing countries, workers are being deprived of their rights. They work under risky conditions but have no access to safety equipment, job security, or a living wage.

Shipbreaking carries a very real risk to life. By any standards, the demolition of ships is a dirty and dangerous occupation. The hazards linked to shipbreaking broadly fall into two categories: intoxication by dangerous substances and accidents on the plots. Explosions of leftover gas and fumes in the tanks are the prime cause of accidents in the yards. Another major cause of accidents is workers falling from the ships (which are up to 70 m high) as they are working with no safety harness. Other sources of accidents include workers being crushed by falling steel beams and plates and electric shocks.

Workers are not aware of hazards to which they are exposed. The overwhelming majority of

workers wear no protective gear, and many of them work barefoot. There is hardly any testing system for the use of cranes, lifting machinery, or a motorized pulley. The yards reuse ropes and chains recovered from the broken ships without testing and examining their strength. There is no marking system of loading capacity of the chains of cranes and other lifting machineries. Gas cutters and their helpers cut steel plates almost around the clock without eye protection. This leaves their eyes vulnerable to effects of welding. They do not wear a uniform and most don't have access to gloves and boots. Those that are "unskilled" carry truckable pieces of iron sheets on their shoulders, and there are no weight limits to the sheets they carry. Usually, these workers carry weights far above the limit prescribed in the Factories Act and Factories Rules. Besides, the beaches are strewn with chemicals and toxic substances, small pieces of pointed and sharp iron splinters causing injuries. Workers enter into the areas without wearing or using any protective equipment.

When there is an injury, some immediate treatment may be given, but there is no long-term treatment for those who have a long-term or permanent injury. In terms of compensation, only a nominal amount of compensation given and often only when there is public pressure. When a worker becomes disabled by a major accident, he gets a maximum of 10 to 15 thousand taka (1 USD = 71 taka) and forced back to his home district. In most cases a worker will only get transportation costs to go back to their home district. When a worker killed in an accident, the contractor, who is responsible for the workers, will only pay the costs of sending the body back to the victim's family and arranging for their burial. So far, the treatment and compensation problems of workers have not been well resolved.

Applications

Beaching activities mainly took place in India, Bangladesh, and Pakistan. Table 1 shows the

Beaching, Table 1 The number of ships dismantled by beaching in 2015–2017

Country	2015	2016	2017
India	194 (25%)	305 (35.4%)	239 (28.6%)
Bangladesh	194 (25%)	222 (25.8%)	197 (23.6%)
Pakistan	81 (11%)	141 (16.4%)	107 (12.8%)
Ships dismantled by beaching worldwide	469 (61.1%)	668 (77.5%)	543 (65%)
Ships dismantled worldwide	768	862	835

application condition of beaching in 2015–2017 (NGO Shipbreaking Platform 2016, 2017, 2018).

The main reasons for the phenomenon including (FIDH 2002):

1. Shipbreaking industry requires cheap human man power, which is available in developing countries with large population, such as India, Bangladesh, and Pakistan. In these nations, millions of people are in the state of poverty, providing abundant cheap labor.
2. The coastal areas of the shipbreaking yards have wide and gently sloping sand tidal beaches, making it possible to drive ships onshore as far up the beach as possible at spring tides and dismantle the ships at grounded condition.
3. In most countries, beaching had already been prohibited for environmental protection and occupational health and safety concerns. However, regulations and standards are loose in India, Bangladesh (Alam and Faruque 2014), and Pakistan compared to other ship recycling nations.
4. The convenient location of the Indian subcontinent, the suitable climate, and domestic market for scrapped steel are also factors that promote the prosperity.

The shipbreaking yards that utilize beaching method concentrated in the following regions:

1. The Alang-Sosiya beach is in the State of Gujarat on the west coast of India. This region has one of the longest continental shelves and the second highest tide port in the world, making it an ideal site for beaching (Patel et al. 2013). The Alang-Sosiya shipbreaking yard is considered the world's largest of its kind. According to statistics, in 2012, there were 171 yards in the region (92 in Alang, 79 in Sosiya) with about 60,000 workers, handling around 350 ships yearly (Deshpande et al. 2012).
2. The shipbreaking yards in Chittagong stretch about 20 km along the coast of Sitakunda Thana region (Gregson et al. 2012), a well-known shipbreaking area which is about 20 km northwest of Chittagong. Chittagong shipbreaking yard was the largest shipbreaking yard in the world from 2004 to 2008. Today, it takes the second place in the industry.
3. Gadani shipbreaking yard located across a 10-km-long beach at Gadani, Pakistan, about 50 km northwest of Karachi. In the 1980s, Gadani was the largest shipbreaking yard in the world with more than 30,000 direct employees. Gadani ranks as the world's third largest shipbreaking yard after Alang-Sosiya and Chittagong in terms of volume.

Cross-References

► Ship Recycling

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Bearing Capacity of Spudcans

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Synonyms

Footing; Foundation

Definition

Spudcan is the term used for the base cones on mobile-drilling jack-up platform. The spudcans are the inverted cones mounted at the base of the jack-up which provide support for the stability of jack-up rig when deployed into ocean-bed systems.

Bearing Capacity of Spudcans, Fig. 1 A typical jack-up unit (<https://cn.bing.com/images/search?q=jack-up&FORM=HDRSC2>)

Jack-up units are self-elevating mobile platforms which are used extensively in offshore oil and gas industry.

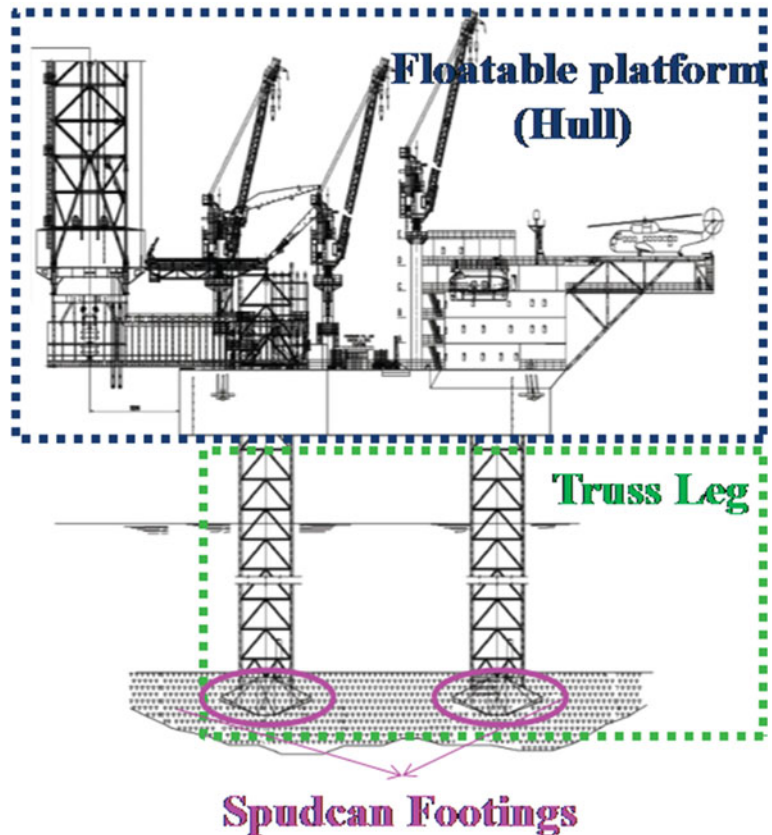
Bearing capacity is the ability of soil to safely carry the pressure placed on the soil from any engineered structure without undergoing a shear failure with accompanying large settlements (Lambe and Whitman 2008).

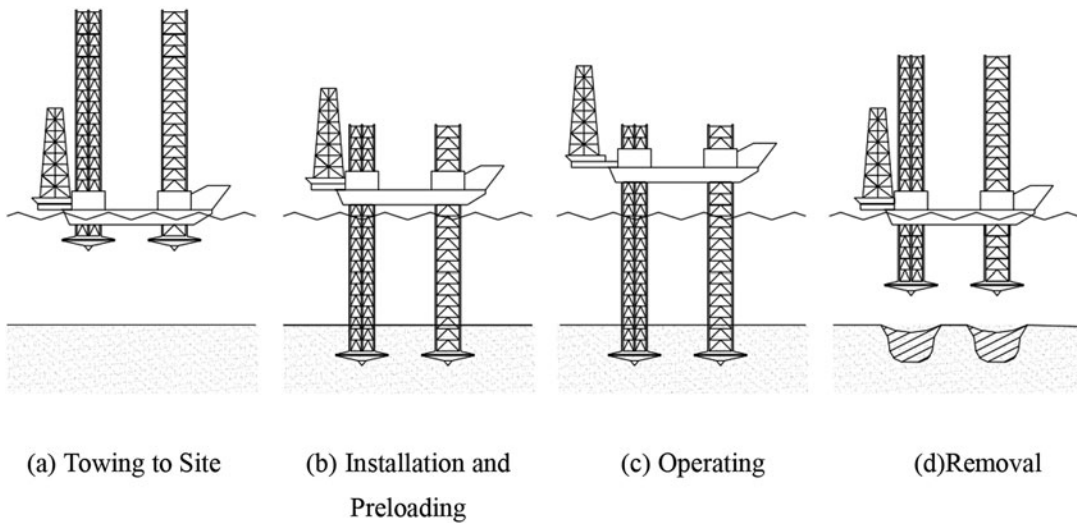
Scientific Fundamentals

Jack-Up Unit

A typical jack-up unit (as shown in Fig. 1) consists of a floatable platform (hull) and three independent retractable truss-work legs each resting on a spudcan footing.

Figure 2 shows the operation sequence of a jack-up unit. Once a jack-up unit is towed to site, its installation begins by lowering the legs to the seabed and pushing the spudcans into the





Bearing Capacity of Spudcans, Fig. 2 The operation sequence of a jack-up unit (Kong 2011) (a) Towing to site (b) Installation and preloading (c) Operating (d) Removal

soil and then rising the hull over the water. Then preloading can be achieved by pumping water into the hull. The preloading makes the spudcan penetrate deeper to provide more resistances. After preloading, the water is pumped out and the spudcan bearing capacity has some reservation. After all the work of the jack-up has finished, it is removed from the site by retracting the legs from the seabed (Kong 2011).

The Typical Geometry of Spudcans

The spudcans are of inverted cone shape commonly in polygonal or circular shape in plane. In the early days, the diameter of spudcans was within 5–10 m (Young et al. 1984). The shape and size of spudcans evolved for the operation of jack-up unit in deeper waters, and modern spudcans are typically between 10 and 20 m in diameter. Along with the increasing demands of operating on very soft soils, the footings become larger and flatter at base, such as HYSY944 (Hai-bo 2016) (as shown in Fig. 3c). Typical spudcan designs are illustrated in Fig. 3.

Alternative Foundations

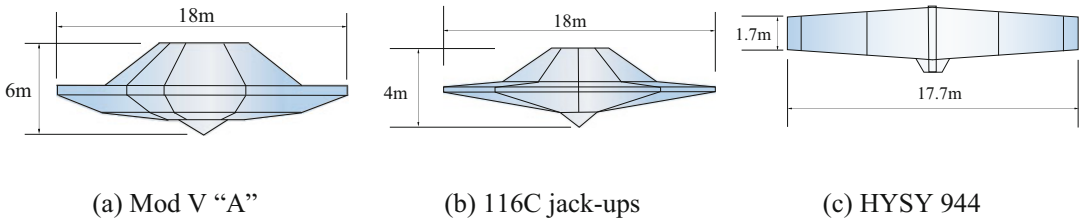
In some complex environment conditions, such as penetrating on layered soil, especially on the sand overlying clay soil, the use of fusiform spudcan is

limited for its high risk of punch-through failure. To mitigate the punch-through failure potential of spudcan foundations for jack-up rigs on layered soils, skirted footings (Fig. 4a) are increasingly being considered (Yu et al. 2010). The skirted foundation is a flat spudcan foundation fitted with a thin skirt at the circumference. It appears like an upturned bucket.

The whole drilling processes of installation and extraction of the spudcan will leave a permanent seabed depression at each footing site, which is referred to as a footprint. With the extensive and more frequent use of jack-up units, the existence of footprints is common. Now jack-up units often have to be installed near or into existing footprints in order to drill, and it can lead to significant loss of time, cost implications, risks to adjacent structures, and potential injury to personnel. Jun et al. (2016) present a novel spudcan with six evenly spaced holes at the base to ease spudcan-footprint interaction issues as shown in Fig. 4b.

The Bearing Capacity of Shallow Footing

When a load is applied on a limited portion of the surface of a soil, the surface settles. The relation between the settlement and the average load per unit of area may be represented by a settlement curve (Fig. 5). If the soil is fairly dense or stiff, the



Bearing Capacity of Spudcans, Fig. 3 Three typical spudcans (a) Mod V "A" (b) 116C jack-ups (c) HYSY 944

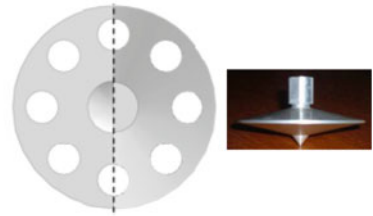
Bearing Capacity of Spudcans,

Fig. 4 Alternative foundations. (a) Skirted footing model (Yu et al. 2010) (b) A novel spudcan (Jun et al. 2016; Yu et al. 2010)



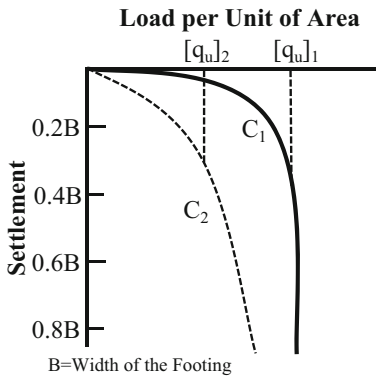
Skirted footing model

(Yu et al., 2010)



A novel spudcan

(Jun et al., 2016 & Yu et al., 2010)



Bearing Capacity of Spudcans, Fig. 5 Relation between intensity of load and settlement of a footing on C_1 dense or stiff and C_2 loose or soft soil (Terzaghi et al. 1995)

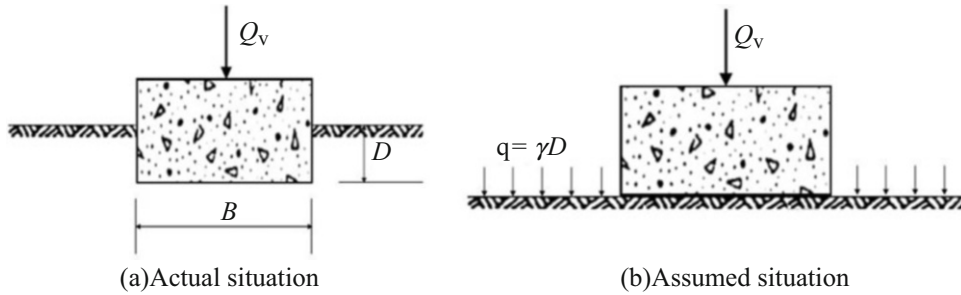
settlement curve is similar to curve C_1 . The abscissa of the vertical tangent to the curve represents the bearing capacity of the soil. If the soil is loose or fairly soft, the settlement curve may be similar to C_2 , and the bearing capacity is not always well defined. The bearing capacity of

such soils is commonly assumed to be equal to the abscissa of the point at which the settlement curve becomes steep and straight.

In practice, loads are transmitted to the soil by means of footings, as shown in Fig. 6. The footings may be continuous, having a long rectangular shape, or they may be spread footings, which are usually square or circular. The distance from the level of the ground surface to the base of the footing is known as the depth of foundation D . A footing that has a width B equal to or greater than D is considered a shallow footing.

For purpose of analysis, the actual situation shown in Fig. 6a is usually replaced by the situation shown in Fig. 6b: the soil above the base of the footing is replaced by a uniform surcharge of intensity $q = \gamma D$, where γ is the unit weight of the soil and D is the depth of the base of the footing below ground surface. The effect of the weight of the soil above the footing base is thus taken into consideration, and the shear resistance of the soil is neglected.

The approximate value of the bearing capacity can be written as the sum of the three terms: the



Bearing Capacity of Spudcans, Fig. 6 Shallow footing under a vertical load (Terzaghi et al. 1995) (a) Actual situation (b) Assumed situation

unit weight of the soil, the surcharge, and the cohesion of soil (only for clay soil) (Terzaghi et al. 1995):

$$q_u = cN_c s_c d_c + qN_q s_q d_q + \frac{1}{2} \gamma B N_\gamma s_\gamma d_\gamma$$

In which N_c and N_q are the bearing capacity factors with respect to cohesion and surcharge, respectively. The surcharge is represented by the weight per unit area γD of the soil surrounding the footing. The bearing capacity factor N_γ accounts for the influence of the weight of the soil. All the bearing capacity factors are dimensionless quantities depending only on ϕ' . s_c, s_q, s_γ are the correction factors corresponding to the footing shape. d_c, d_q, d_γ are the correction factors corresponding to the embedment depth of footing. B is the diameter of a circular footing or the width of a strip footing.

The Analysis Model of the Bearing Capacity of a Spudcan

For conventional bearing capacity analyses, the spudcan is often modeled as a flat circular footing. The equivalent diameter is determined from the area of the actual spudcan cross section in contact with the seabed surface, or where the spudcan is fully embedded, from the largest cross-sectional area. Bearing capacity analyses are then performed for this circular foundation at the depth (D) of the maximum cross-sectional area in contact with the soil (Fig. 7). The depth of simplified spudcan penetration is usually defined as the distance from the spudcan bottom to the mudline (SNAME 2008).

The Soil Parameters

Where geotechnical analyses are performed, they should be based on geotechnical data obtained from a site investigation incorporating soil sampling and/or in situ testing. Uncertainties regarding the geotechnical data should be properly reflected in the interpretation and reporting of analyses for which the data are used.

Key Applications

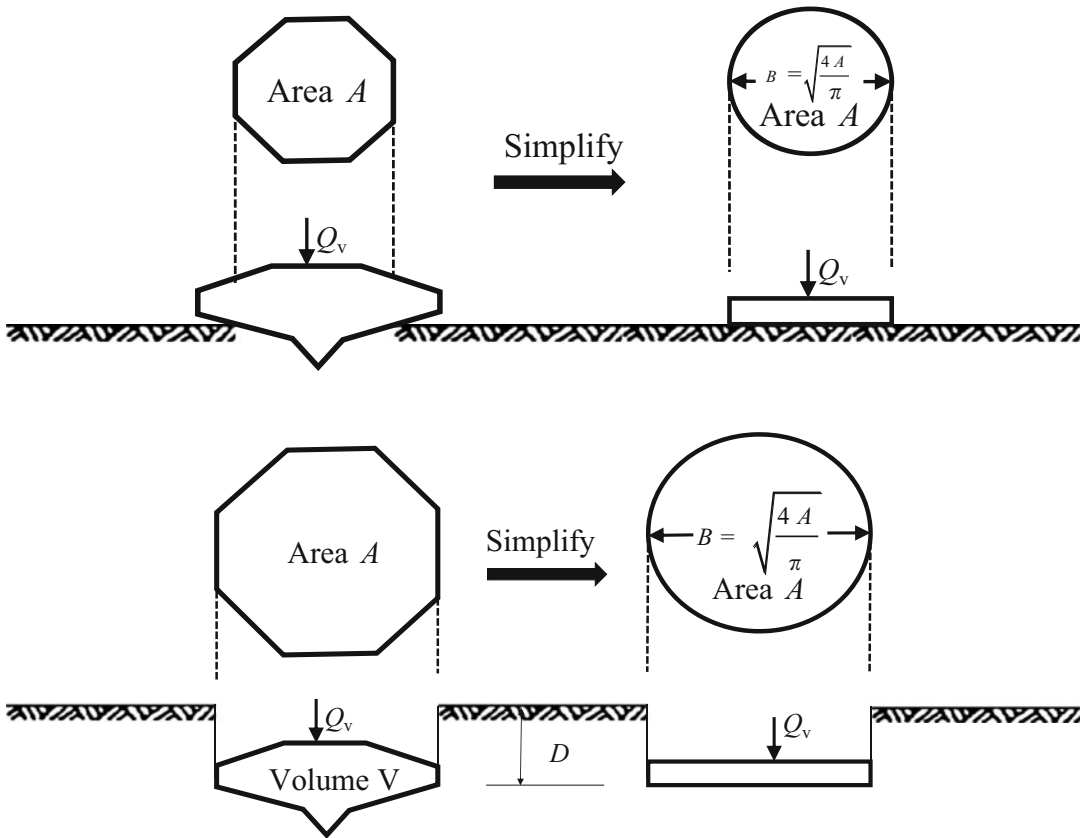
In determining whether a jack-up unit may be safely used at a particular site, two separate analyses of foundation bearing capacity must generally be undertaken:

- Prediction of footing penetration during preloading (vertical load only)
- Assessing of footing stability under design storm conditions (combined vertical, horizontal, and moment loading)

Prediction of Footing Penetration During Preloading

The Influence of Soil Backflow

During penetration, the possibility of soil backflow over the footing should be considered when computing bearing capacity. In very soft clays, complete backflow may occur, whereas in firm to stiff clays and granular materials, where limited footing penetration may be expected, the



Bearing Capacity of Spudcans, Fig. 7 Spudcan foundation model (SNAME 2008)

significance of backflow diminishes. Backflow in clay may be assumed not to occur if:

$$D \leq \frac{Nc_s}{\gamma'}$$

In this case, c_s is taken as the average undrained cohesive shear strength over the depth of the excavation, N is a stability factor, and γ' is the submerged unit weight of the soil. Conservative stability factors in uniform clays, as a function of excavation depth and diameter, are summarized in Fig. 8. For spudcan penetration analyses, it is recommended that conservative criteria are used and the excavation depth be considered as the depth to the maximum spudcan bearing area.

For the conservative evaluation of the hole backflow, the stability factors of Meyerhof (1972) are used. However, for normally

consolidated clay profiles, the Britto and Kusakabe (1983) curve may be more appropriate.

It should be noted that the expression above is based on static hole stability. In reality, during penetration of the spudcan, the soil will probably flow along the spudcan upward onto the top of the spudcan. Hence, the hole stability derived from the expression may be too optimistic.

Penetration in Clays

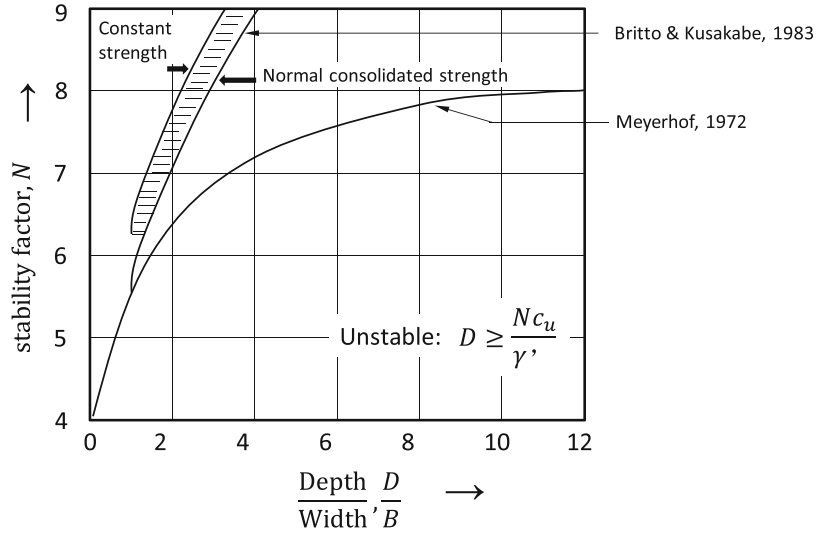
The ultimate vertical bearing capacity of a foundation in clay (undrained failure in clay, $\phi = 0$) at a specific depth can be expressed by:

$$F_V = (cN_{cs}d_c + q')A$$

In which c is the undrained cohesive shear strength at $D + B/4$ below mudline, and B is the effective spudcan diameter at uppermost part of

Bearing Capacity of Spudcans,

Fig. 8 Stability factors for cylindrical excavations (SNAME 2008)



bearing area in contact with the soil. q' is the effective overburden pressure at depth D of maximum bearing area ($q' = \gamma' D$). For circular footing, the value of N_c is taken as 5.14.

The maximum preload is equal to the ultimate vertical bearing capacity, F_V , taking into account the effect of backflow, $F'_0 A$, and the effective weight of the soil replaced by the spudcan, $\gamma' V$, i.e.:

$$VL_0 = F_V - F'_0 A + \gamma' V$$

In which F'_0 is the effective overburden pressure due to backflow at depth of uppermost part of the bearing area. The terms $-F'_0 A + \gamma' V$ should always be considered together. Because when the soil flows back during penetration, the backflow soil covers on the top of the footing, which reduces the maximum preload VL_0 . At the same time, the footing is fully buried in soil and the footing subjected to an up-floating force $\gamma' V$, which increases the maximum preload VL_0 .

The bearing capacity formula given above has been empirically derived for surface foundations and does not account for foundation roughness, shape (conical for most spudcans), or the effects of increased shear strength with depth.

Penetration in Silica Sands

The ultimate vertical bearing capacity of a circular footing resting in silica sand or other granular

material ($c = 0, \phi \neq 0$) can be computed by the following equation:

$$F_V = (0.5\gamma' B N_\gamma s_\gamma d_\gamma + q' N_q s_q d_q) A$$

The maximum preload is equal to the ultimate vertical bearing capacity, F_V , taking into account the effect of backflow, $F'_0 A$, and the effective weight of the soil replaced by the spudcan, $\gamma' V$, i.e.:

$$VL_0 = F_V - F'_0 A + \gamma' V$$

Footing penetrations in a thick layer of clean silica sand are usually minimal with the maximum diameter of the spudcan rarely coming into contact with the soil. It is therefore not usual to consider the effects of soil backflow in this situation.

Typically observed load-penetration data for large-diameter spudcans suggest that reduced friction angles may be applicable for this analysis method. To account for this, it is recommended that laboratory triaxial ϕ test values should be reduced by 5° for the prediction of large-diameter footing penetrations in silica sands, i.e., $\phi_{\text{design}} = \phi_{\text{triaxial}} - 5^\circ$.

For spudcans on sand, the effects of cyclic loading may be to either increase or decrease the pore water pressure. Positive excess pore water pressure will weaken the soil and in severe cases

may cause partial fluidization. Negative excess pore water pressures may temporarily strengthen the soil. Approximate methods are available for the assessment of excess pore water pressure development and associated foundation settlement.

Penetration in Silts

For silts, it is recommended that upper and lower bound analyses for drained and undrained conditions are performed to determine the range of penetrations. The upper bound solution is modeled as a loose sand and the lower bound solution as a soft clay. Cyclic loading may significantly affect the bearing capacity of silts.

Cyclic loads imposed in silty fine sands/silt foundations may cause liquefaction due to the generation of excess pore water pressures. In this situation foundation settlements would be anticipated. Conservative assessments of reduced bearing capacities and increased settlements should be conducted as appropriate.

Penetration in Carbonate Sands

Penetrations in carbonate sands are highly unpredictable and may be minimal in strongly cemented materials or large, in uncemented materials. Extreme care should be exercised when operating in these materials.

Relatively large footing penetrations have been reported for uncemented carbonate materials despite high laboratory friction angles (Dutt and Ingram 1988). This may be attributed to either the high compressibility of these materials or low shear strengths due to high voids ratio and a collapsible structure.

The leg penetration is governed by both strength and deformation characteristics of foundation soils. The compressibility of carbonate sands is relatively higher than for silica sands. Hence, greater penetrations should be expected for carbonate sands relative to silica sands despite the similar or even higher laboratory friction angles.

The predictions of footing penetrations in carbonate sands are likely to be performed to a lower degree of accuracy compared with those for silica sands. The conventional method is to use the plasticity-based formulation for bearing capacity of shallow foundations in sand. However, friction angles to be used in the formulae should be

considerably smaller than laboratory values to account (in an artificial manner) for the soil behavior.

Penetration in Layered Soils

Three basically different foundation failure mechanisms are considered in spudcan predictions in layered soils:

1. General shear
2. Squeezing
3. Punch-through

The first failure mechanism occurs if soil strengths of subsequent layers do not vary significantly. Thus an average soil strength (either c or ϕ) can be determined below the footing. The footing penetration versus foundation capacity relationship is then generated using criteria from the formulae mentioned above. The punch-through condition concerns a potentially dangerous situation where a strong layer overlies a weak layer and hence a small additional spudcan penetration may be associated with a significant reduction in bearing capacity (Liu et al. 2005; Yu et al. 2012). But it is complicated and no good solution has been proposed. In this section, criteria for squeezing are given below.

On a soft clay subject to squeezing overlaying a significantly stronger layer (Fig. 9), the ultimate vertical bearing capacity of a footing given by Meyerhof (1972) is:

For no backflow conditions:

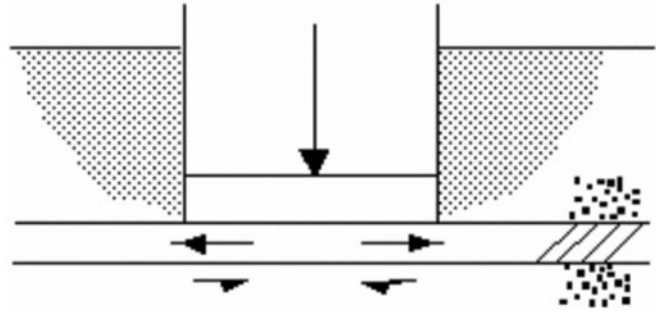
$$F_V = A \left\{ \left(a + \frac{bB}{T} + \frac{1.2D}{B} \right) c + q' \right\} \\ \geq A \{ c N_{cs} d_c + q' \}$$

and for full backflow conditions:

$$F_V = A \left\{ \left(a + \frac{bB}{T} + \frac{1.2D}{B} \right) c \right\} + \gamma' V \\ \geq A \{ c N_{cs} d_c \} + \gamma' V$$

where the following squeezing factors are recommended: $a = 5.00$, $b = 0.33$, and c refers to the undrained shear strength of the soft clay

Bearing Capacity of Spudcans, Fig. 9 The failure mechanism of squeezing



B

layer. T is the thickness of the soft clay beneath the bottom of footing.

It is noted that the lower bound foundation capacity is given by general failure in the clay layer and that squeezing occurs when $B \geq 3.45T$ ($1 + 1.1D/B$). The upper bound capacity (for $T \ll B$) is determined by the ultimate bearing capacity of the underlying strong soil layer.

For a squeezing clay layer, the resistance on the footing cannot become larger than the resistance of the layer underneath the soft clay layer. Thus, there is a limit to the squeezing process.

Assessing of Footing Stability Under Combined Vertical, Horizontal, and Moment Loading

Combined Loading: Vertical Load and Moment ($H = 0$)

Bearing capacity under combined loads V and M has traditionally been studied as the statically equivalent problem of an eccentric vertical load. This interpretation led Meyerhof (1953) to the concept of effective area, in which the eccentric vertical load is assumed to act centrally on a strip foundation of reduced area, as shown in Fig. 10b. Brinch Hansen (1970) was the first to propose that eccentrically loaded circular footing be treated as shown in Fig. 10c.

The bearing capacity interaction for a circular footing on clay is:

$$\frac{V}{V_0} = \left(\frac{\pi + 2 + (B'/L')}{\pi + 3} \right) \cdot \frac{A'}{A}$$

where V_0 is the bearing capacity under central vertical loading and A is the gross footing area πR^2 .

Combined Loading: Vertical Load and Horizontal Load ($M = 0$)

This loading combination is statically equivalent to a single load acting at the footing center line, inclined at an angle to the vertical (Fig. 11).

Penetration in Sand

The general ultimate vertical/horizontal bearing capacity envelope for jack-up footings in sand is as follows:

$$F_{VH} = (0.5\gamma'BN_\gamma s_\gamma d_\gamma i_\gamma + q'N_q s_q d_q i_q)A$$

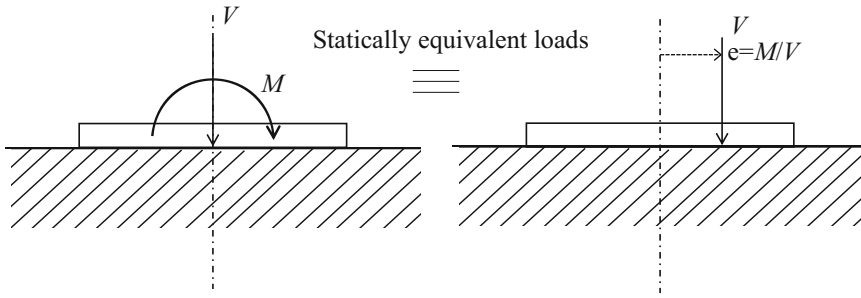
In which F_{VH} is the vertical foundation capacity in combination with horizontal load, and i_γ and i_q are inclination factors which consider the influence of horizontal force. Substituting for i_γ and i_q , the appropriate relationship may be written for generation of the foundation capacity for combined vertical and horizontal loading as:

$$F_{VH} = \left(0.5\gamma'BN_\gamma s_\gamma d_\gamma [1 - (F_H/F_{VH})^*]^{m+1} + q'N_q s_q d_q [1 - (F_H/F_{VH})^*]^m \right) A$$

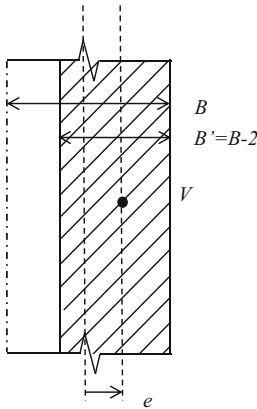
This may be solved by the use of assumed values for (F_H/F_{VH}) designated $(F_H/F_{VH})^*$. For example, use $(F_H/F_{VH})^* = 0, 0.04, 0.08, 0.12$, etc. For these values corresponding F_{VH} values may be determined. The correct F_H values may then be determined as F_{VH} and $(F_H/F_{VH})^*$ are known, e.g., for:

$$(F_H/F_{VH})^* = 0.12, F_H = 0.12F_{VH}^*$$

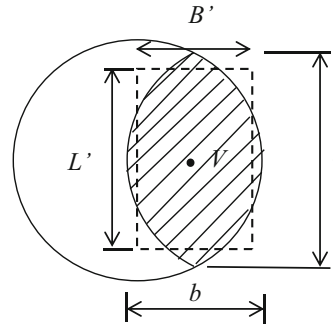
The corrected horizontal capacity, F_H , may then be given as:



(a) The equivalent model for combined V and M loads



(b) Effective area concept: strip footing (Meyerhof, 1953)

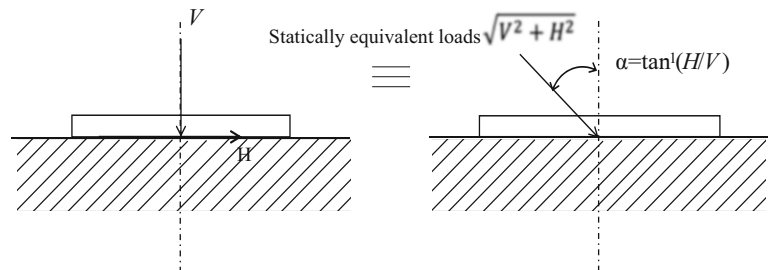


(c) Effective area concept: strip footing (Brinch Hansen, 1970)

Bearing Capacity of Spudcans, Fig. 10 Combined vertical load and moment (Martin 1994) (a) The equivalent model for combined V and M loads (b) Effective area

concept: strip footing (Meyerhof 1953) (c) Effective area concept: strip footing (Brinch Hansen 1970)

Bearing Capacity of Spudcans, Fig. 11 The equivalent model for combined V and M loads (Martin 1994)



$$F_H = F_H^* + 0.5\gamma'(k_p - k_a)(h_1 + h_2)A_S$$

In which k_a is the active earth pressure coefficient, for the case of $c = 0$, $k_a = \tan^2(45 - \phi/2)$.

Accordingly k_p is the passive earth pressure coefficient and $k_p = 1/k_a$. h_1 is the embedment depth to the uppermost part of the spudcan (if not fully embedded $h_1 = 0$), and h_2 is spudcan bottom

embedment depth. A_S is spudcan laterally projected embedded area.

The sliding capacity envelope of a footing in sand is given by:

$$F_H = F_{VH} \tan \delta + 0.5\gamma'(k_p - k_a)(h_1 + h_2)A_S$$

where δ is the steel/soil friction angle which for a flat plate, $\delta = \phi - 5^\circ$, and for a rough surfaced conically shaped spudcan, $\delta = \phi$.

Penetration in Clay

The general equation for the horizontal and vertical bearing capacity envelopes for footings in clay is as follows:

$$F_{VH} = (cN_c s_c d_c i_c + q'N_q s_q d_q i_q)A$$

Substituting for the inclination factors for a circular footing, the equation may be written as:

$$F_{VH} = \left\{ cN_c s_c d_c \left[1 - \left(1.5 F_H^* / c_{AN_c} \right) \right] + q'N_q s_q d_q \left(1 - F_H^* / F_{VH} \right)^{1.5} \right\} A$$

The ultimate bearing capacity envelope under inclined loading may be determined by substituting values of F_{VH} and solving for F_H^* ; F_H may then be given as:

$$F_H = F_H^* + (c_0 + c_1)A_S$$

In which c_0 is undrained cohesive shear strength at maximum bearing area (D below mudline) and c_1 is undrained cohesive shear strength at spudcan tip. When $0 \leq Q_V \leq 0.5F_V$, the sliding capacity in clay may be conservatively assumed constant, determined by $F_H = Ac_0 + (c_0 + c_1)A_S$.

Penetration on Layered Soils

The above formulae can also generally be used to make a conservative estimate of the ultimate $F_{VH} - F_H$ relationship for layered soils by considering failure through the weakest zones in such a soil profile. The bearing capacity of layered soils may be determined using the principles of

limiting equilibrium analysis or the finite element method.

Combined Loading: Vertical, Horizontal, and Moment Loads

This case has usually been approached in the manner first suggested by Meyerhof (1953): an inclined load of magnitude $\sqrt{V^2 + H^2}$ is assumed to act centrally on a reduced foundation area determined by the eccentricity $e = M/V$, as illustrated in Fig. 12.

For shallow embedment for both sand and clay, the yield interaction is defined by the following expression:

$$16 \left[\frac{F_{VHM}}{V_{L0}} \right]^2 \left[1 - \frac{F_{VHM}}{V_{L0}} \right] \left[1 - \frac{F_{VHM}}{V_{L0}} \right] - \left[\frac{F_{HM}}{H_{L0}} \right]^2 \left[\frac{F_M}{M_{L0}} \right]^2 = 0$$

In which F_{VHM} represents vertical foundation capacity in combination with horizontal and moment load; F_{HM} represents horizontal foundation capacity in combination with moment. F_M is the moment capacity of foundation, where V_{L0} is taken to be equal to the vertical spudcan load achieved during preloading and H_{L0} and M_{L0} are defined as follows:

For sand:

$$H_{L0} = 0.12V_{L0}, M_{L0} = 0.075BV_{L0},$$

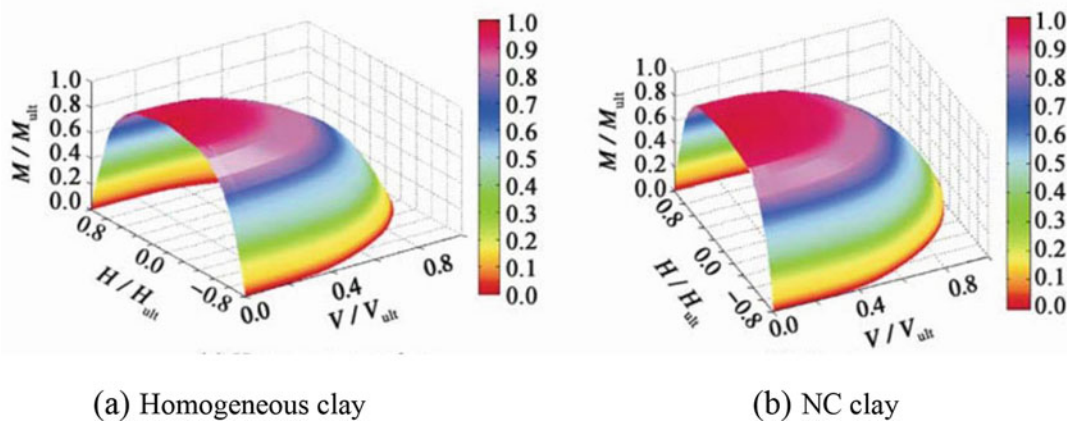
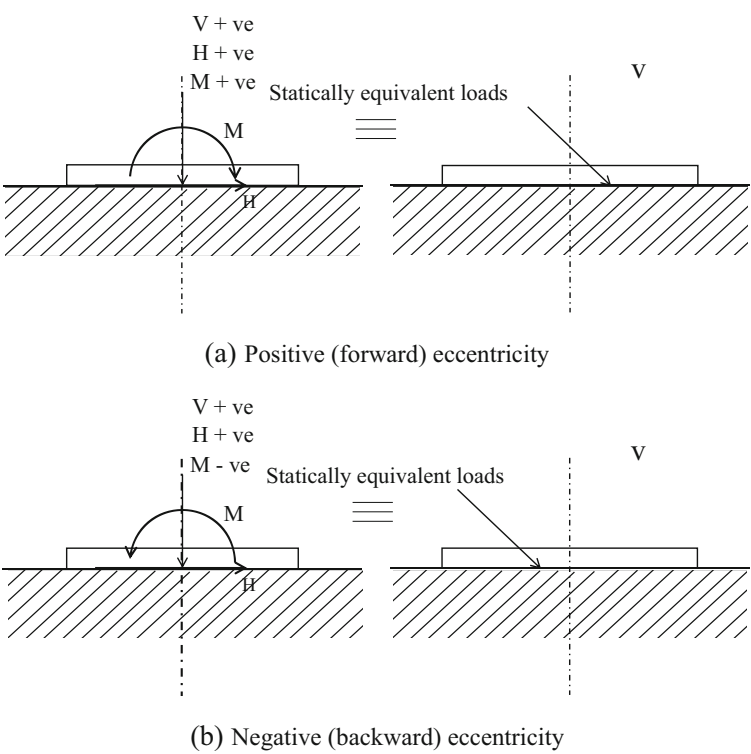
For clay:

$$H_{L0} = c_0A + (c_0 + c_1)A_S, M_{L0} = 0.1BV_{L0}$$

Note that in the above expression for the yield surface, if a load combination (Q_V, Q_H, Q_M) satisfies the equality, then $(Q_V, Q_H, Q_M) = (F_{VHM}, F_{HM}, F_M)$. The load combination (Q_V, Q_H, Q_M) lies outside the yield surface if the left-hand side is less than zero. Conversely, the load combination lies inside the yield surface if the left-hand side is greater than zero.

Embedded footings in clay achieve greater moment and sliding capacities as compared to shallow penetrations in clay. Figure 13 is an example of 3-D failure envelope of embedded

Bearing Capacity of Spudcans, Fig. 12 The equivalent model for combined V , H , and M loads (Martin 1994) (a) Positive (forward) eccentricity (b) Negative (backward) eccentricity



Bearing Capacity of Spudcans, Fig. 13 A three-dimensional failure envelop for combined loading V , H , and M (Wang et al. 2011) (a) Homogeneous clay (b) NC clay

spudcan in clay. There is no existing data for deeply embedded footings in sand. The application of the yield surface calibrated to shallow penetrations will likely be conservative for the deep penetration case.

Cross-References

- [Installation of Spudcans](#)
- [Pile Capacity](#)
- [Shallow Foundations](#)

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BEM – Blade Element Momentum

- [Analysis of Renewable Energy Devices](#)

Bending Strength of Sea Ice

- [Flexural Strength of Sea Ice](#)

Big Data

- [Big Data-Based Decision Support Systems](#)

Big Data-Based Decision Support Systems

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Synonyms

[Big data](#); [Data analysis](#); [Machine learning](#); [Ship navigation](#); [Ship navigation sensor](#)

Definition

Big data is not a newly developed concept from an evolutionary perspective. Various data warehouses have been created to store huge amount of data, such as weather data, consumption data, and stock index data, ever since the 1990s. During that time, one terabyte was considered as big data, but any numerical definition is expected to be modified over time as the development of collecting, storing, and analyzing data technology (Watson 2014). One important characteristic is

that big data can be loosely defined as data sets with sizes beyond the ability of common statistical software or database management system (Snijders et al. 2012). An alternative approach is to feature big data mainly in 3 Vs – volume, velocity, and variety (Hilbert 2016). In today's digital world, every single thing can be regarded as of generating data, for example, sensors/meters and activity records from electronic devices, social media, business transactions, electronic files, etc.

Overview of Big Data Techniques

Hence, the data process ability is the crucial point for the big data era. The big data process generally consists of data collection, data processing, data cleaning, exploratory data analysis, and modeling and algorithms; see Fig. 1 for flowchart (O'Neil and Schutt 2013).

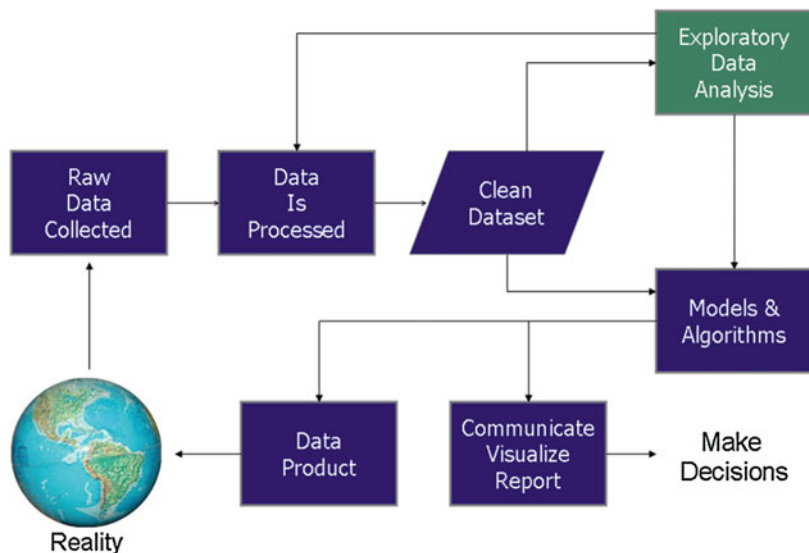
In addition, the collection and storage of data have little values for practical applications if no methods have been applied for the exploitation of the data, i.e., so-called data analysis. The data analysis is the key to make use of these tremendous databases, which can help to enhance decision-making, provide insight and projection, and support and optimize processes.

Within the field of data analysis, machine learning is one of the promising methods that integrate mathematical technologies with computer systems. They can increase the capability of formulating complex models and algorithms to learn and make projections on data (Koza et al. 1996). Those analytical models allow researchers and engineers to make dependable and repeatable results and decisions; see Fig. 2 (Business Link Solutions). It is used extensively in the development of autonomous vehicles, such as self-driving cars, unmanned ships, speech recognition, information retrieval, credit card fraud, spam detection, product recommendation, medical diagnosis, customer segmentation, etc. Some of the largest online selling companies also use the big data techniques to target the potential customers of their products to boost their business.

Depending on the strategies used in the data analysis process, machine learning is commonly categorized as the following techniques: supervised learning, semi-supervised learning, active learning, unsupervised learning, and reinforcement learning. When using the supervised learning, the existing data contains the desired output and the goal. For example, if one wants to predict a ship's power in terms of the ship's sailing speed, heading, and encountered sea

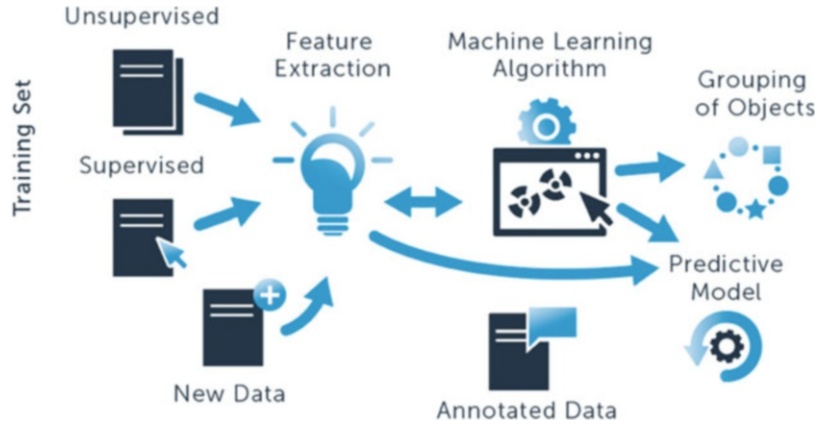
Big Data-Based Decision Support Systems,

Fig. 1 General big data science process flowchart (O'Neil and Schutt 2013)



Big Data-Based Decision Support Systems,

Fig. 2 The typical process of machine learning (Business Link Solutions)



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environment, both the predicted target, i.e., a ship's power and the dependent variables, i.e., speed, sea parameters, etc., should be included in the data for supervised learning. This learning technique is to obtain a certain model that describes the relationship between inputs and outputs (also called targets). There are two types of supervised problems. In classification problems, the target outputs are discrete, while regression problems deal with continuous outputs. A typical example of a classification problem is to predict whether incoming mail is spam or not. A regression example is to predict the value of a house given its size, number of rooms, etc. In unsupervised learning the desired outputs are unknown, and the learning algorithm looks for patterns and/or structure in the data. A typical example is a clustering problem like customer segmentation, finding out which customers exhibit similar behavior. In reinforcement learning, the computer/algorithm interacts with a dynamic environment and learns the ideal behavior within a specific context. Some sort of feedback is required during the learning process to obtain this behavior. For example, a computer-controlled playing chess game or autonomous driving often adopts this type of learning. Following are the most common tasks, regression, classification, clustering, multivariate querying, density estimation, dimension reduction, testing, and matching (Christopher 2006).

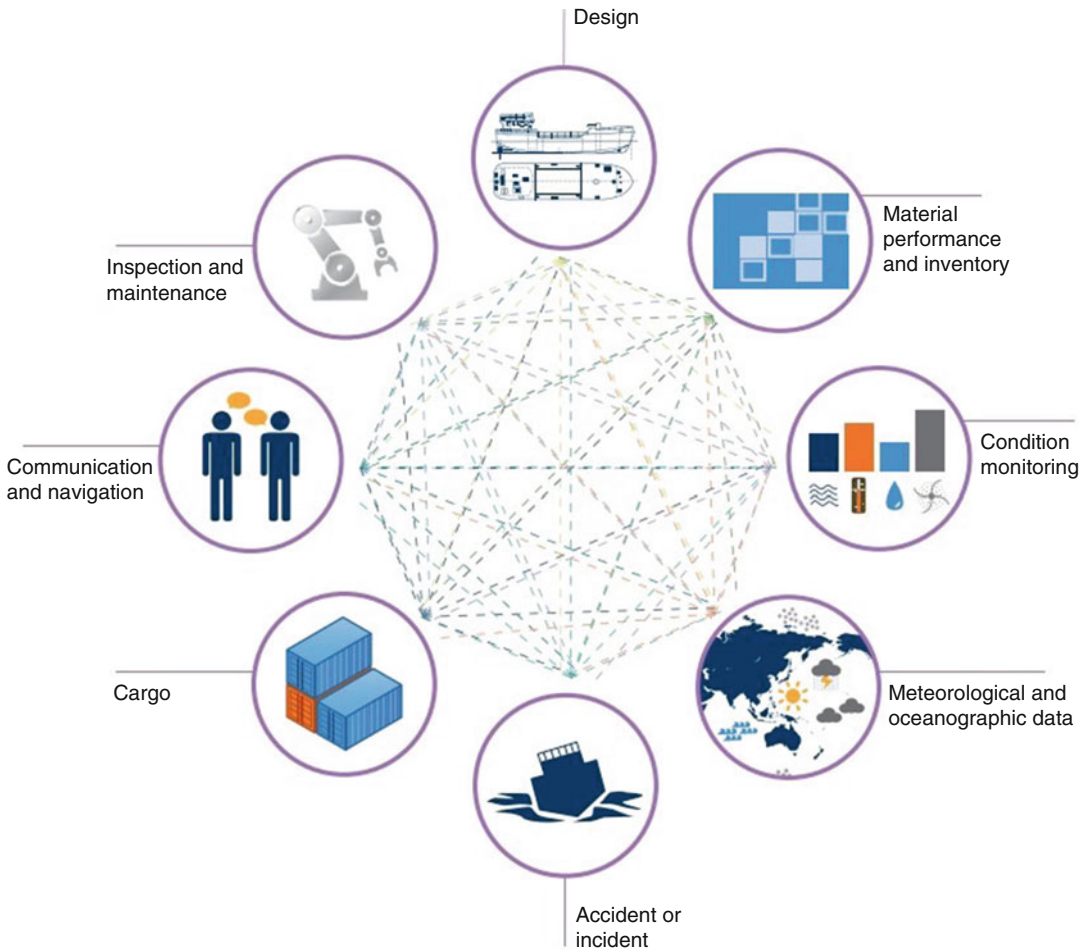
The most classical outlining framework for approaching machine learning (supervised) is

proposed by Guo (Towards Data Science); the following are the seven systematic steps:

1. Gathering data: for per-collected data, the quantity and quality highly influence how accurate the model and algorithm is.
2. Data preparation: clean the required data (deal with missing values, remove duplicates, normalization, etc.), split into training and evaluation set.
3. Choosing a model: choose the correct models and algorithms for different cases.
4. Training: the aim of this step is to make the prediction correctly; the process is repeated until to an ideal separation.
5. Evaluation: test the model against previous data that has never been used in training, to measure the objective performance of model.
6. Parameter tuning: tuning the parameters to further improve the training, the hyperparameter may include number of training iteration, learning rate, etc.
7. Prediction: use further data to test the model and to assess how the model will perform in the real application.

Big Data Applications in Marine Technology

As stated in Global Marine Technology Trends 2030 report, big data will be one of the most important transformational techniques being



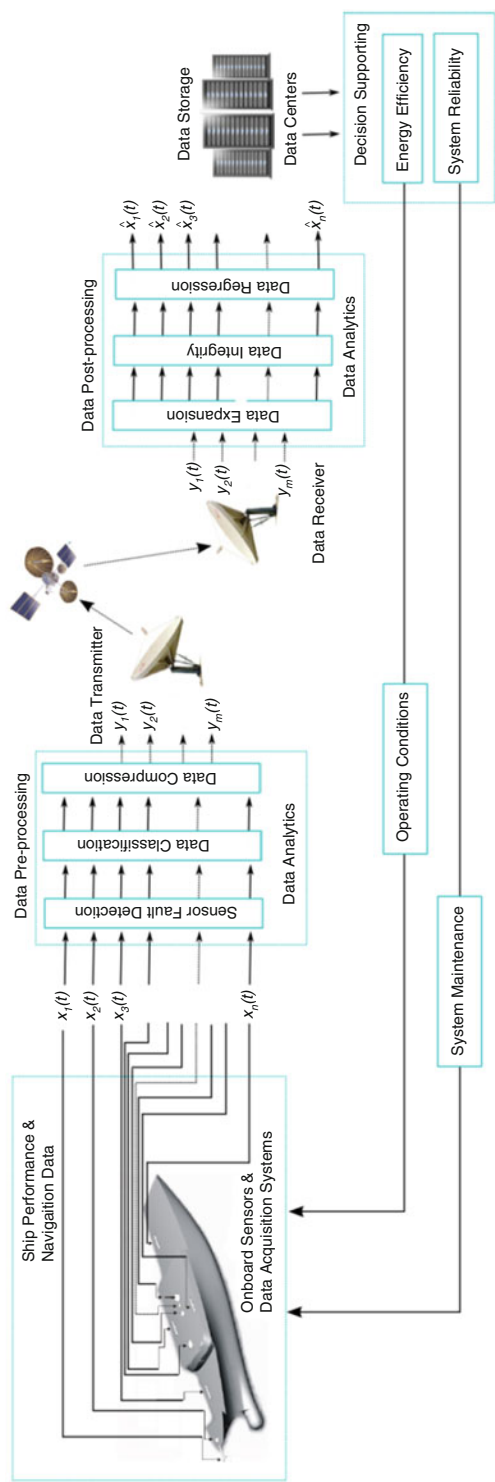
Big Data-Based Decision Support Systems, Fig. 3 Big data analytics are expected to be used in the marine industry (Shenoi et al. 2015)

used in marine technology (commercial shipping, naval, and ocean); see Fig. 3 (Shenoi et al. 2015). The data generated and collected from aerial remote sensing, ship, stations, buoys, and satellites are rapidly growing all over the world. Those tremendous data can be processed and analyzed to provide new knowledge and added values that we never know before in marine industry, which is relatively traditional and slow-moving (Trelleborg Marine System).

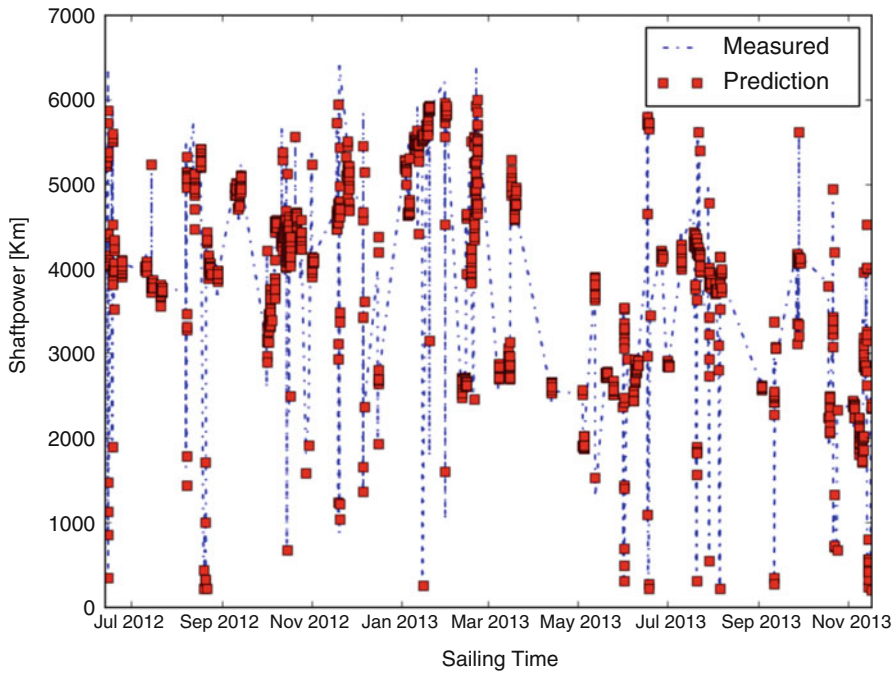
This cross technology will introduce the big data-driven innovations to ship design, material performance and inventory, condition monitoring, meteorological and oceanographic projections,

inspection and maintenance, communication and navigations, etc.

Big data support decision-making systems have been widely used in maritime transport. Shipping has taken over 90% of the world trade volume. As the development of sensors, tracking system onboard, and communication techniques between ship and shore, the maritime transport is embracing the digital era (Mirovic et al. 2018). DNV-GL, ClassNK, and major classification societies have set their blueprints in big data, optimizing the decision-making, to lower the fuel consumption and overall cost and ensure the safety (Koga 2015). A classical maritime



Big Data-Based Decision Support Systems, Fig. 4 Data flowchart for ship navigation and performance (Perera and Mo 2016)



Big Data-Based Decision Support Systems, Fig. 5 Shaft power prediction and the corresponding full-scale measurements over the whole navigation campaign (Mao et al. 2016)

transport data flowchart is presented in Fig. 4 (Perera and Mo 2016).

The onboard sensors measure the ship performance, navigation data, and process and transmit to data center, where the recorded data are stored and analyzed. Then the decision support system optimizes and improves the operations and ship management.

For the shipping management, one of the top issues is the ship route plan and optimization, based on the ship fuel consumption estimation. Conventional fuel consumption model contains marine engine configuration, ship resistance, hull and propeller efficiency, etc., which has high uncertainties in the inputs. In order to estimate the ship fuel consumption accurately, the big data technique, more specifically machine learning, has been successfully applied in (Mao et al. 2016) to predict shaft power for a C-class 45000 DWT chemistry tanker; see Fig. 5 (Mao et al. 2016).

Machine learning has shown the capacity to predict ship power and fuel cost with an acceptable discrepancy. In addition, it has the potential for ocean weather prediction and even ship route optimization directly. As more and more organizations are investing and shipping companies are willing to share the data information, big data will become the “oil of the twenty-first century” for the maritime industry, and the shipping will wake up eventually to the new digital era (Marine Digitalization & Communications).

Cross-References

- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)
- [Impact of Maritime Transport](#)
- [Ship Navigation System](#)
- [Ship Operational Environment](#)

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Biodiversity Conservation

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Synonyms

Conservation biology; Conservation of biological diversity

Definition

Biodiversity is “the variety of different forms of life on earth, including the different plants, animals, micro-organisms, the genes they contain and the ecosystem they form” (Rawat and Agarwal 2015). Biodiversity conservation is “about saving life on Earth in all its forms and keeping natural ecosystems functioning and healthy” (Rawat and Agarwal 2015).

Scientific Fundamentals

Three Foundational Elements of Biodiversity Conservation

In 1990, the International Union for the Conservation of Nature (IUCN) identified the conservation of genetic diversity, species diversity, and ecosystem diversity as the three foundational elements of biodiversity conservation (McNeely et al. 1990).

Genetic diversity is “the sum of genetic information and trait contained in the genes of individuals of plants, animals and micro-organisms” (Pearce et al. 1994). Genetic diversity provides the basis for adaptation to environmental change, and such adaptation is the key to the long-term survival of species (Schemske et al. 1994). Preserving high levels of heritable variation and preventing fixation of deleterious alleles are the two primary goals of conservation biologists in their study and application of genetics (Dyke and Lamb 2020).

Species diversity is “the variety of species or the living organisms” (Rawat and Agarwal 2015). There are many different definitions of species. The Most widely employed is the Mayr’s definition that a species is “a group of interbreeding populations that are reproductively isolated from other such groups” (Mayr 1969). As the molecular techniques have developed, people can examine and compare the information of the DNA, chromosomes and gene loci of organisms, the phylogenetic species concept was proposed. According to the phylogenetic species concept, a species represents “a group of organisms with an assumed or determined common ancestral lineage whose genetic similarities, differences, and distances are distinguishable from other such groups” (Dyke and Lamb 2020). Mayr’s definition is applied to organisms that reproduce sexually and the phylogenetic species concept is applied to any organism that has genes.

Ecosystem diversity deals with the variety of habitats, biotic communities, and ecological processes in the biosphere and its overall impact on human existence and the environment (Pearce et al. 1994). Diversity of ecosystem can be described as functional diversity (the relative abundance of functionally different kinds of organisms), community diversity (the number sizes and spatial distribution of communities or patchiness), and landscape diversity (the diversity of scales of patchiness) (Pearce et al. 1994). By conserving biodiversity at the ecosystem level, not only can constituent species be preserved, but the ecosystem functions and services can also be protected.

Conservation Biology

Biodiversity is the common wealth of human beings. Biodiversity has very important values,

not only utilitarian value but also intrinsic value. The utilitarian value can be divided into direct utilitarian value (e.g., as food, fuel, and materials), ethical and moral value, ecological value (e.g., soil formation and protection, nutrient storage and cycling), aesthetic value (e.g., Ecotourism), and so on (Mutia 2009). Intrinsic value of biodiversity refers to the value of biodiversity in and for itself, regardless of its utility and usefulness. Biodiversity is disappearing at an unprecedented rate as a result of human activities. The key pressures causing biodiversity loss are overexploitation of biological resources, environmental pollution, climate change, invasive alien species, the degradation, fragmentation, destruction of habitats, and so on (Butchart et al. 2010). With the increasing problems of biodiversity, people recognized that the protection of biodiversity is very necessary. In order to maintain the sustainable development of biodiversity, conservation biology as a crisis discipline was birthed.

In 1978, the First International Conference on Conservation Biology was held at the San Diego Wild Animal Park in the United States. In 1985, the formal organization and incorporation of the Society for Conservation Biology (SCB) was established. At the beginning, Soulé noted that “conservation biology is a crisis discipline grounded in the recognition that humans are causing the death of life – the extinction of species and the disruption of evolution” and its goal was to provide principles and tools for preserving biodiversity (Soulé 1985). Currently, conservation biologists have shifted focus from the problem of “endangered species” to “loss of biodiversity.” The two major missions of conservation biology are to conserve the biodiversity on the Earth and safeguard the benefits that human welfare access through the functional operation of ecosystems (Dyke and Lamb 2020). Conservation biology is a synthetic, eclectic, and multidisciplinary discipline.

Protecting and improving the ecological integrity and long-term viability of the core regions rather than patches should be the basic feature of biodiversity conservation strategies (Ayyam et al. 2019). In-situ and ex-situ conservations are the two basic types of conservation methods. Ex-situ conservation refers to conservation of components of biodiversity in selected areas outside

their natural habitats, including botanical gardens, zoological gardens, and conservation stands (Ayyam et al. 2019). In-situ conservation refers to conservation of species in their natural habitats or natural ecosystems, including national parks, sanctuaries, and biosphere reserves (Ayyam et al. 2019).

Marine Biodiversity

Marine system possesses its own unique array of biodiversity. It must be considered carefully and separately in evaluating their abundant life and resources. Marine systems are disproportionate contributors to the global biodiversity. The estimates number of extant marine species range from ~300,000 to 2.2 million using different assessment methods (Luypaert et al. 2020). Of the estimated marine species, approximately 226,000 eukaryotic marine species have been described as of 2012 (Appeltans et al. 2012), and now, approximately 235,000 have been described (WoRMS Editorial Board 2020). There is considerable uncertainty in estimates of the exact number of marine species and this uncertainty is particularly caused by most of the marine habitats being under-sampled and the biological groups with few taxonomic experts (Luypaert et al. 2020).

Marine biodiversities are vulnerable to human activities and they are under threat when their resources extracted in destructive ways. The threats listed by the IUCN can be grouped into six categories: habitat destruction and modification, over-exploitation, invasive alien species, pollution, climate change, and other (Young et al. 2016; Luypaert et al. 2020). Habitat destruction and over-exploitation are the major cause of declines of marine species. Habitat destruction is “the reduction in habitat quality or complete removal/conversion of ecosystems and their related functions” (Luypaert et al. 2020). Marine habitat can directly effect by human intervention (e.g., bottom trawling, pollution and mining) or indirectly caused by ocean warming, acidification, and so on (Rossi 2013). Overexploitation can not only cause the decline of target species but also affect populations of other non-target species through multiple collateral effects. Almost 80% of the world’s fisheries are overexploited and some of

them are in a state of collapse (Imtiyaz et al. 2011). Overfishing is the greatest threat to the endangered or vulnerable extinction species, and it threatens many other species, particularly large sized fishes (e.g., sharks, swordfish) (Imtiyaz et al. 2011). Marine pollutions, including heavy metals, oil spill, and marine debris, can have important consequences for marine species (Ayyam et al. 2019). An invasive alien species, intentionally or accidentally transported into an environment outside of its historic geographical range, can create and change the original habitat and influence the structure and function of marine ecosystems (Dyke and Lamb 2020). Climate change with rising temperatures and more frequent extreme climate events, causing ocean acidification and expanding marine dead zones, led to wide and varied consequences for marine biodiversity (Ayyam et al. 2019; Dyke and Lamb 2020).

The loss of marine biodiversity is widely acknowledged and it is very important and urgent for the conservation of marine biodiversity. To date, 7.7% of the ocean is protected according to the World Database on Protected Areas (WDPA) and 2.7% of the ocean is fully or highly protected from fishing impacts (Marine Conservation Institute 2021). Marine biodiversity hotspots, a combination localities of high species richness, endemism or threat, are always prioritized sites of conservation (Jefferson and Costello 2019). There are 18 marine biodiversity hotspots globally, including The Coral Triangle, the waters of Japan, and China, the South Africa or the Eastern African region, and so on (Jefferson and Costello 2019). Marine protected areas (MPAs) have shown promising roles on marine biodiversity conservation. Large MPAs may be ideal for biodiversity conservation but may limit the exploitation of fish stocks to well below sustainable levels, and small MPAs may provide a conservation of the sedentary species but unlikely to provide an effective refuge for highly mobile species (Luypaert et al. 2020). According to the experience of already known MPAs, successful MPAs often have the key features of being no-take zones, well-enforced, older than 10 years, larger than 100 km², and isolated by deep water or sand (Edgar et al. 2014).

Important International Conventions/Plans

Convention on Biological Diversity

The Convention on Biological Diversity (CBD) is a legal international treaty for biodiversity conservation. CBD was first signed at the United Nations Conference on Environment and Development in Rio de Janeiro in June 1992 and came into force in 1993. Over 150 governments signed the document at the 1992 Rio Earth Summit, and since then more than 175 countries have ratified the agreement. CBD is dedicated to promoting sustainable development and established with three main goals: conserve biological diversity, use its components sustainably, and share the benefits from the use of genetic resources fairly and equitably (Convention on Biological Diversity 2000). To advance these goals, the signatories pledged to develop plans to “protect habitats and species; provide funds and technology to assist developing countries to provide protection; ensure commercial access to biological resources for development; share revenues fairly among sources and developers; establish safety regulations and accept liability for risks associated with biotechnology development” (Parson et al. 1992). In order to make the conventions properly implemented, the Conference of the Parties (COP), composed of governments that have ratified the convention, take periodic meetings to check the progress of the convention, identify new protection priorities for member states and develop work plans (Cooper and Mooney 2013). The Global Environment Facility (GEF) projects, providing financial mechanisms, help forge international cooperation and finance actions to address critical threats to the global environment.

The Census of Marine Life

The Census of Marine Life (CoML) is the largest, broadest, and most invested international cooperative of marine life surveys by far. The program lasted from 2000 to 2010, involving more than 2700 marine biologists from more than 80 countries and setting out more than 540 voyages (Ausubel et al. 2010). The voyages involved the survey of marine biodiversity at almost all depths, environments and ocean areas of the world, especially in the deep sea, hot springs, seamounts, cold springs,

and other marine environments. The study collated a large amount of historical data, evaluated the diversity of species and the distribution of marine life, and established a research system for marine biodiversity. The research content of CoML mainly includes the following aspects: History of Marine Animal Populations (HMAP), Future of Marine Animal Populations (FMAP), and the Ocean Biogeographic Information System (OBIS) (O’Dor and Yarincik 2003). The Census made tremendous achievements, many books, papers, websites, videos, films, maps, and databases were published or established. It increased the estimate of known marine species from about 230,000 to nearly 250,000, found more than 6000 potentially new species, and completed formal descriptions of more than 1200 of them (Ausubel et al. 2010). The Census found life everywhere even in unexpected places at the site of a hydrothermal vent in the deep Mediterranean where the first discovery of anaerobic animals living without oxygen (Ausubel et al. 2010). However, the study also found that overfishing, habitat destruction, alien species, pollution, rising sea temperatures, and ocean acidification are threatening the survival of marine life, and the populations of some marine organisms are gradually shrinking or even on the verge of extinction.

Key Applications

Biodiversity and Ecosystem Services

Biodiversity can provide human beings economic values through its use values and non-use values (Pearce et al. 1994). It plays an important role in providing ecosystem services and the demand of people from those services. Ecosystem services are “the benefits people obtain from ecosystems. These include provisioning services such as food and water; regulating services such as regulation of floods, drought, land degradation, and disease; supporting services such as soil formation and nutrient cycling; and cultural services such as recreational, spiritual, religious, and other non-material benefits” (Millennium Ecosystem Assessment 2005). The loss of biodiversity

caused by intensified human activities has led to serious degradation of ecosystem services and conservation efforts have only recently begun to recognize the role of biodiversity in ecosystem services. Scientists and government have made some efforts from biodiversity and ecosystem level approaches, such as ecosystem-based management (EBM) and the Intergovernmental Science-Policy Platform on Biodiversity and Ecosystem Services (IPBES). EBM is “a management approach that defines management strategies for entire ecosystems, not simply individual components of ecosystems and it takes into account interactions among ecosystem components and management sectors” (Dyke and Lamb 2020). The IPBES was established in Panama City in 2012 to “strengthen the science policy interface for biodiversity and ecosystem services for the conservation and sustainable use of biodiversity, long term human well-being and sustainable development” (<https://www.ipbes.net/>).

Deep-Sea Marine Biodiversity

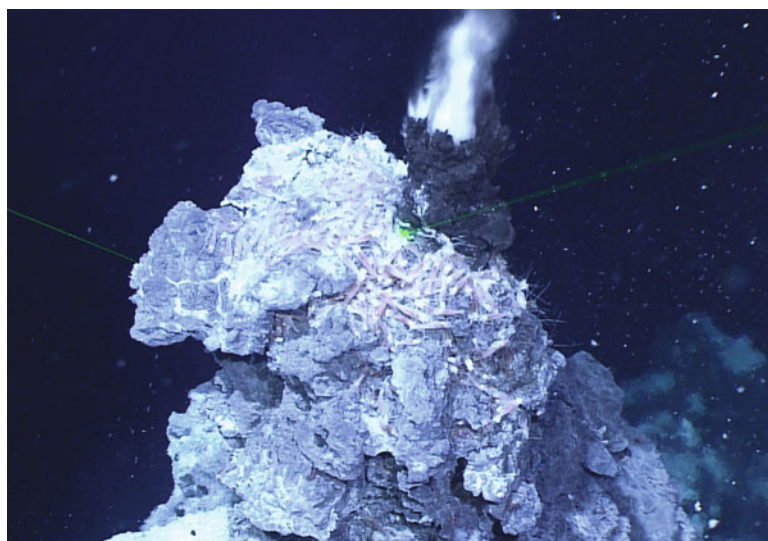
Deep-sea is the largest reservoir of ecosystems and biodiversity on the planet. It contains a variety of complex environments, such as plains, seamounts, trenches, abysses, hydrothermal vents, and cold seeps, giving birth to unique ecosystems and life processes. The World Register of Deep-Sea Species database has recorded 25,902 species

at depths of more than 500 m, including plankton and benthos, as well as many microorganisms (Glover et al. 2020). The richest biodiversity areas involved in hydrothermal vents, cold seeps, and seamounts in deep-sea.

Hydrothermal vents are fissures on the seafloor from which geothermally heated water issues (Fig. 1). They are usually found near deep-sea areas with active crustal movements, such as mid-ocean ridges, back-arc basins, and intracontinental rift valleys. There are many large invertebrates congregated on there and formed oases of life in the deep-sea. In 1977, the first hydrothermal vent was discovered at the depth of 2500 m of the Galapagos in the East Pacific Rise mid-oceanic ridge using the manned submersible “Alvin” (Gallardo 1977). Varieties of macrobenthos, such as tubular worms, shellfish, and crustaceans, thrive around hydrothermal vents by grazed, filtered, or hosted with these microorganisms. Chemosynthetic microbes are the primary producers in the vent food web, and they can provide nutrition for the vent animals. By 2011, the Biogeography of Chemosynthetic Ecosystems (ChEss) databases listed more than 700 hydrothermal vent species (German et al. 2011). Among them, arthropods, molluscs, and polychaetes are the most abundant taxa and account for more than 90% of all hydrothermal biological species (Tunnicliffe and Fowler 1996).

Biodiversity

Conservation, Fig. 1 A chemosynthetic community with caridean shrimp as a dominant species found around a chimney of hydrothermal vent from the Okinawa Trough at about 970 m depth. (Photographed by ROV “Faxian” of the Institute of Oceanology, Chinese Academy of Sciences, provided by Dr. Xin Zhang)



Cold seeps are areas of the seafloor where hydrocarbon-rich fluid seepage occurs and gases leak from fissures and emerge through the sediments and into the water column, which support a characteristic biome based on chemosynthetic organisms (Dong et al. 2020) (Fig. 2). The first cold seep was discovered on the Florida Escarpment in the Gulf of Mexico at a depth of 3200 m (Paull et al. 1984). By 2011, some 600 cold seep species have been described and listed in the ChEss database (German et al. 2011). The density of macroinvertebrates in cold seeps is significantly higher than that in the surrounding seafloor environment, but the species diversity is lower (Oliver et al. 2011). The species of ecosystems are similar within seep and vent habitats in the composition at the high taxon level, but there are significant differences at the species level (Sibuet and Olu 1998). The epibenthic macroinvertebrates always have low diversity and high dominance of single species, especially the molluscan and polychaete species (Oliver et al. 2011).

Seamounts are widely distributed topography with summit depths at least 100 m above the seafloor, which are often formed by volcanic activity (Yesson et al. 2011). Seamounts may be divided into hills (<500 m from seafloor), knolls (>500 m from seafloor), and seamounts (>1000 m from seafloor) (Yesson et al. 2011). There are more than 150,000 seamounts and

knolls and more than 25 million seamounts with summit depths at least 100 m (Wessel et al. 2010; Yesson et al. 2011). Seamounts are hotspots of pelagic biodiversity in the ocean. The biomass, abundance, and diversity of plankton, swimming organisms, and benthos of seamounts are often higher than those of other deep-sea ecosystems (Rogers 2018). Suspension feeders (e.g., sponges, corals) usually dominate the communities of the megabenthos on seamounts, and they can provide important habitat for mobile invertebrates such as molluscs, crustaceans, and echinoderms (Rogers et al. 2008). Some fish can form aggregations around seamounts and pelagic marine predators (e.g., shark, tuna) are attracted to improve opportunities for foraging. Very few data of invertebrates living at seamount are available and over 100,000 seamounts were estimated worldwide, only 200 of seamounts have been biologically sampled (Samadi et al. 2008). The reported number of seamount species may be seriously underestimated of the total number due to very few surveys have confirmed the complete specimens collected into an accurate species as the lacking of taxonomical expertise globally (Samadi et al. 2008). The limited habitat, apparently limited recruitment between seamounts, extreme longevity, and highly localized distribution of many species make seamount communities highly vulnerable to the impacts of fishing

Biodiversity

Conservation, Fig. 2 A chemosynthetic community (*Bathymodiolus platifrons*–*Shinkaia crosnieri* community) found in Jiaolong Cold Seep I at about 1120 m depth in northeastern South China Sea (Li 2017)



(Samadi et al. 2008). Temporary fishing closures and the establishment and enforcement of permanent protection zones are required to achieve efficient and long-term protection of seamounts (Samadi et al. 2008).

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Biofouling

► New Technologies in Auxiliary Propulsions

Block Assembly Line

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Synonyms

Flux copper backing (FCB); TRYGGHET TILLIT SERVICE (TTS)

Definition

Block assembly is a technological process of sub-assemblies and parts into ship block, also known as block manufacturing. According to the characteristics of block production, it can be divided into flat section, curved section, and superstructure section. According to statistics, in modern shipbuilding, hull assembly and welding account for 12–18% of the total labor in shipyard construction. Therefore, improving the mechanization and automation level of assembly and welding work can not only reduce the labor intensity of workers but also effectively shorten the shipbuilding cycle and improve the product quality. Due to the adoption of new technologies such as digital amplification and numerical control, the accuracy of parts processing is continuously improved and the

mechanization of assembly work is implemented (Liu and Wang 2011).

Introduction

Subassembly Line

There are usually four types of subassemblies: composite members of T iron, internal members of flat section, internal members of curved section, and iron outfits. Foreign large and medium-sized shipyards generally use T iron production lines (some of which are automatic production lines), equipped with automatic transfer rollers, gantry type cutting machines, robot welding machines, and corresponding sorting devices, which have higher production efficiency. Some steel mills roll different specifications of T iron for shipyards, and even there are specialized factories that specially assemble various T iron.

For the subassemblies of the flat and curved sections, several technological processes are generally adopted as followed: parts positioning, assembly welding, turning, and assembly. According to the size of the batch, manual welding is generally used on the platform, or mechanized installation and welding equipment is adopted with the auxiliary of transfer roller. In the case of mass production, mechanized frame production lines can be set up or the production lines can be formed as part of a certain sectional production line. For some iron outfits, they are generally assembled on the platform in the specified area of the hull workshop (Xu 2005).

For some large batches of splicing plates, reinforced webs and flat grillages, welding stations with higher degree of automation, such as pressure frame splicing plates and reinforced webs with high efficiency, can be used as part of the flat sectional assembly line. For many skeleton frameworks and frames in ship construction, shipyards generally adopt special and efficient welding assembly stations. For example, when Hudong Shipyard was building 70,000 DWT grade bulk carrier, it considered that there were 180 frames in the bottom section, so it adopted a special station and a flow line operation method. Its production process was as follows: steel plate pretreatment, numerical control cutting, partial assembly welding, distortion

correction by frame, frame welding and hoisting into the outfield for block manufacturing. In this way, the speed of block construction is greatly improved (Liu and Wang 2011).

Block Assembly Line

The structure and shape of the hull are complex and diverse, and it is unrealistic to use a mechanical device (or an assembly line) to complete the manufacture of all blocks. Therefore, when studying the construction of hull block assembly line, it is necessary to divide the sections into several types according to the similarity of the structure and appearance characteristics of various blocks, and formulate standard process regulations according to the section types, so as to form various mechanized intelligent production lines. Flat section and curved section are two basic sections in hull section, and their structures are typical, accounting for a considerable proportion of the workload, of which flat section has more workload. Therefore, at present, the research and test work of mechanized, automatic, and intelligent production lines are all carried out with these two sections as the objects (Ryu et al. 2020).

Flat Section Line

Generally speaking, the percentage of flat section must be at least 20%, and the establishment of flat section line is of practical value (Li and Wei 2006). With the large scale of ships, shipbuilding industry attaches great importance to block manufacturing, especially flat sectional manufacturing. Major shipyards generally adopt flat section assembly lines, and the technical level and production efficiency are continuously improving.

1. Coverage of Flat Section Assembly Line Process

Under normal circumstances, the flat section assembly line can include: waiting station (plate, longitudinal, floor, preoutfitting item), splicing plate assembly station, splicing plate welding station, inspection and repair station, marking and cutting transposition, longitudinal assembly station, longitudinal welding station, transverse movement station, floor assembly station, floor welding station, turning station, preoutfitting station, and delivery station.

The number of work stations in the assembly lines varies from factory to factory. According to the specific situation, there are usually 8–10 work stations. The main work station consists of steel material preparation, panel tailor-welding, marking and cutting, skeleton framework assembly welding, and transportation. The transportation of blocks between stations usually adopts transfer roller.

2. Typical Process Flow of Flat Section Production Line

Flat section is a unit structure formed by welding two or more than two steel plates into a panel, and then welding with various skeleton frameworks such as longitudinal trusses and frames. It is mainly composed of flat panel and flat frame. Therefore, its manufacture can be divided into two parts: tailor-welded panel and welding frame. According to the welding methods adopted, the automatic welding of the flat panel can be divided into double-side automatic submerged arc welding and one-side submerged arc welding with flux copper backing (FCB). In the former, the panel turning station should be added to the splicing plate assembly line so as to carry out panel turning and back sealing welding. The latter can save the plate turning station. Typical process of plane sectional production line includes: the whole line assembly and welding method (Fig. 1) and the frame assembly and welding method (Fig. 2).

In the whole line assembly and welding method, one component or a group of components is positioned one by one, installed and welded to the strake, and another component is installed and welded after the one component is installed and welded to form a flat section.

The frame assembly and welding method is to assemble and weld the longitudinal and transverse components into box frame of well shape, and then assemble and weld it to the strake.

At present, large shipyards in our country generally have flat section assembly lines. Most shipyards have introduced advanced flat section assembly lines from abroad, and some shipyards have built their own flat section production lines (Huang 2013).



Block Assembly Line, Fig. 3 TTS curved panel production lines

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Block Erection Technology

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Definition

Block erection, also known as ship assembly, is the technological stage of completing the overall assembly, welding, outfitting, and painting of the ship on the shipbuilding berth (dock) on the basis of pieces assembly and welding, block and block assembly and welding, pre-outfitting, and pre-

painting. Ship assembly is commonly known as ship general closure. As ship assembly generally needs to be carried out in the berth or the dock which are very limited and important resources of the shipyards, the quality and speed of ship assembly have a great impact on ensuring the quality of ship construction and shortening the period of ship construction (Liu et al. 2011).

Introduction

Ship Assembly Facilities for Ships: Berth and Dock

The shipbuilding berth (dock) is the working place where the block is assembled into the whole hull. It should have a solid foundation and be generally set up near the hull assembly and welding workshop and close to the launching water area of the ship, so as to shorten the transportation route of the block and facilitate the ship launching.

Shipbuilding Berth

The shipbuilding berth is a place where ships are built on land. It can be moved to the water area by launching device. Generally, it can be divided into inclined berth, horizontal berth, and semi-dock berth. The shipbuilding berth also includes a longitudinal inclined berth and a transverse inclined berth, which determine the direction of the ship's gravity launching.

The inclined berth plane and the horizontal plane have a certain inclination (called berth slope), which is generally taken as $1/14 \sim 1/24$.

Taking the longitudinal inclined berth as an example, the foundation of the longitudinally inclined berth is composed of reinforced concrete (Xu 2011). Parallel crane tracks are laid along both sides of the berth, and cranes with large lifting capacity are equipped. According to the inclination angle of the berth and the requirements of ship closure, the lifting height is required to be higher. The advantage of this kind of berth is that the ship is built and launched at the same position, and the construction site is relatively compact. Generally, there is no need to move the ship. The ship can slide into the water under the action of its own gravity, so there is no need for a special ship moving device. Inclined berth is usually used in combination with oiling and steel ball slideway, which is the most widely used at present (Fig. 1).

The Dock

A dock is a hydraulic structure that is lower than the horizontal level and has a gate at its end. After the gate is closed, the water can be drained to be used for shipbuilding. It has some advantages of horizontal berth, and the ship is also built in a horizontal state (Li et al. 2006). Moreover, the dock bottom of the building ship is lower than the ground level, the lifting height of the block is reduced, and gantry cranes with large span and large lifting mass can be configured across the dock and the pre-installed welding area on the dock side, which greatly improves the mechanization degree of ship construction. Dock launching can greatly simplify the process of ship launching and is suitable for building large

ships. In addition, the dock is not only used for shipbuilding but also convenient for ship repair. However, the investment in the dock is large, and the scale of ships built in the dock is strictly limited (Fig. 2).

General Assembly Method of Ship

Due to the different product objects and shipyard production conditions, the ship assembly methods (called construction methods) are also various. These methods are all determined according to the structural characteristics of the ship and the production conditions of the shipyard and are conducive to balancing the production load, improving the efficiency, shortening the shipbuilding cycle, and improving the working conditions.

The most commonly used methods are the block method of hull construction, tower method of hull construction, island method of hull construction, and series method of hull construction (Tokola et al. 2013).

Block Method of Hull Construction

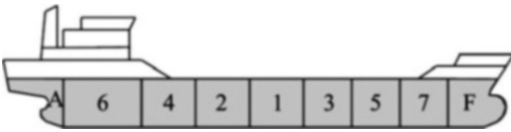
The construction method of taking block as the hull assembly unit can reduce the berth workload and welding deformation, improve the integrity of pre-outfitting and pre-painting operations, move the workload forward, and shorten the berth construction period due to the large size, good rigidity, and relatively complete space. However, the size and weight of the block are generally limited



Block Erection Technology, Fig. 1 Shipbuilding berth



Block Erection Technology, Fig. 2 The dock



Block Erection Technology, Fig. 3 Block method of hull construction

by the lifting capacity of the berth. In the block method of hull construction, the block in the middle of the ship (or close to the middle of the ship) is hoisted to the berth for positioning and fixing, and then the adjacent blocks before and after are hoisted in turn. When the butt joint welding of the two blocks is completed, the outfitting work can be carried out (Fig. 3).

Island Method of Hull Construction

The method of hull construction has only one construction area, the berth area cannot be fully utilized at the initial stage of assembly, the berth building period is long, and the stem and stern of the completed ship are tilted greatly. Therefore, if the hull is divided into 2–3 construction areas (hereinafter referred to as “islands”), each island selects a benchmark block and is constructed simultaneously according to the tower method of hull construction. The islands are connected with each other by embedded blocks, which can make full use of the berth area, expand the construction surface, and shorten the berth building period. Moreover, the length of its construction area is shorter than that of the tower method of hull construction and the hull rigidity is larger, so its

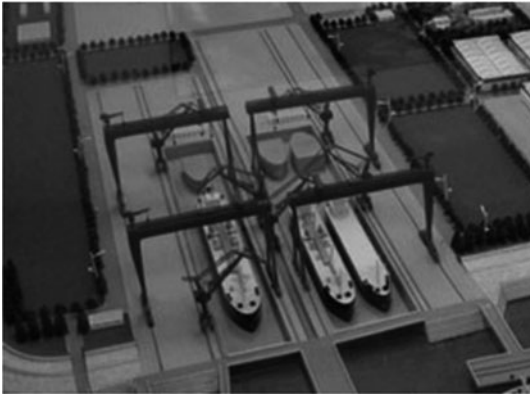
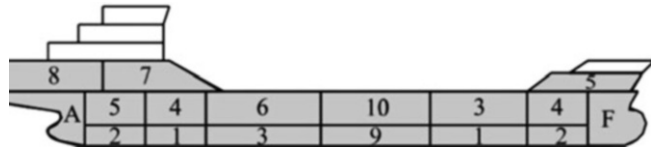
total welding deformation is smaller than that of the tower method of hull construction. This construction method is called the island method of hull construction. The hull is divided into two construction areas, which is called the two-island method of hull construction. The method of dividing into three construction areas is called the three-island method of hull construction. However, the assembly and positioning operation of its embedded blocks are relatively complicated. As shown in Fig. 4, it is a two-island method of hull construction, in which the blocks are embedded and complementary blocks (Bao et al. 2009).

Series Method of Hull Construction

When some shipyards use larger berths to build ships with smaller tonnage ships, they often use the series method of hull construction to organize production and improve the utilization rate of the berths. The series method of hull construction is to build the stern of the second ship at the stem of the berth (dock), while the first ship is built at the stern of the berth (dock). After the first ship is launched, the stern of the second ship is moved to the stern of the berth (dock), and other blocks are continuously hoisted to form the whole hull. At the same time, the stern of the third ship is built at the stem of the berth, and so on.

This form can greatly improve the utilization rate of berths, shorten the construction period of berths (docks), and carry out outfitting operations in advance. It has many advantages in improving production management and balancing

Block Erection Technology, Fig. 4 Two-island method of hull construction



Block Erection Technology, Fig. 5 Series method of hull construction

production rhythm. However, it can only be used when the length of the berth is greater than the length of the building ship (approximately 1.5 times the length of the ship), and when this method is used on the inclined berth, ship moving equipment must also be equipped. Similarly, if a large dock is used to build a small tonnage ship, and if the dock can accommodate more than one ship, many ships will be built simultaneously in the dock. When a ship needs to be launched after the final assembly, the dock door shall be opened for water inflow. Other ships that have not been completed shall be limited in position by ballast and other technical means. The launched ships shall float upward by buoyancy and be towed outside the dock. After the dock door is closed and the water is drained, other ships shall continue to be built. In actual production, there are also various construction methods. At present, on the basis of vigorously promoting pre-outfitting shipbuilding, the domestic shipbuilding method is changing toward the modern shipbuilding mode, which is guided by the theory of overall planning and optimization, applies the principle of group technology, takes intermediate products as the guidance, organizes production according to

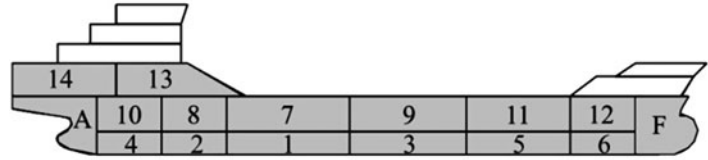
regions, separates the hull, outfitting and painting operations in space and orders in time, realizes the integration of design, production, and management, and realizes the balanced and continuous assembly shipbuilding (Kim et al. 2012). It is worth mentioning that no matter what kind of modern shipbuilding mode, in the shipbuilding process, it is based on several common methods of sectional construction that have been described, and optimizes the combination in time and space respectively, so as to achieve the purposes of expanding the production scale, improving the production efficiency and reducing the manufacturing cost (Fig. 5).

Tower Method of Hull Construction

During construction, the bottom block at the back of the middle part is taken as the reference block (for medium-sized ships, the engine room block can also be taken). The reference block is positioned and fixed on the berth, and then each block is hoisted from bottom to top to the stern and stern and both sides. Since the installation area formed during the construction process is in the shape of a pagoda with wide bottom and narrow top, it is called the tower method of hull construction, as

Block Erection Technology,

Fig. 6 Tower method of hull construction



B

shown in Fig. 6. The sectional installation of this construction method is relatively simple, but the berth building period is relatively long, and due to the large number of welded closing joints, the welding deformation is not easy to control, and the stern and stem of the ship are seriously tilted after completion.

General Assembly Technology of the Berth (Dock)

Tower method of hull construction and island method of hull construction are commonly used for ship assembly. Although the construction methods are different, the sequence of sectional hoisting and the fixing method of sectional positioning in a construction area are the same. At the same time, through the discussion and analysis of the block assembly technology, the block assembly technology of the block construction method can also be obtained. Therefore, taking the tower method of hull construction as an example, the relevant issues of berth assembly and welding sequence and welding process of berth assembly are discussed.

Sequence of Berth Assembly and Berth Welding

In general, the sequence of berth assembly and berth welding for the tower method of hull construction is as follows:

1. Hoisting reference block.

The hoisting reference block is the starting point for berth closure, and its selection should enable berth outfitting and hull construction to be completed at the same time. Due to the heavy outfitting workload of the engine room, the starting point is often selected in and

around the engine room, so as to complete the hull of the engine room as soon as possible and carry out outfitting operations as soon as possible.

2. Bulkhead block and front and rear bottom block on the hoisting reference block.
3. Hoist broadside block. Continue to hoist the bottom block and bulkhead block in the stern and the stem directions.
4. Hoist the deck block. Continue to hoist the bottom block, bulkhead block, and broadside block, and sectional seam welding is carried out at the part where the mega-block of the ship hull has been formed.
5. Continue to hoist the bottom block, bulkhead block, broadside block, and deck block in the stern and stem directions, weld the assembled block large joints, and carry out outfitting operation on the cabins where the block large joints have been welded.
6. Hoist the stern and stem block, and continue to complete the welding of large seams in blocks and outfitting in the cabin.
7. Hoist and weld superstructure, and continue outfitting operation.

When the island method of hull construction is adopted, the assembly and welding sequence of each construction area are the same as that of the tower method of hull construction, except that the work of hoisting, embedding, and repairing blocks is added at the end to connect each construction area. In actual production, sometimes due to the influence of sectional supply, reasonable utilization of hoisting equipment, and other temporary factors, it is required to adjust the sequence of hoisting in certain blocks. In this regard, it is necessary to proceed from reality. As long as it does not cause difficulties in berth assembly and welding, appropriate adjustments can be allowed (Huang 2013).

Ship Berth Assembly and Welding Technology

Sectional positioning, allowance drawing and cutting, large joint fixing requirements, and large joint welding are common technical problems in berth assembly and welding, which are discussed and analyzed respectively.

Positioning of Blocks on the Berth

1. Primary Positioning.

The correct position of the blocks in the hull is determined directly by the four factors of the length, width, height direction, and horizontal inspection line of the blocks in the hull. Take the positioning of double bottom blocks as an example, the frame inspection lines on the blocks are aligned with the corresponding frame lines on the berth to determine the position of the blocks along the length direction; align the block centerline with the berth channel centerline to determine the position in the width direction; determine the position of the sectional height direction based on the height benchmark; measure the block left-right level inspection line to determine the block left-right level. After the positioning is completed, the blocks will be fixed.

2. Secondary Positioning.

If the blocks are assembled on the berth with the allowance, a secondary positioning is often required. Take the bulwark block with allowance on the berth as an example. First of all, the blocks are hoisted onto the berth. And align the horizontal inspection line of the block with the same horizontal inspection line on the previous side block that has been assembled. Draw the allowance line in the height direction, remove the broadside block and cut off the allowance, then hoist the block onto the berth, align the frame inspection line of the block with the same frame inspection line on the bottom block, draw the allowance line in the length direction, and remove and cut off the allowance. At last, hoist the block onto the berth, complete the block according to the frame inspection line and the horizontal inspection line, and finally complete the secondary positioning.

Determine the Allowance, Draw the Allowance Line, and Cut the Allowance

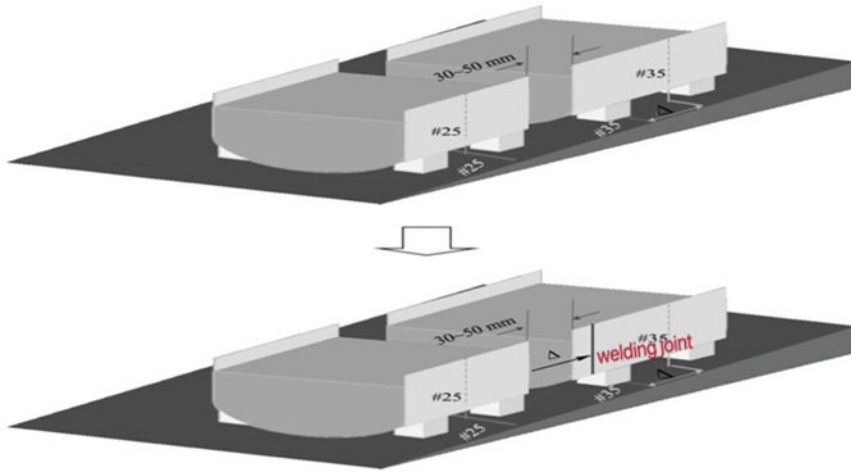
In the shipyards where precision control is implemented, the assembly is carried out on the berth without any allowance in blocks. The allowance of the large joints in blocks is cut off on the sectional mold bed and then hoisted on the berth. This can cancel the operations of drawing lines, cutting and secondary positioning of the block, reduce the amount and difficulty of berth assembly operations, and shorten the assembly period of the berth. However, due to the limitation of technical conditions, some shipyards have not yet implemented the precision control of berth assembly, so the sectional large joints need to have an allowance, and the allowance drawing and cutting will be carried out when the ship is closed.

Sectional Drawing, Butt and Positioning Welding

After cutting the block allowance, the groove shall be correctly processed, then the block shall be drawn in, and the block installation position and joint clearance shall be checked. After meeting the requirements of positioning welding, the block positioning welding shall be carried out, and the internal framework near the large joint shall be installed. First, the internal framework of the block is positioned and welded, and then the external plate is positioned and welded.

In the method of sectional drawing and matching, the bottom block and the block are generally supported by a hydraulic jack and are drawn and positioned by elastic screw buckle (also known as flower basket screw). The other blocks are drawn and matched with elastic spiral buckles.

When positioning welding is carried out for sectional joints, the phenomenon that two frameworks are not aligned often occurs. At this time, the positioning welding between one framework and the plate can be removed by about one gear frame distance, or the positioning welding between the two butted frameworks and the plate can be removed to make them straight or aligned (Fig. 7).



Block Erection Technology, Fig. 7 Cutting the block allowance

Welding Work in Berth Assembly

In order to prevent welding deformation, a strong back should be added after positioning welding of the outer plate, and welding should be carried out symmetrically left and right. Before welding, several blocks should be assembled to increase the rigidity of the hull and make it difficult to generate welding deformation. During welding, the inner surface shall be welded first, and the outer back sealing shall be welded.

The welding operation of sectional large joints is carried out in parallel with the sectional hoisting operation. This is not only conducive to reasonably organizing the production process of berth assembly and welding but also can reduce the warping deformation of the hull when the length of the construction area is short.

In order to reduce the total welding deformation of the hull, the welding operation should be carried out in the area where the mega-block has been formed, so that it has strong structural rigidity of the hull and is convenient to meet the requirements of symmetrical welding.

In recent years, with the continuous improvement of shipbuilding equipment automation, some shipyards have begun to use vertical welding robots in the welding of large hull joints, greatly improving the speed and quality of berth closure seam welding.

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Blowout Preventer (BOP)

- [Christmas Tree](#)
- [Hydrodynamics for Subsea Systems](#)
- [Steel Pipelines and Risers](#)

Bridge System

► Ship Navigation System

Brilliance of the Seas

► Luxury Cruises

Broaching

► Surf-Riding and Broaching

Broaching Stability

► Surf-Riding and Broaching

Brushless DC Motor (BLDCM)

► AUV/ROV/HOV Propulsion System

Bucket Foundations

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Conception of Bucket Foundations

The world's escalating demands for energy, combined with the continued depletion of oil and gas reserves in shallow waters, has resulted in offshore developments moving into deeper waters and untested environments. In the Gulf of Mexico, West Africa, offshore Brazil, and, more recently, off Australia, developments have proceeded into water depths in excess of 1000 m. These deep

water developments usually consist of moored floating facilities, which are tethered to the seabed via an anchoring system.

Bucket foundations (also named as suction caissons or skirted foundation) have been proven as a cost-effective alternative to more traditional anchoring solutions such as piles and drag anchors, because of their ability to confine surface soil and transfer loads to stronger soils at skirt tip level. Bucket foundations are a type of shore shallow foundations, which consist of a plate resting on the seabed and a peripheral skirt (often supplemented by internal skirts) penetrating into the seabed and confining a soil plug. Bucket foundations are best described as upturned foundations that are installed into foundations to anchor floating structures or to support superstructures. The skirts also assist in foundation installation by constraining the lateral movement as the foundation approaches the seabed, compensating for seabed irregularities and reducing scour around the foundation periphery. Bucket foundations (also called buckets, skirted foundations, or suction anchor piles) can be used to sustain high axial and lateral loads. These foundations may be broadly divided into two types; relatively short buckets, often referred to as bucket foundations, which support jacket structures, and relatively long buckets, which are used as foundations for moored structures, which typically have higher aspect ratio than bucket foundations. The typical shapes of bucket foundations are circular and square. For square buckets, the width of edges usually ranges from 3 m to 10 m (Hossain et al. 2012). For buckets with circular shape, it comprise large diameter cylinders, typically in the range of 3–8 m (shown in Fig. 1), open at the bottom and closed at the top, and generally with a length to diameter ratio L/D in the range three to six, considerably less than offshore piles, which have slenderness ratios up to 60. Bucket foundations are a comparatively new foundation system, which have however found relatively widespread use. Bucket foundations have proven to be a cost-effective, excellent performing, deep foundation system, that is utilized in offshore engineering worldwide. Typically, they are used for bridges, offshore wind power, cofferdam, and large structures, where large loads and lateral resistance are

major factors. The first bucket foundation employing the system as permanent foundations in clay or sand were built in the early 1990s (Tjelta et al. 1990).

The foundation keeps water out of the construction area while its open bottom allows workers to place foundations and piers in the seabed or riverbed. In shallow water, an open bucket is used; its open top allows light and air to enter from above the water line. For deep-water construction, a pneumatic bucket has a closed top; pressurized air is pumped in, and personnel enter and leave through an airlock. Either type has a sharply inclined lower edge, allowing the structure to be deeply embedded in the ground so water cannot seep in.

A bucket must be firmly mounted on a stable foundation for maximum security and efficiency.



Bucket Foundations, Fig. 1 Bucket foundation used in offshore engineering. (After Dendani and Colliat 2002)

Engineers prefer to choose a layer of stiff soil to construct on it. If bedrock is too far beneath the ground, they will sometimes rely on impacted mud or earth, or create an artificial foundation.

Bucket foundations are similar in form to pile foundations, but are installed using a different method. It is used when soil of adequate bearing strength is found below surface layers of weak materials such as fill or peat. It is a form of deep foundation, which are constructed above ground level, then sunk to the required level by excavating or dredging material from within the foundation.

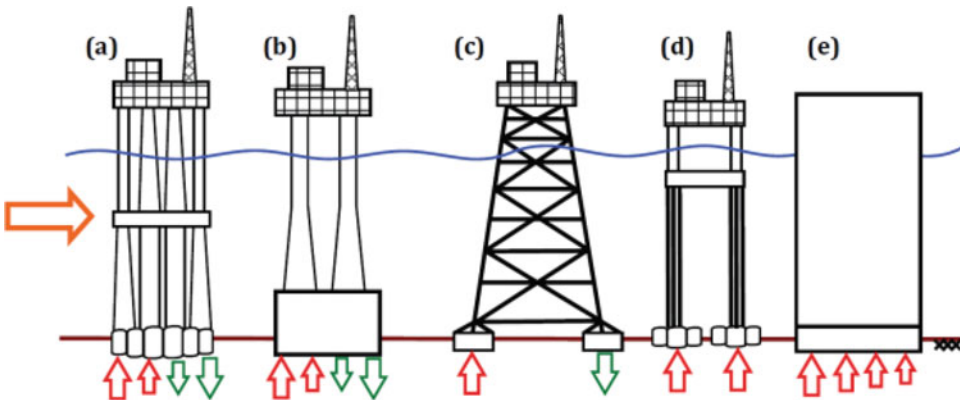
Functions of Bucket Foundation

The application of bucket foundation and loading condition in various offshore structure are shown in Fig. 2, which includes (a) & (b) gravity-based structures; (c) jacket/template structure; (d) tension leg platform; (e) storage tank

Support Vertical Loads

Bucket foundations have proven to be competitive alternatives to more traditional foundation solutions like piles in various types of soils and for a wide range of fixed offshore platforms. Bucket foundations have been used for gravity platform jackets, jack-ups, subsea systems, and seabed protection structures.

Even more critical than settlement is differential settlement. This occurs when parts of the offshore structures settle at different rates,



Bucket Foundations, Fig. 2 Application of bucket foundations and loading conditions in various offshore structures. (After Acosta-Martinez 2010)

resulting in cracks, some of which may affect the structural integrity of the offshore structure.

Support Horizontal Loads

Bucket foundations are used to anchor floating production vessels using catenary or taut wire mooring chains; in these cases, the foundation is under quasi-horizontal or angled load. In shallow water, a length of the bucket foundation chain will rest on the seabed during calm conditions, with the small mooring loads being resisted by friction against the seabed. The chain may be connected to the foundation at the soil surface, or it may be connected via an embedded pay-eye, which provides a more efficient arrangement.

Types of Bucket Foundations

Box Bucket Foundation

Box bucket foundations are watertight boxes that are constructed of heavy timbers and open at the top. They are generally floated to the appropriate location and then sunk into place with a masonry pier within it.

Excavated Bucket Foundation

Excavated bucket foundations are, just as the name suggests, foundations that are placed within an excavated site. These are usually cylindrical in shape and then back filled with concrete.

Floating Bucket Foundation

Floating bucket foundations are also known as floating docks and are prefabricated boxes that have cylindrical cavities.

Open Bucket Foundation

Open bucket foundations are small cofferdams that are placed and then pumped dry and filled with concrete. These are generally used in the formation of a pier.

Pneumatic Bucket Foundation

Pneumatic bucket foundations are large watertight boxes or cylinders that are mainly used for under water construction.

Mechanical Principle of Bucket Foundation

Application of Bucket Foundation

Foundation for Offshore Wind Turbines

Bucket foundations have been widely used in offshore wind turbines in recent decades. The foundation of a wind turbine must be strong enough to withstand the maximum force acting on it and not fail during its operation. Furthermore, it must be stiff enough so that the displacements and rotations during its lifespan fall within the serviceability design criteria of the wind turbine manufactures. Bucket foundations, with high vertical and horizontal bearing capacities, have been used as the foundation for offshore wind turbines (Ibsen and Brincker 2004). A foundation concept that is gaining acceptance for the support of bottom-fixed offshore wind turbines is the suction bucket foundation. This may support the turbine superstructure either as an individual bucket foundation or as a group of three or four bucket foundations below a jacket substructure, as shown in Fig. 3

Bucket Foundations Construction Process

Suction

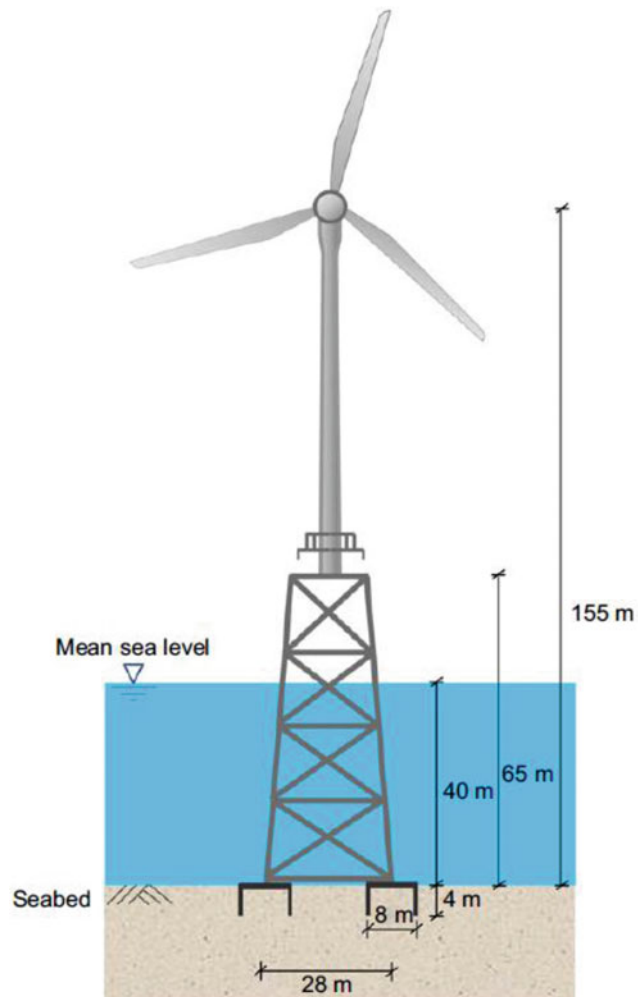
A suction bucket foundation is installed pumping the water inside of the foundation to install it by the differential pressure inside and outside as shown in Fig. 4. Initially, bucket foundation was installed to a certain depth by the weight of itself. And then, penetration into the seabed is achieved by suction, via a pump connected to a valve in the top cap of the foundation. Water is pumped out from inside the foundation causing the pressure inside the foundation to fall below that outside. This causes a net downward pressure on the top of the foundation forcing it into the seabed.

Jacking

Offshore developments moving beyond the immediate continental shelf into deeper waters (now approaching 3000 m depths) has been driven by the vibrant oil and gas industry and the world's ever-increasing demands for energy.

Bucket Foundations,

Fig. 3 Offshore wind turbine with jacket substructure supported by suction bucket foundations.
(After Bienen et al. 2018)



B

These deep water developments rely on floating facilities (e.g., floating production storage and offloading vessels, tension leg platforms, SPAR platforms, and emerging concepts such as floating liquefied natural gas (FLNG) facilities) moored to the seabed through mooring chains and anchoring systems, with suction bucket foundations being identified as the most viable option. Bucket foundations are also used as foundations to support pipeline manifolds and end terminations, subsea structures, and riser towers. In the renewable energy industry, they are increasingly being considered for anchoring floating turbines.

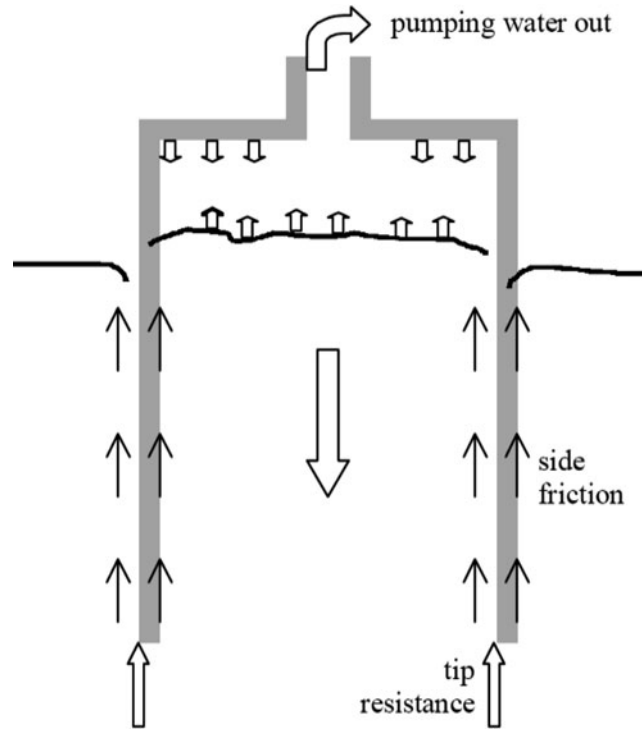
Suction bucket foundations are installed by pumping water from inside the foundation after it is allowed to penetrate under its self-weight.

Analyses are sometimes carried out using jacking installation process to simplify the problem, especially for bucket foundations in clay, where a foundation is pushed in soil up to the full penetration depth (similar to driven pile).

To comply with the increasing size of the floating facilities to be anchored (e.g., the Prelude FLNG is 488 m long and 75 m wide), suction bucket foundations are designed as longer and wider – currently up to 30 m long, with a length to diameter (aspect) ratio L/D in the range 2–7 (Andersen et al. 2005). As the thickness of the skirt (t) is restricted to less than 50 mm to ensure installation viability, the longer bucket foundations are required to include horizontal ring stiffeners at intervals along the inner wall of the thin skirt with local thickening of the wall in the

Bucket Foundations,

Fig. 4 Bucket foundation installation by suction pressure. (After Andersen and Jostad 1999)



vicinity of the padeye, with or without transverse struts, for structural integrity.

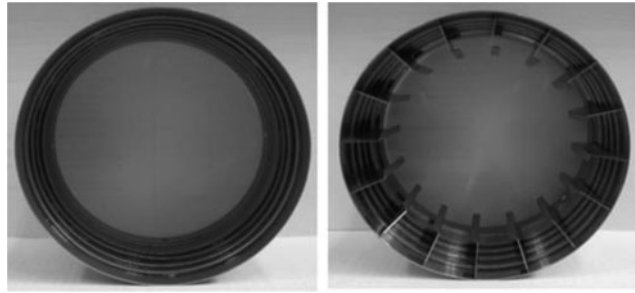
Suction bucket foundations are thin-walled (thickness, $t \leq 50$ mm) and thus prone to buckling during installation or distortion under the horizontal or moment loading. In order to deal with the buckling problems, horizontal ring stiffeners at intervals along the inner wall of the thin skirt are employed with local thickening of the wall in the vicinity of the padeye, with or without transverse struts. As such, horizontal ring stiffeners at intervals along the inner side of the thin skirt, and occasionally vertical stiffeners around the internal periphery, are employed with local thickening of the wall in the vicinity of the padeye, with or without transverse struts, as shown in Fig. 5. Recent designs of rectangular steel foundations for skirted gravity-based platforms have required combined vertical and horizontal stiffening (after Watson and Humpheson 2007). Consequently, Engineers are faced with the challenge of designing foundations with sufficient structural integrity to allow longer skirt lengths while ensuring their installation feasibility. These stiffeners may increase the required installation pressure by as

much as 50–100% and can have a significant influence on the drained tension capacity (House and Randolph 2001; Westgate et al. 2009).

The addition of these stiffeners has created significant uncertainties regarding the soil flow mechanisms, in particular the inner soil heave with the risk of potential penetration refusal prior to reaching the designed installation depth (or achieving the designed capacity to sustain operational loadings). The pattern of soil flow at the foundation tip, and the proportion of the bucket foundation wall that is accommodated by inward or outward displacement of the soil, has important consequences for quantifying (i) the external radial stress and excess pore pressure, and ultimately long-term external shaft friction following consolidation; (ii) the internal side friction and stiffener end bearing. The behavior of the clay plug can also affect the maximum penetration depth of the foundation. This is more critical for stiffened bucket foundation. If the plug remains fully or partially self-supporting above the horizontal stiffeners, the gaps formed between the stiffeners result in greater heave volume, and hence higher inner seabed elevation. This entry

Bucket Foundations,

Fig. 5 Bucket foundation with stiffeners. (After Westgate et al. 2009)



B

has specifically focused on the quantification of inner soil heave during installation of stiffened bucket foundations.

Andersen et al. (2005) discussed predictions for four different hypothetical installation cases and six case histories carried out by four predictors using their normal design method. For the hypothetical cases, the predictors calculated different soil heave height inside bucket foundation due to different assumptions in terms of the proportion of soil flow inside bucket foundation, soil plug heave standing ability, and soil infilling in the gaps between the embedded stiffeners. For the case histories presented, comparison between the calculated and observed soil heave showed that soil infilling in the gaps between the embedded stiffeners dictated the soil heave height, with the assumption of no soil infilling in the gaps and fully filled gaps providing over and under predictions, respectively.

The stiffeners have a significant effect on the soil flow mechanism of stiffened bucket foundation installation (Zhou et al. 2016). They found that the soft soil will be trapped between stiffeners and move vertically with the same velocity as foundation installation, and the stiffeners have effect on the soil movement around skirt tip and stiffeners. Consequently, it also has effect on the inside heave as reported by Zhao et al. 2017.

Advantages and Disadvantages of Bucket Foundation

Advantages of Bucket Foundations

- Economics
- Reduced weight
- Increased installation speeds
- Ease of decommissioning

- Suitable for deep waters and large turbines
- Slightly less noise and reduced vibrations
- Easily adaptable to varying site conditions
- High axial and lateral loading capacity

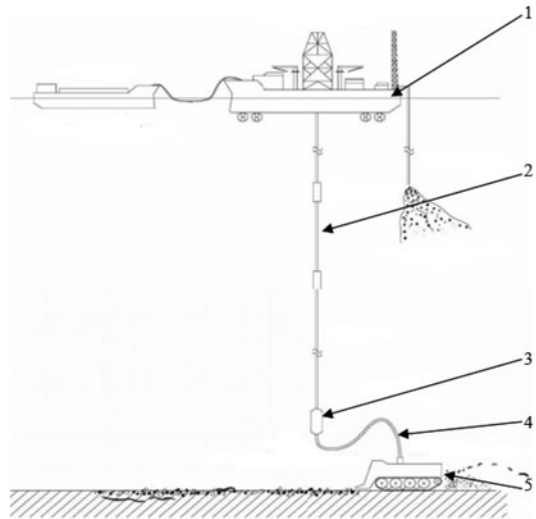
Disadvantages of Bucket Foundations

- Extremely sensitive to construction procedures
- Not good for contaminated sites
- Lack of construction expertise
- Lack of Qualified Inspectors

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Buffer Station, Fig. 1 The ocean mining system. (1) The surface ship; (2) The lifting riser; (3) The buffer station; (4) The flexible riser; (5) The miner robot

Buffer Station

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Synonyms

DCOR – dynamic characteristics of the riser;
EIP – equipment installation platform

Introduction

The basic function of the ocean mining system is to collect the mineral resources of the seafloor, lift them to the sea surface, and transfer them to the port. It is generally composed of four subsystems such as the undersea mining system, the lifting system, the control and power system, and the surface supporting system. The lifting system is the intermediate link between the seafloor mining

system and the surface supporting system. It is the channel to transfer the polymetallic nodules from the seafloor to the surface ship. Besides, it also acts as the carrier of power and signal cables and the heart of the ocean mining system (Fig. 1).

The buffer station is located in the middle of the lifting system. The upper end of the buffer station is connected to the deep-sea mining ship through the rigid riser, and the lower end is connected to the mining vehicle through the flexible riser.

Research Process

For deep-sea mining, many research institutions have done a lot of work in recent decades.

In March 1979, several international mining consortia countries, led by the United States, including Canada, the United Kingdom, West Germany, Belgium, the Netherlands, Italy, and Japan, etc., first adopted a deep-sea mining system with a buffer station to collect nodules from 5000 m seabed. Russia began to develop the deep-sea mining system with a buffer station in 1980. In 1991, the system design was completed and a 1/10 prototype test was carried out in 100–1000 m water depth of the Black

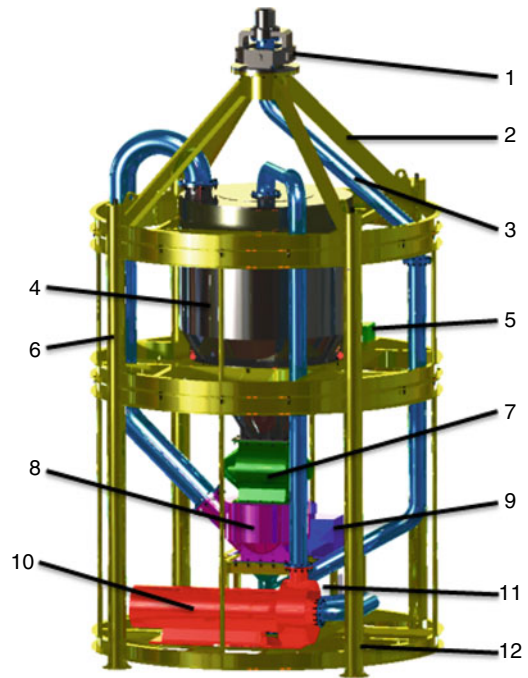
Sea (Wang 2015). Korea has been engaged in the development and research of deep-sea mineral resources since the 1990s. The conceptual design of the riser lifting system of the subsea collector with a buffer station and the structural design of the hydraulic mechanical composite ore collecting system with the lifting system were completed (Yoon 2011). In 2013, the system was tested in 1370 m deep-sea area, 130 km southeast of Puxiang in Qingshang North Road. Canada used a diaphragm positive displacement pump system as the underwater mineral lift pump, which plays a buffer role in the whole deep-sea mining lifting system (Smith 2011). The pump system consists of two pump modules, and each module includes five diaphragm pumps driven by the return water of mineral dehydration from mining vessels. In 2012, the underwater mineral lift pump was put into operation factory test evaluation, and some components were tested in 2500 m of water.

Chinese research institutions carried out basic research on deep-sea mining during “the 8th Five-Year Plan” period and proposed a hydraulic lifting deep-sea mining system with buffer function (Tang et al. 2013). An elastic blade wheel feeder was added to the buffer station during the “9th Five-Year Plan” period, which solved the problems of feeding jam, ore bin arching, and emergency discharging. In the “13th Five-Year Plan,” the buffer station of the deep-sea mining mineral lifting system was planned to complete the 1000 m sea trial in 2021 (Yang et al. 2020).

Buffer Station Component

Overview

The buffer station mainly includes an external frame, an internal equipment system, and a riser system. The external frame includes a swing connecting device, top frame, upper frame body, and the lower frame body. The internal equipment includes a storage bunker, hose pump, constant feeder, hydraulic pressure station, electronic lifting cabin, and power distribution box, etc. (Fig. 2).



Buffer Station, Fig. 2 Buffer station component system. (1) Swing connecting device; (2) top frame; (3) swing hose; (4) storage bunker; (5) power distribution box; (6) upper storage bunker; (7) constant feeder; (8) mixer; (9) hydraulic pressure station; (10) hose pump; (11) counterweight; (12) lower frame

Storage Bunker

The volume of the storage bunker is generally determined by the principle that it can continuously feed ores by the bunker if there are no nodules in the collection path in 15 min. The top of the bin is open and the lower part is conical, which is conducive to ores discharge and prevents ores from piling up.

The Constant Feeder

The constant feeder adopts an elastic blade feeder, which is composed of a shell, a rotor, a rotor support end cover, a hydraulic pressure motor, and a speed measuring device. The shell is processed by connecting the flange, the side plate, and the cylinder body. Its feed inlet is asymmetric and biased to one side of the feed port, to reduce the resistance moment of feeding. An arc-shaped baffle plate is fixed at the outlet to make the feed out. The feed port has a parallelogram shape so

that the material falls uniformly and continuously. The constant feeder can solve the blocking problem and achieve uniform feeding. The feeding fluctuation of ores is less than 5%.

Swing Connection Device

It is located at the top of the buffer station and connects the buffer station body to the rigid riser. It adopts the movable hinged form, which releases the swing of the buffer station body. It can optimize the stress condition of the riser system and improve the stability of the whole system. The flexible riser passes through the middle of the swing connection device and connects the buffer station to the upper rigid riser.

External Frame

The unsealed frame structure is adopted to protect the internal equipment and prevent the impact when placing and recycling. To facilitate the installation and maintenance, the external frame is modularly designed. It includes a top frame, an upper frame, and a bottom frame.

The Hose Pump

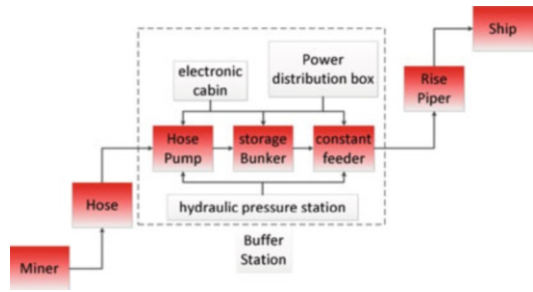
The ores are pumped from the deep-sea mining vehicle to the storage bunker of the buffer station through the flexible riser. The hose pump should have a robust characteristic.

The Function

The main functions are as follows.

The ores pretreatment function. The deep-sea ores collected by the mining machine are delivered to the buffer station's storage bunker. Then the ores are continuously and quantitatively feeds to the lifting pipe by a constant feeder. This will avoid the influence of ore concentration caused by the change of nodule abundance on the operation of the underwater transportation system. That will ensure the stability of the ore lifting process parameters and high lifting efficiency.

Providing EIP for mining underwater systems. For example, it can be used as a reference settlement unit of the acoustic positioning system.



Buffer Station, Fig. 3 Mining ore transportation procedure

The hydrodynamic optimization function.

The buffer station helps to keep the riser vertical and improves the DCOP. If the weight and shape are designed reasonably, the vertical displacement and axial stress of the riser system caused by ship heave may be reduced by 50% and 17%, respectively, according to the model test (Fig. 3).

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Buoyancy

- [AUV/ROV/HOV Hydrostatics](#)

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Cable

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Synonyms

Bare wire; Conductor; Insulation layer; Polyester rope; Power cable; Winding wire

Definition

An electric cable is usually made up of several or sets of wires twisted together. There is insulation between wires, with a layer of highly insulated cover enclosed outside. There are also shield and armor in some cables. Electric cables can be classified into the power cable and the communication cable. The former one is mainly used to provide electric power, and the latter one is mainly used

for transmission of power, control of electricity, and signal transmission (Limei 2016). Insulation is used to provide dielectric strength and electric insulation. There are sheathes in all cables which can protect the cables, making the cables round and adjusting the size and weight. Shield is mainly used for the function of shielding magnetic field. And armor plays an important role in strengthening (Hammons 2003).

Besides electric cable, polyester rope has been put into usage as part of mooring lines for two decades and more frequently used in deeper water globally (Mukoyama et al. 2006). Polyester cable is widely used in the mooring system of deep water due to its high strength, light weight, and good anti-fatigue performance, which is mainly consist of polyester, polypropylene, nylon, manila hemp, Dyneema, etc (Honjo 2001).

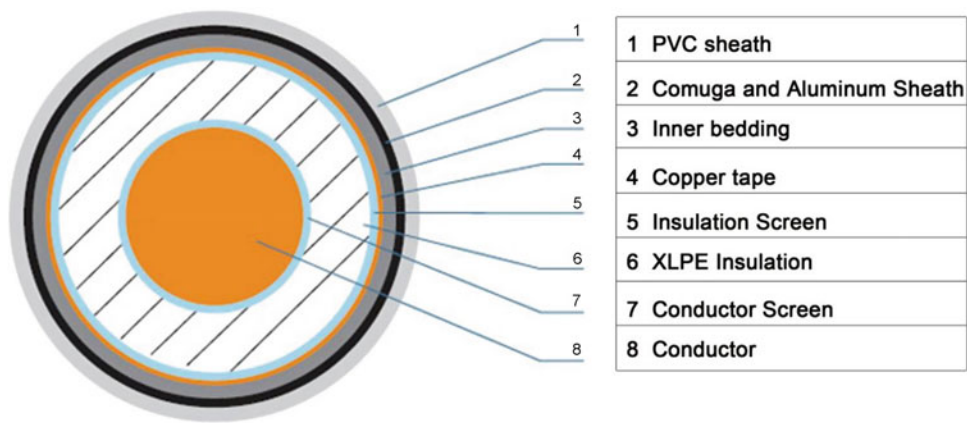
Scientific Fundamentals

Structure Composition of Electric Cable

This entry takes a typical structure of cable as an example, which contains conductor, insulation, sheath, and armor (Schmidt 2012). The effects they make in the cable are as follows (Fig. 1).

Conductor

Conductor is a component of conducting electric current, which is a vitally important core component in the structure of the electric cable and



Cable, Fig. 1 The structure of electric cable (http://www.orientcable.com/v_pros.asp?/5.html)

which determines the basic performance (Zuijderduin 2012). The specifications of the wires and cables are up to the cross section of the conductors, which are mainly made of high conductivity materials (copper or aluminum). According to the operating conditions of the required flexibility, each core may be twisted by a single wire or multiple wires of the electric cables (Zhong 2015). In general, bare copper wire, tin-plated copper wire, silver-plated copper wire, tin-copper alloy, enameled wire, copper wire, copper clad steel, and so on are the common conductors in the market (Xuyang 2017).

Insulation Layer

Insulation layer is used as the insulation material for electric cables. According to different requirements of the withstand voltage conditions, insulation materials are covered outside the conductors with different degree of thicknesses (Ouatah et al. 2013). Insulation layer usually has excellent insulation performance and heat resistance. Generally, the higher the voltage is, the thicker the insulation layer is, but it is not proportional. Usually, the insulation materials used in cables are oil-immersed paper, polyvinyl chloride, polyethylene, cross-linked polyethylene, rubber, and so on (Zhang 2015). Cables are often classified by insulation materials, namely, the oil-immersed paper-insulated cables, the PVC cables, and the cross-linked polyethylene cables.

Seal Sheath

Seal sheath can protect the wire core from damage caused by general external factors or other environmental factors, including machinery, water, moisture, chemicals, and light. Due to the insulation which is susceptible to moisture, the lead or aluminum extrusion seal sheath is generally used (Ohsaki et al. 2012).

Armor Layer

As a part of protecting electric cables, armor layer is used to protect seal sheath from mechanical damage. In general, galvanized steel tapes, steel wires, or copper belts and copper wires are used as armor wrapped around the sheath (so it is called armor cable). The armor layer can shield the electric field and can prevent from external electromagnetic interference (Dvorsky et al. 2011). In order to avoid the corrosion of steel tape and steel wire by the surrounding media, they are usually coated with asphalt or wrapped around a jute-immersed layer or extrusion polyethylene and PVC sheath.

Classification

Wires and cables can be generally classified into the following five categories according to different actual engineering uses.

Bare Wire

There are only conductors in bare wires, which have no insulation. Bare wire can be classified

into copper, aluminum and other metals, composite metal round-single wire and stranded wire, soft wiring, and transmission lines of various structures (Courty and Garo 2017).

Power Cable

Power cable is a wire product used in transmission and distribution of high power energy in the main line of power system, which includes power cables that have various kinds of voltage from 1 to 500 or above and various kinds of insulation, such as overhead bare wires, power cables (plastic power cables, oiled paper power cables which are almost replaced by the plastic power cables, rubber sheathed cables, overhead insulated cables), branch cables, electromagnetic wires, electrical equipment cables for electric power equipments, and so on (Grayson et al. 2001).

Wires and Cables for Electrical Equipments

There are various power cables whose wires and cables are for electrical equipments and which can transmit power directly from the distribution point of the power system to different electrical equipments. These cables are also applied in the electrical installation lines and control signal for kinds of industry and agriculture (Chen et al. 2012). The structures and performances should be determined in combination with the characteristics of the equipment and the environmental conditions, due to the extensive uses and different types of these cables.

Communication Cable and Optical Fiber Cable

Communication cable is used to transmit data and other telecommunication information in telephone, telegraph, television, radio, and fax. There are many kinds of wires and cables which are used in the information transmission system, such as telephone cables, television cables, electronic cables, radio-frequency cables, optical fiber cables, data cables, electromagnetic wires, power communication, and other composite cables.

Winding Wire

Winding wire is a conductive metallic wire with an insulation layer, which is used to produce coil or winding used for electrical products. Its

function is to generate a magnetic field by electric current or generate an inductive electric current by cutting the magnetic force line, which realizes the mutual conversion of electric and magnetic energy. That is why it is also called an electromagnetic wire (Bicen 2017).

Basic Property

As is known to all, breakdown often occurs during the electric cable operation. The so-called cable breakdown is the short circuit between the core wire of the inner conductor layer and the metal net on the ground under the high-voltage field after the insulation layer of the cable is punctured. And the short circuit of high-voltage secondary coil is formed on the circuit, which increases the secondary current. Cable breakdown is quite harmful, which will burn the cable directly, leading to serious economic loss and threatening production seriously. Many reasons contribute to the breakdown of cables in operation, and the main reasons are the reduction of cable performance and the damage caused by external forces. Therefore, the basic property indicators of cable are summarized as follows.

Shielding Property

In the process of signal transmission, unshielded cable has antenna effect, which not only radiates signals outward but also receives external signals. It is likely to form electromagnetic interference to the signal, which leads to the communication failure, increasing noise, signal error, and other influences. Therefore, cable itself has a certain shielding performance, which is very important in the process of cable operation. Shielding attenuation and transfer impedance are two important indexes to measure the shielding performance of cables. Shielding attenuation is based on electromagnetic field theory which reflects the shielding performances of cables directly. Transfer impedance is based on circuit theory which reflects the shielding performance of cables indirectly. Besides, the structure of conductor has a certain influence on the shielding performance of the cable, which can be enhanced by wearing the outer conductor or the smooth aluminum pipe outer conductor (Chen et al. 2017).

Voltage

Since electric cables can be used in power systems and signal transmission systems, the voltage of electric cables usually includes the voltage of power and signal cable. Apparently, the voltage of power cable is the source of power for the cable, and the voltage of signal cable can be used to transmit signals. There are three important indexes which can be used to measure the voltage of power cable and signal cable. The first one is the nominal power frequency voltage(U_0), which is between conductor and the ground or metal shield for cable design. Another one is the nominal power frequency voltage(U), which is between conductors for cable design. And the third one is the maximum of the maximum system voltage (U_m) in which the device can withstand. According to the ISO 13628-5/API Spec 17E, the voltages of electric cables are as follows:

For power cable, the selection of power cable voltage level ranges from 0 V to the nominal power frequency voltage.

$$\frac{U_0}{U(U_m)} = 3.6/6(7.2)kV(rms) \quad (1)$$

For signal cable voltage, the selection of signal cable voltage level ranges from 0 V to the nominal power frequency voltage.

$$\frac{U_0}{U(U_m)} = 0.6/1.0(1.2)kV(rms) \quad (2)$$

Electric Current

Electric current is the electricity flow that a cable line can be up to when transmitting power. Electric current must flow through the cable line in order to transmit electricity. The more currents flow through the line, the more power it transmits. The cable load flow can be called the cable long-term allowable load flow when the temperature of conductor reaches the long-term allowable working temperature under thermal stability conditions. As for the working conditions, the maximum load current which is required by the load must be less than the long-term allowable load flow of the conductor in the air. In accordance

with the relevant specific provisions, the maximum allowable load flow for each type of wire can be obtained. The property of conductor is the internal factor that affects the cable load flow, which can be increased by extending the area of the wire cores; using high conductive materials, insulation materials with good high-temperature resistance, and thermal conductivity; and reducing contact resistance. The environment is the external factor that affects the cable load flow, which can be increased by the reasonable increase of the wire spacing and the choosing of the appropriate place for laying.

Attenuation

Attenuation refers to the signal attenuation along the length of the cable. When signals pass through the cable, some of the signals can be converted to heat, and some of the signals will leak out of the cable through the external conductor. In most applications, there is the purpose of minimizing the attenuation or keeping it within budget. In general, the larger the diameter of the cable is, the less attenuation of the cable is. In another word, by increasing the size of cable, the attenuation can be reduced. In addition, the higher the temperature is, the more the attenuation is.

Direct Current Resistance

Direct current resistance is an important parameter in cable and all aspects of testing, which determines the quality level of the cable. To make it specific, it is possible to judge whether the cross-sectional area of conductor is consistent with the manufacturing specifications, whether the total conductivity of cable is qualified, and whether the conductor is broken.

Insulation Resistance

Insulation resistance is the most basic insulation index for electrical equipments and circuits. The greater the insulation value is, the less the current is, and the higher the insulation performance index is. General electric conductor has a small resistance value of only a few omega. However, the value of insulation resistance is relatively greater, at least dozens MΩ or above. The value

of the insulation resistance varies due to the changes of the temperature and the length of cable. Therefore, it is calculated with the electric resistance converted to 20°C per unit length generally for comparison (Dian 2016).

Mechanical Property

In general, the properties, such as tensile strength, elongation, bending, flexibility, softness, shock resistance, wear resistance, and mechanical impact resistance, are the main mechanical properties. If the mechanical property cannot meet the actual working strength, the electric cable will be damaged.

Polyester Rope

Generally, there are three layers in a polyester rope with the outer layer as a braided protective layer, which is wear-resistant and protects the core rope. The secondary outer layer is a filter layer, which prevents particles of 3–5 μm or larger from entering the cable to cause wear and tear. The inner layer is a polyester core rope, which is subjected to tensile force. In addition, there can be two protective layers in the outer layer and two filter layers in the secondary outer layer according to the requirements of customers (Kim et al. 2013).

There are some advantages of polyester rope as follows: light weight, which can reduce the weight of the mooring system and increase the efficiency of the mooring level; low cost, much lower than steel cable; excellent in seawater corrosion resistance and no corrosion in seawater; and its fatigue performance which is much better than steel cable from experimental results. It also provides high strength and strength/quality ratio. Disadvantages of polyester rope are concluded as short application history, non-wear resistance, easy damage, and significant nonlinearity of stiffness.

Key Applications

An electric cable is made up of conductor, insulation, sheath, and so on, which can be used for

communication, power, and other related transmission and which are installed in the air, underground, or subsea. In fact, the distinction between cables and wires is not very strict. Usually the diameter and core of the wires are small, and the structure is simple. The wires without insulation and sheath are called naked wires. And the others are called electric cables (Calmet et al. 2002).

In 1879, it was an American inventor named Thomas Edison who wrapped jute around a copper rod, put it into an iron pipe, and filled in an asphalt mixture to form an electric cable. It was this man who laid this cable in New York, which pioneered the underground transmission of electricity. The next year, Cattender from Britain invented a power cable of insulation with bituminous impregnated paper. In 1908, a 20-KV cable network was built in Britain, which made the application of power cable more and more extensive. Three years later, a high-voltage cable with 60 KV was laid in Germany, indicating the scientific revolution of high-voltage cables. Since 1979, ultra-high-voltage cables of cross-link have been put into trial operation in Japan, Finland, Sweden, and Norway. Electric cables have been used for more than 100 years, which have been playing an important role in communication up to now with the development of the information technology.

In addition to transmitting electricity, electric cables are also indispensable in communication. In 1839, Cook and Wheatstone built the first 21-kilometer-length telegraph line in London. The first subsea cable in the world was laid across the English Channel between England and France by John Watkins Brett in 1850. But a fisherman mistakenly thought that he had found a big fish and pulled the cable out of the subsea. The following year, Brett laid the cable again and succeeded finally, meaning the UK and the European continent were connected with each other successfully (Fig. 2).

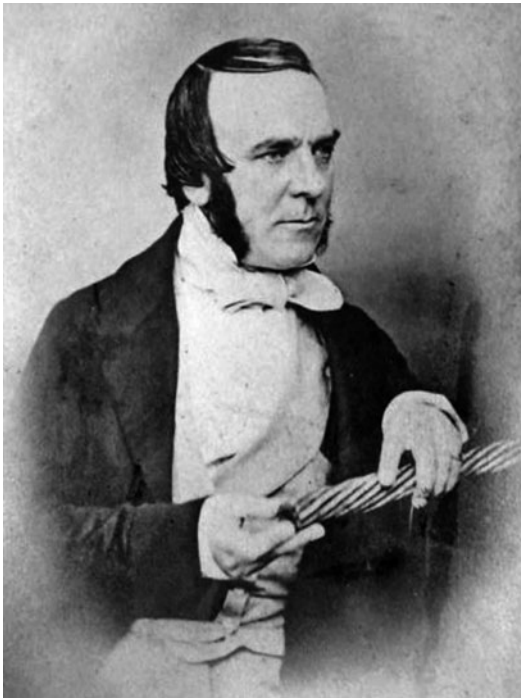
The world's first transoceanic subsea cable was laid successfully by an American rich businessman named Cyrus West Field in 1866. He began this project since 1854, and it lasted more than 10 years, which experienced four failures. In

1876, thanks to the invention of telephone by Bell, it began to accelerate the laying of subsea cables on a large scale around the world. Until 1902, the global submarine communication cable system was completed basically. Ever

since, communication cables have developed rapidly in western countries and spread among big cities, which reached a climax after the 1920s, and many countries have realized connections with communication cables between each other (Fig. 3).

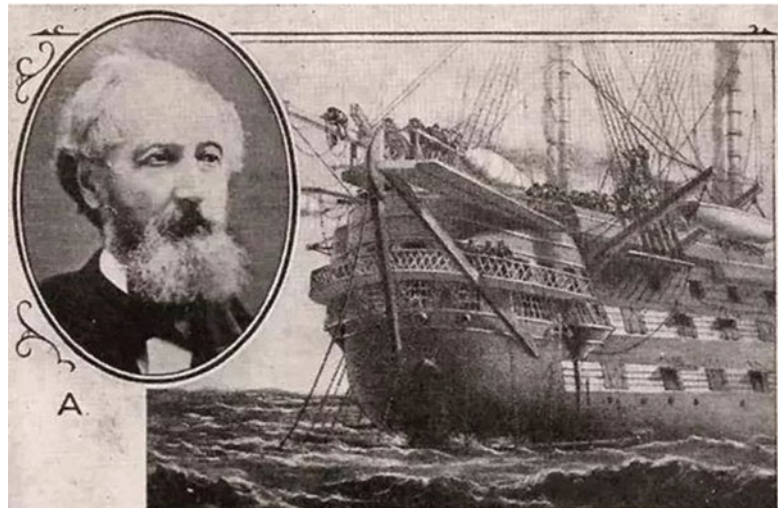
Polyester rope mooring systems have been widely used around the world, especially in the waters of Brazil and the Gulf of Mexico. Brazilian State Oil Company began to use experimental polyester rope in 1995 for the first time and replaced within 2-year period. The first practical application was an external mooring buoy by Brazilian State Oil Company in 1997.

As mentioned above, electric cables have been used for more than 100 years, which were used to transmit electricity originally. Electric cables played an important role in communication up to now with the development of the information technology. Power system used in wire and cable products mainly includes overhead bare wires, power cables (plastic cables, rubber sets of cables, overhead insulated cables), electromagnetic wires, and power equipments with electrical wire and cable equipment, etc. In addition, the main wires and cables used in the information transmission system are municipal telephone cables, TV cables, electronic cables, radio-frequency cables, optical fiber cables, data cables, electromagnetic cables, power communication cables, and other composite cables.



Cable, Fig. 2 John Watkins Brett (https://nl.wikipedia.org/wiki/John_Watkins_Brett)

Cable, Fig. 3 Cyrus West Field (<http://atlantic-cable.com/Field/>)



Cross-References

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- [Polyester Rope](#)
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- [Winding Wire](#)

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CAD (Computer-Aided Design)

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Capacity of Suction Anchor

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Definition

The suction anchors are usually cylindrical units with large diameter (D of 3–8 m), open at the

bottom and closed at the top, and generally with an aspect ratio (length to diameter, L/D) of two to six. The anchoring line is connected to the padeye attached to the side of the caisson at a depth down the mudline. The capacity of suction anchor is therefore the ability to resist the permanent and environmental loading from the anchor line during the operational conditions.

Failure Mechanism

The capacity of suction anchor is highly dependence of soil plasticity flow around the foundation. Many influential factors, especially for the complicated boundary conditions, such as anchor geometry, type of mooring system (loading direction and magnitude), installation performance, soil conditions, time after installation, durations of applied sustained loading, cycles of loading, etc., should be explicitly accounted for to investigate the failure mechanism.

In the following subsections, the typical failure mechanisms under uniaxial (including mainly vertical and horizontal) and inclined loading are discussed.

Under Vertical Uplift Loading

Under conditions of pure uplift loading, three failure mechanisms have been identified by a

number of experimental or numerical analyses: three important components of failure modes including reverse end bearing failure, sliding failure and tensile failure, as illustrated in Fig. 1.

Reverse End Bearing Failure

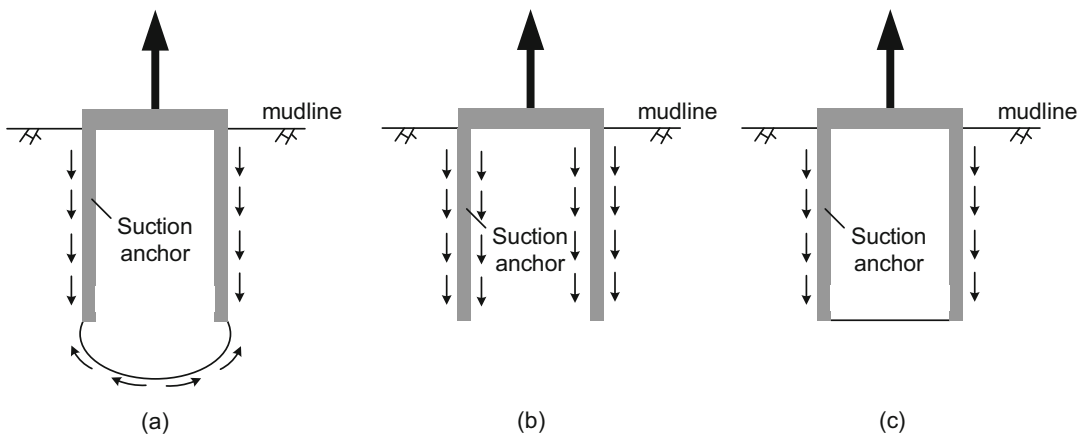
Reverse end bearing failure can be relied upon for a suction anchor if the top cap is sealed and the soil response is undrained, typically when the anchor is subjected to short-term (transient) loading. Passive suction is generated in the anchor, so that the soil plug is pulled out together with the caisson. The failure mode at the bottom during uplift was similar to that during compression and was referred to as “a reverse end bearing mechanism.”

Sliding Failure

When suction anchor is subjected to a sustained (drained) loading or the anchor lid is vented, reverse end bearing cannot be generated, and the failure occurs along the internal and external skirt.

Tensile Failure

When partially drained condition prevails, such as when the suction anchor is pulled out at intermediate rates, passive suction may be generated partially, so that the caisson and the internal soil plug tends to detach from the soil below the caisson.



Capacity of Suction Anchor, Fig. 1 Failure mechanisms for vertical pullout resistance (a) reverse end bearing, under undrained conditions; (b) caisson pullout, under

drained conditions; (c) caisson and plug pullout, under partially drained conditions

Under Optimal Horizontal Loading

The suction anchors under pure horizontal loading is rarely found offshore, while drag angle at the padeye is typically at a small angle of 10° – 20° in the catenary mooring system. Meanwhile, the padeye position is often optimized in design, allowing the anchor to slide horizontally with minimal rotation.

As shown in Fig. 2, the corresponding failure mechanism for lateral capacity of the suction anchor is considered to be similar to those for laterally load piles (Murff and Hamilton 1993) that a passive/active conical wedge extending from the edge of the caisson at shallow depth to the seabed dominating the soil mechanism; and a confined flow region is assumed below the wedge with the center of rotation locating below the anchor tip (Martin and Randolph 2006). For caissons loaded above the optimal depth, an external scoop mechanism can replace the flow region, with the center of rotation locating within the anchor.

Under Inclined Loading

The failure mechanism under inclined loading depends on the position of the centerline intercept for the applied load (varying with padeye position and loading angle), the length of the suction anchor, and the soil properties (Supachawarote 2006). The impact of tensile crack on the failure mechanism is examined by Fu et al. (2020).

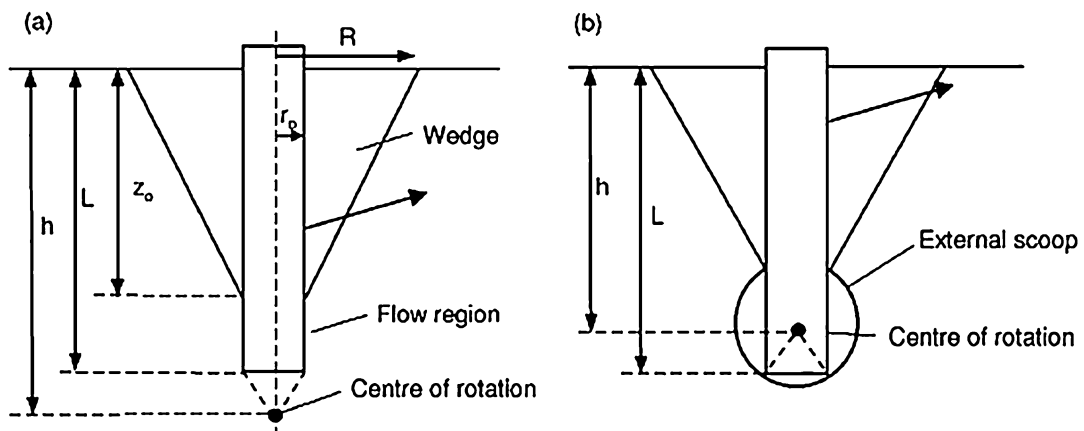
The maximum holding capacity is obtained if the chain is attached at a depth where the anchor moves largely in horizontal translation with minimal rotation. This is called the “optimal padeye depth.”

When the centerline intercept is located above the optimal padeye position, the anchor rotates forward, and combinations of wedge flow around and full circular rotational failure were observed. For the short anchor, the wedge and rotational flow mechanisms were observed, without the flow around region. For the longer anchor, the flow around region increasingly dominates.

While when the centerline intercept is located below the optimal padeye position, a backward rotation was observed, and the rotational flow failure extends to the mudline. And with the centerline loading point moves deeper, the full circular rotational failure eventually moves below the soil surface, which is evident for anchor with increasing length. For long anchor, the circular rotational flow failure is only observed below the tip of the anchors; and only combination of the wedge and flow around mechanisms was observed within the anchor length.

Design Method of Evaluating Capacity

The load capacity of a suction caisson anchor, assuming a sealed cap, is derived from bearing



Capacity of Suction Anchor, Fig. 2 Soil failure mechanism for suction anchor (a) conical wedge and flow region and (b) external base rotational scoop. (After Randolph and House 2001)

resistance between the soil and the projected area of the caisson (on a vertical plane for horizontal resistance and the cross-sectional area for reverse end bearing), aided by frictional resistance along the outside of the caisson shaft. Reverse end bearing relies on passive suctions developed within the soil plug, and so consideration needs to be given to the time over which these can be sustained.

Vertical Uplift Capacity

Equation

The vertical pullout capacity is equal to one of the following depending on the drainage conditions (essentially whether the passive suction is sustained during loading):

- (a) For undrained conditions, with a non-dimensional diameter $T_k (= c_v/vD$, where c_v is the consolidation coefficient, v is the loading velocity, D is the diameter of the anchor) recommended to be less than 0.002 (Deng and Carter 2000), the passive suction is thought to be sustained during loading

$$V_{ult} = W' + A_{se}\alpha_e\bar{s}_{u(t)} + N_c s_u A_e \quad (1)$$

- (b) For drained condition, ($T_k > 0.6$), the passive suction is not generated during loading

$$V_{ult} = W' + A_{se}\alpha_e\bar{s}_{u(t)} + A_{si}\alpha_i\bar{s}_{u(t)} \quad (2)$$

- (c) For partially drained condition ($0.002 < T_k < 0.6$), passive suction is partially generated

$$V_{ult} = W' + A_{se}\alpha_e\bar{s}_{u(t)} + W'_{plug} \quad (3)$$

where A_{se} and A_{si} are the external and internal shaft surface area, A_e is the external cross-sectional area, α_e and α_i are the coefficient of external and internal shaft friction, N_c is the reverse end bearing factor, s_u is the undrained shear strength at tip level, $\bar{s}_{u(t)}$ is the average

undrained shear strength over penetrated depth at time t after installation, W'_{plug} is the effective weight of the soil plug, and W' is the submerged caisson weight.

Evaluation of Parameters α and N_c

The soil shear strength around the anchor is thought to be remold due to disturbance during skirt penetration during self-weight penetration. And this strength recovers later with time following installation, due to a combination of thixotropy and consolidation (Andersen et al. 2005), which however remains below the intact undrained shear strength. This process is referred as “setup” and has been expressed conveniently with α_s , with $\alpha < 1$. A base value of α of 0.65 was proposed for suction caisson design (Anderson and Jostad 2004), subjected to their specific soil properties. The shaft friction depends on the anchor surface roughness, soil type, and over-consolidation ratio. And Chen et al. (2009) suggests method of installation (referring jacking or suction installation) leads to negligible difference in α based on the centrifuge and LDFE numerical studies. And their results also suggest α of 15–20% lower than those recommended by the American Petroleum Institute (API) for driven piles in normally consolidated clays. It should also be noted that lower values of friction coefficient for internal shaft friction compared with external friction were reported in the model tests conducted on double-walled caisson (Jeanjean 2006).

The reverse end bearing N_c is a function of soil property and the aspect ratio (L/D) of the anchor. For an anchor with L/D of 2 in the over-consolidated kaolin clay, Fuglsang and Steensen-Bach (1991) reported the centrifuge and laboratory tests, suggesting the reverse bearing capacity factors varying between 6.5 and 8.5. This is lower than the theoretical lower bound results of 9.2 for the caisson with L/D of 2 (Martin 2001). Other centrifuge tests for the anchor with the same aspect ratio (Clukey and Morrison 1993) suggested that reverse end bearing is around 11. This high magnitude of N_c may be related to the vane shear strength adopted for the interpretation, which leads to an overestimation of 25% (Watson

et al. 2000). Considering the loading mode (monotonic transient loading, cyclic loading, and long-term sustained loading), the reverse end bearing falls in a wider range of 9.1–14.6 (Randolph and House 2001). And a conservative N_c value of 9 is suggested to be taken, due to the strain-softening nature of the response as the caisson is extracted. A bearing capacity factor of $N_c = 9$ irrespective of the aspect ratio for suction anchor is also suggested by El-Gharbawy and Olson based on the results of the laboratory tests in kaolin clay. And by contrast, an N_c value of 12 is found to be mobilized at large displacements (Jeanjean 2006), although values of around 9 were mobilized at the displacement where peak external shaft friction was achieved. Hence it is rational to take N_c as of 9 is appropriate.

Optimal Horizontal Capacity

Equation

Assuming the padeye is positioned at the optimum level and pure translation failure occurs, the maximum horizontal resistance is

$$H_{\max} = LD_e N_p \bar{s}_u \quad (4)$$

where L is the embedded length of caisson, D_e is the external diameter of the caisson, N_p is the lateral bearing capacity factor, and $\bar{s}_u$ is the average undrained shear strength over penetrated depth.

Evaluation of Parameters N_p

Zhang et al. (2016) recommended the following approach to calculate the lateral bearing capacity factor

$$\begin{aligned} N_{p0} &= N_1 \\ &- (N_1 - N_2) \left[1 - \left(\frac{z/D}{Z} \right)^{0.6} \right]^{1.35} \\ &- (1 - \alpha) \\ &\leq N_{pd} \end{aligned} \quad (5)$$

where

$$N_1 = 11.94 \quad (6)$$

$$N_2 = 3.22 \quad (7)$$

$$Z = 16.8 - 2.3 \lg \lambda \geq 14.5 \quad (8)$$

$$\lambda = \frac{s_{um}}{kD} \quad (9)$$

$$N_{pd} = 9.14 + 2.8\alpha \quad (10)$$

In Eq. 8, Z stands for the normalized depth (z/D) at which the soil failure mechanism transits from the wedge failure to the localized flow around failure for a fully rough soil-pile interface in idealized weightless soil. The value of Z is related to the normalized strength homogeneity parameter λ . N_{pd} stands for the limiting bearing capacity factor mobilized in a localized flow-around mechanism, expressed as a function of the interface roughness factor α ($0 \leq \alpha \leq 1$), according to the plasticity solution.

Inclined Load Capacity

In the presence of both vertical and horizontal loading, a reduction occurs in the pure vertical and horizontal capacities, as the caisson is simultaneously displaced vertically and laterally (or rotated). Clukey et al. (2003) suggested for the loading angles applied by the catenary mooring line system, which are generally less than 20° from the horizontal, the caisson capacity is dominated by the horizontal capacity, and the capacity may be estimated by the capacity under pure translation divided by the cosine of the loading angle at the padeye. Conversely for high loading angles from the horizontal applied by taut and semi-taut mooring systems (generally in excess of 30°), the caisson capacity is essentially governed by the vertical capacity of the caisson and maybe approximately taken as the vertical capacity divided by the sine of the loading angle.

For the more general cases, the interaction between vertical and horizontal loading may be conveniently modelled as a failure envelope in the combined vertical-horizontal load space, which was found to vary with the aspect ratio (L/D), location, and direction of the applied load. The shape of the failure envelopes for suction anchor may be modelled by an elliptical relationship

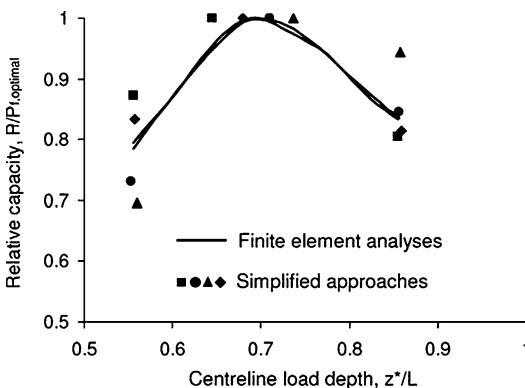
$$\left(\frac{H}{H_{ult}}\right)^a + \left(\frac{V}{V_{ult}}\right)^b = 1 \quad (11)$$

where H_{ult} and V_{ult} are the uniaxial horizontal and vertical capacities, respectively, and the exponents a and b vary with caisson aspect ratio L/D according to Supachawarote (2006).

$$a = \frac{L}{D} + 0.5 \quad (12)$$

$$b = -\frac{L}{3D} + 4.5 \quad (13)$$

The sensitivity of caisson capacity to the changes in padeye position or loading angle was also investigated using the three-dimensional finite element analysis, limit equilibrium solutions upper bound limit analyses, profiles of lateral resistance based on Murff and Hamilton (1993), and centrifuge modelling (Clukey et al. 2003). It was found that the variation of centerline intercept for the applied load (varying with padeye position and loading angle) results in the large discrepancy of the capacity. As shown in Fig. 3, the centerline loading depth around 0.7 gives the largest capacity. With the shift of the padeye position and loading angle, the capacity reduces gradually within the depth of $\pm 0.15 L$ from the optimal centerline location and reduces sharply beyond this range.



Capacity of Suction Anchor, Fig. 3 Effect of padeye depth on capacity. (After Andersen et al. 2005)

Consideration in Design

The Effect of Crack on the Suction Anchor Capacity

The inclined load capacity of a suction anchor will also depend on whether a crack develops along the trailing edge of the caisson. A crack is normally formed for suction anchors with a long aspect ratio in over-consolidated soils. The finite element results (Fu et al. 2020) show a reduction of up to 50% for any loading angle (the reduction is mainly found at the optimal padeye depth and maintained for loading angles of up to about 45° for $L/D = 1.5, 2$, and 3 in homogeneous soil).

Considering Site Conditions

Some changes in site conditions should be explicitly accounted for in the design of geotechnical capacity of suction anchor. The soil scour around a suction foundation is an important scenario in design, where the soil migration around foundation might occur under the wave and current loading. The volume of soil mobilized by foundation is therefore changed, resulting in a significant decrease in capacity. In some cases, the shallow gas accumulation inside a suction foundation also should be given attention (Gylland and de Vries 2008). Additionally, the trenching is another issue and most likely to occur particularly when using semi-taut to taut mooring configurations with caisson employed in the soft deposits. Considerable motion of ground chain could lead to the remolded softening and erosion of soil particles, forming a curved trench in the vicinity of foundation. O'Neill et al. (2018) presented a method to interpret and estimate the primary mechanism of trenching development.

Considering Cyclic Loading Effect

Offshore foundations are often cyclically loaded in both calm sea condition and extreme events (i.e., storm or hurricane event). For suction anchor, the cyclic wind, wave, current, and structural loadings are transferred to the foundation by mooring lines. It gives rise to pore pressure accumulation for soil around foundation, which is one of the principle concerns in design of foundation's capacity. Regarding clay, the strain softening may occur as the pore pressure accumulates, and both

soil stiffness and strength can significantly reduce. Regarding sands, the earlier densification may occur during cyclic loading, then pore pressure starts accumulating, and finally a state of liquefaction may be present with the effective stress of soil almost being zero. Therefore, the continual buildup of excess pore pressure in soil under cyclic loading potentially results in a significant decrease in capacity of foundation.

Based on laboratory cyclic triaxial (TA) and direct simple shear (DSS) tests, Andersen (2015) presented a method to assess the accumulation of pore pressure, which is dependence of cyclic shear ratio τ_{cy}/σ'_{vc} . τ_{cy} is the cyclic shear stress (kPa), and σ'_{vc} is the vertical consolidation pressure (kPa). The corresponding cyclic shear strength $\tau_{cy,f}$ at failure was indicated using a strain contour diagram, with the equivalent number of cycles to failure captured at the permanent cyclic shear strain of 15%. Besides, many soil models based on the critical soil mechanics theory were numerically developed in the last three decades, to describe and interpret the cyclic behavior of soil. However, no constitutive model has been developed to date, enabling to represent all the key characteristics of cyclic soil response, as many factors may influence the cyclic performance of soil, i.e., cyclic stress level, loading frequency, over-consolidation ratio, static pre-shearing, etc. In practice, the alternative option available for engineers is choosing a soil material factor (ISO 2016) or considering the uncertainty by a partial factor of capacity (ISO 2016), when the cyclic loading effect is necessary to be considered.

Considering Reconsolidation Effect

For suction anchor, an allowable duration is often found after its installation, e.g., 3 to 6 months. This gives an opportunity for consolidation under self-weight prior to commencing operations. During consolidation, the excess pore pressures generated by the self-weight preloading dissipate reducing the void ratio of soil and enhancing the soil strength, hence increasing the foundation capacity. It is notable that the dissipation of pore pressure also occurs during the cyclic loading condition, as the pore pressures in fact generated during a cyclic duration and will

dissipate in the subsequent cycles. The appropriate consideration of reconsolidation in service design is necessary to develop a reliable and economical method.

From some centrifuge model tests (Bienen et al. 2010; Fu et al. 2015), the significant increase in bearing capacity of a preloaded foundation in clay was found, for instance, a gain in bearing capacity of 30% after 10% of consolidation and a doubling after 80% of consolidation under a vertical preload of 50% of the ultimate vertical load.

Notation

a	Parameter depending on L/D
A_e	External cross-sectional area
A_{se}	External shaft surface area
A_{si}	Internal shaft surface area
b	Parameter depending on L/D
c_v	Consolidation coefficient
D	Diameter of suction anchor
D_e	External diameter of suction anchor
H_{max}	Optimal horizontal capacity
H_{ult}	Uniaxial horizontal capacity
L	Length of suction anchor
N_c	Reverse end bearing factor
N_p	Lateral bearing capacity factor
s_u	Undrained shear strength at tip level
$\bar{s}_u$	Average undrained shear strength over penetrated depth
$\bar{s}_{u(t)}$	Average undrained shear strength over penetrated depth at time t after installation
T_k	Nondimensional diameter
W'	Submerged weight of suction anchor
W'_{plug}	Submerged weight of soil plug
v	Loading velocity
V_{ult}	Vertical pullout capacity
z_{cl}	Centerline loading depth
α	Coefficient of shaft friction
α	Parameter depending on L/D
α_e	Coefficient of external shaft friction
α_i	Coefficient of internal shaft friction
β	Parameter depending on L/D
τ_{cy}	Cyclic shear stress
$\tau_{cy,f}$	Cyclic shear stress at failure
σ'_{vc}	Vertical consolidation pressure

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Capex (Capital Expenditure)

► Tension-Leg platform

Catenary Anchor Leg Mooring

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Synonyms

Catenary mooring; Mooring system; Single anchor leg mooring; Single point mooring; Taut mooring

Definition

A typical CALM system consists of the mooring buoy, chain and anchor system, mooring assemblies, and floating and submarine hose strings. The buoy body provides the necessary buoyancy to keep the system afloat. A rotating deck or turntable is mounted on top of the buoy body to transmit mooring force from the mooring assemblies through the buoy to the anchor chains. The CALM anchoring system consists of 4–8 anchor chains extending from the buoy body radially out and terminating at firmly embedded anchors or anchor piles (Luo et al. 2015).

Scientific Fundamentals

System Composition

The CALM system consists of a large buoy, which supports a number of catenary chain legs anchored to the sea floor (Fig. 1). Riser systems or flow lines that emerge from the sea floor are attached to the underside of the CALM buoy. Some of the systems use a hawser, typically a synthetic rope, between the vessel and the buoy. Since the response of the CALM buoy is totally different than that of the vessel under the influence of waves, this system is limited in its ability to withstand environmental conditions. When sea states attain a certain magnitude, it is necessary to cast the vessel off. In order to overcome this limitation, rigid structural yokes with articulations are used in some designs to tie the vessel to the top of the buoy (Vryhof 2005).

CALM system can be used both in shallow water and deepwater area and are connected to a shore storage facility (tank farm) or to offshore production platforms by means of a submarine pipeline. For this application, the CALM is used as an offloading system for a deepwater Floating Production Storage and Offloading unit (FPSO).

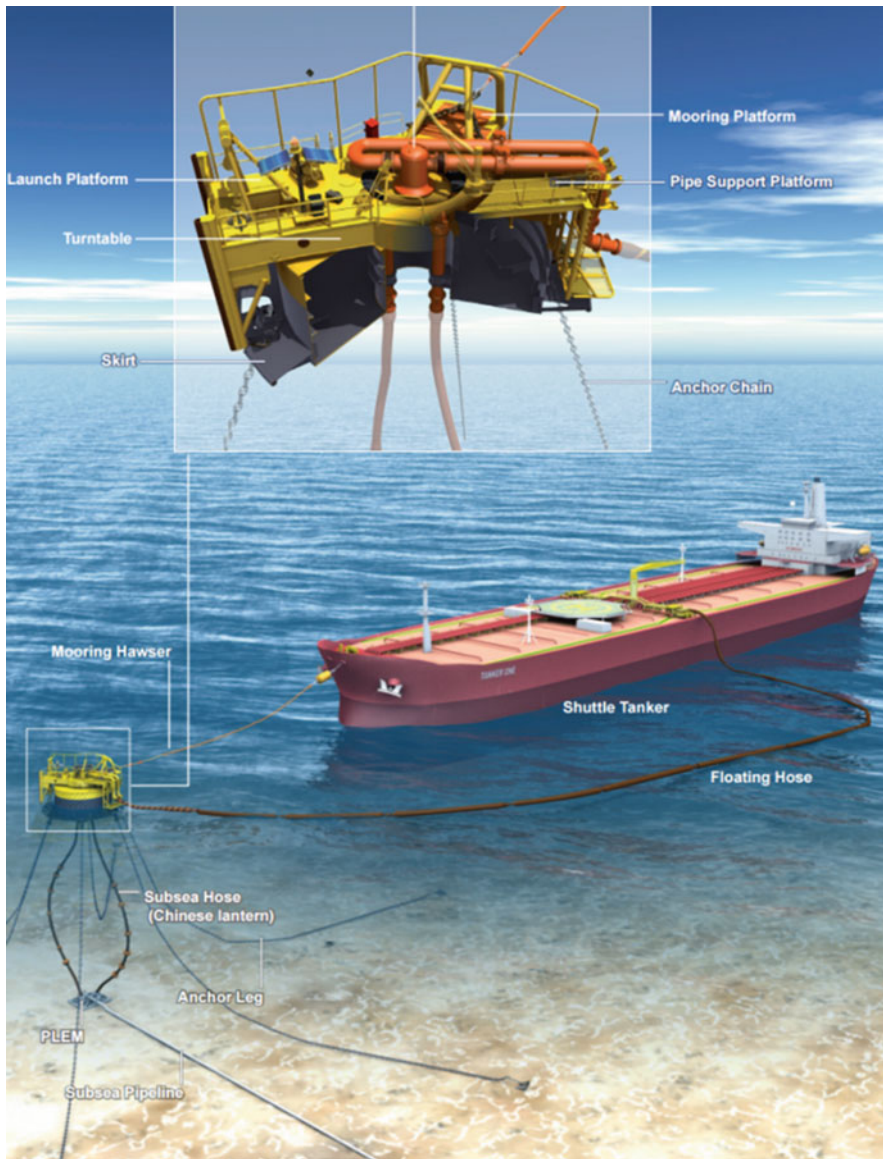
CALM is also a complex nonlinear system for oil tanker or FPSO. In addition to the three degrees of freedom in the horizontal plane (surge, sway and yaw), the entire external transmission

system has 12 degrees of freedom (buoys 6, tanker or FPSO 6) of the motion response. If the simultaneous CALM system-FPSO-the coupling motion analysis of the tanker, it will reach 18 degrees of freedom. Because the incident wave period is generally greater than 8 s, the longitudinal oscillation is the main motion form of the buoy. Therefore, it is necessary to take into account the pitching motion characteristics of single point mooring system in the design of pipeline (SBM 2012).

The CALM system allows the moored tanker to weathervane. With this principle, the tanker offers to the environment (waves, current, and wind), a direction of least resistance, thus the system can operate in much higher conditions than the other systems. The CALM system reacts against the force created by the environment on the moored tanker by a set of two spring effects: (a) The nylon mooring hawser, (b) The buoy mooring legs, which are installed in a Catenary configuration (GL 2014).

CALM-catenary anchor leg mooring can be capable of handling very large crude carriers. This configuration uses six or eight heavy anchor chains placed radially around the buoy, of a tonnage to suit the designed load, and attached to an anchor or pile to provide the required holding power. The anchor chains are pre-tensioned to ensure that the buoy is held in position above the PLEM. As the load from the tanker is applied, the heavy chains on the far side straighten and lift off the seabed to apply the balancing load. Under full design load, there is still some meters of chain lying on the bottom. The flexible hose riser may be in one of three basic configurations, all designed to accommodate tidal depth variation and lateral displacement due to mooring loads. In all cases, the hose curvature changes to accommodate lateral and vertical movement of the buoy, and the hoses are supported at near neutral buoyancy by floats along the length (Chakrabarti 2005). These are

- Chinese lantern, in which two to four mirror symmetrical hoses connect the PLEM with the buoy, with the convexity of the curve facing radially outwards.



Catenary Anchor Leg Mooring, Fig. 1 CALM (www.sbmoffshore.com, 2012)

- Lazy-S, in which the riser hose leaves the PLEM at a steep angle, then flattens out before gradually curving upwards to meet the buoy approximately vertically, in a flattened S-curve.
- Steep-S, in which the hose first rises roughly vertically to a submerged float, before making a sharp bend downwards followed by a slow curve through horizontal to a vertical attachment to the buoy (API RP 2SK 1993).

Historical Development

The history of mooring system started from the development and use of the single point mooring device of catenary buoy by the US navy during World War II, and then developed rapidly. Numerous different mooring systems have been developed over the years (Kent et al. 2007).

The world's earliest single point mooring system is CALM. CALM mooring system is the most widely used single mooring. There are over

560 single point mooring system over the world. And there are about 500 CALM systems by 2008. In 1997, a polyester rope + anchor chain catenary mooring system is first used for a FPSO in Brazil. Afterwards, more and more fiber mooring systems appeared in the Gulf of Mexico (Winkler and Mckenna 1994). Since early 2000, the CALM design has been used and adapted to deepwater conditions, greater than 1,000 m (ABS 2014). Installed CALM in deepwater are listed as following: Agbami (Nigeria, 1435 m), Kizomba A&B (Angola, 1200 m, 1000 m), Dalia (Angola, 1341 m), Erha (Nigeria, 1190 m), Akpo (Nigeria, 1285 m), Bonga (Nigeria, 1000 m), Girassol (Angola, 1320 m), Greater Plutonio (Angola, 1310 m).

Key Technology in the Development of CALM

Since the 1970s, the research on the CALM system can be divided into the following two aspects: (1) engineering design – the deploy/recovery of mooring system, the composition of mooring line, the arrangement of mooring system, and the development of anchor. (2) theoretical research – environmental load, static analysis of anchor mooring line, dynamic response between mooring system and mooring floating body.

Haixiao Liu et al. (2014) used a specially designed experimental system to investigate the nonlinear mechanical behavior of synthetic fiber cables, including how stiffness changes, how the main factors affect the hydrodynamic stiffness changes, and the nonlinear tension-elongation relationship. The similarity criterion of the hydrodynamic stiffness of fiber cables derives from scale analysis and experiments verification. The accuracy of empirical expressions for hydrodynamic stiffness currently used is verified by measured data. The results show that the uniform load is an important factor affecting the hydrodynamic stiffness, and the effect of the amplitude of strain change on the stiffness cannot be ignored. The cyclic load is also an important factor affecting the hydrodynamic stiffness. Based on the measured data, an empirical expression considering both uniform load, strain amplitude, and the number of cyclic loads is presented.

It is the only expression that can evaluate the change of hydrodynamic stiffness under long-term cyclic loading.

Minzhe Li (2013) designed a mooring system for semi-platform by combining the semi-physical simulation and mooring system control method, to control the mooring line retracting and deploying to realize platform positioning and to detect the positioning effect under different sea conditions.

Amir G. Salem et al. (2012) studied the method of estimating pitch motion of CALM system in frequency domain and verified it by experiment. The hydrodynamics of CALM system is directly related to the fatigue life of the mooring system under the coupling interaction of the mooring line and the oil pipeline. It is very important to accurately predict the hydrodynamic response of the buoy.

H. S. Da Costa Mattos and Chimisso (2011) studied the image model of creep test for low-density and high-strength HMPE fibers. In the macroscopic method, besides the traditional variables (pressure, total strain), scalar variables related to damage induced by creep process are introduced, and the evolution law of the damage variable is proposed. The life and elongation of HMPE specimens were predicted by creep tests at different loading levels and room temperature, which were in good agreement with the experimental results.

Davies P et al. (2011) studies the effects of uniform load, load range, and stiffness loading frequency on synthetic fiber cable including polyester cable, aramid cable, and HMPE in dry environment experimentally. In subsequent experiments, the bending stiffness of aramid cable and HMPE cable and the operation in deep water were discussed.

Meng Yuan (2010) used finite element time-domain method to construct the numerical calculation model of mooring cable for single-point and multi-point mooring system. Given a sinusoidal movement at the top of SPM system, the dynamic response of the whole mooring line was calculated. Taking the multi-point mooring system of Spar platform as an example, the time-history variation of the tension at the end of the mooring

line was calculated. The theoretical calculation results are consistent with the model test.

Francois and Davies (2008) carried out a sampling polyester cable test of cable with breaking strength of 70 tons and a full-scale test with the breaking strength of 800 tons. Viscoelastic response of the cable is considered for the quasi-static stiffness to reduce the change of uniformly distributed load under changing environmental conditions. The upper and lower boundaries of stiffness only considered the effect of uniformly distributed load.

Jing He (2007) used single-objective and multi-objective optimization methods to optimize the length, material price, and fracture strength of the two-component catenary mooring line of CALM system.

Y. T. Chai et al. (2002) proposed a semi-analytical quasi-stationary method based on catenary equation to deal with the interaction between the three-dimensional multi-mooring line partial subsection state and all subsection suspension state and the seabed, which can efficiently analyze the parameters of multi-point mooring system and flexible riser under different states.

Casey and Banfield (2002) investigates polyester cables with hydrodynamic axial stiffness ranging from 600 to 1000 tons, and points out that the strain amplitude is indeed a variable affecting the hydrodynamic stiffness.

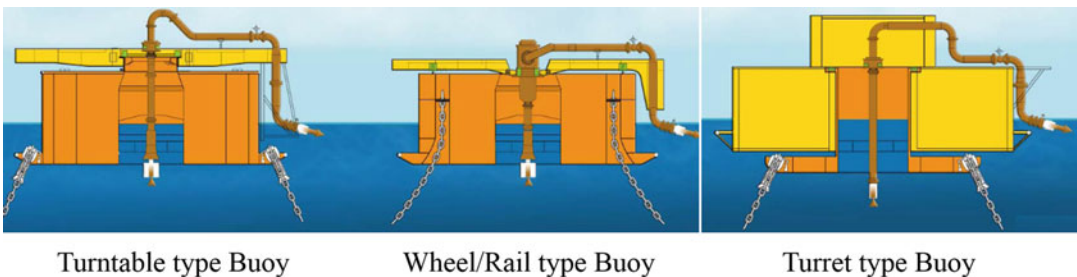
Fernandes et al. (1999) carried out a full-scale polyester cable test with a diameter of 0.127 m. It is found that the dependence of hydrodynamic stiffness on frequency is weak.

Key Applications

The CALM is the most popular and widely used type of offshore loading terminal. CALMs have been deployed worldwide for a variety of applications, water depths, and vessel sizes ranging from small product carriers to Very Large Crude Carriers (VLCC). Because of safe and easy berthing and un-berthing operations, the CALM is equally the preferred offshore terminal of Mooring Masters and Tanker Captains (Luo et al. 2015).

CALM type single point is divided into the earliest Wheel CALM type, Turntable CALM type, and the newly developed Turret CALM type. At present, the most commonly used type is Turntable CALM. Compared with Wheel CALM, it uses large diameter rolling bearing at the center of the buoy to replace Bogey Wheel and transfer mooring force. In this way, the liquid rotary joint is completely independent of the turntable and bearing structure and no longer acts as a bearing member. The single point in this form consists of the following parts: buoy body, rotary turntable device, pipeline system and liquid rotary joint, anchor system, pipe manifold baseplate, underwater hose and floating hose, mooring rope device, and auxiliary equipment. The schematic diagram of the three types of buoy structure is shown in Fig. 2 (BV 2006).

The CALM system has no restrictions on the size of the tanker, is easy to maneuver, works around the clock, requires the least staff, and has strong adaptability to the environment. The advantages of the CALM system can be summarized as follows:



Catenary Anchor Leg Mooring, Fig. 2 Schematic diagram of the three types of buoy structure (www.sbmooffshore.com, 2012)

1. There is no need for a natural and excellent deep-water port, and there is no need for fixed dock facilities to provide loading and unloading operations.
2. The system is of lower cost.
3. It can be installed in open sea areas that do not require protection.
4. Can work normally all the year round.
5. Easy to install.
6. The mooring system is simple.
7. It can transmit a variety of oil and gas products at the same time.
8. The shuttle tanker has its own mooring propulsion system.
9. The terminal does not require a special tugboat for berthing and offshore, only a smaller vessel is required to serve (Luo et al. 2015).

Cross-References

- [Catenary Mooring](#)
- [Mooring System](#)
- [Single Anchor Leg Mooring](#)
- [Single Point Mooring](#)
- [Taut Mooring](#)

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Catenary Mooring

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Synonyms

[Drag embedment anchor](#); [Single-point mooring system](#); [Spread mooring system](#)

Introduction

Mooring systems have been around just as long as man has felt the need for anchoring a floating structure at sea. These systems were used, and are still used, on ships and consisted of one or more lines connected to the bow or stern of the ship. Generally, the ships stay moored for a short duration of time (days). When the exploration and production of oil and gas started offshore, a need for more permanent mooring systems became apparent. Due to the demand for marine resource development, a large number of new floating structures have been designed and constructed, and various mooring systems have been emerged over the years; the catenary mooring system is a common and traditional way of locating floating structures, and several pretension mooring lines are arrayed around the structure to hold it in the desired location (Fig. 1).

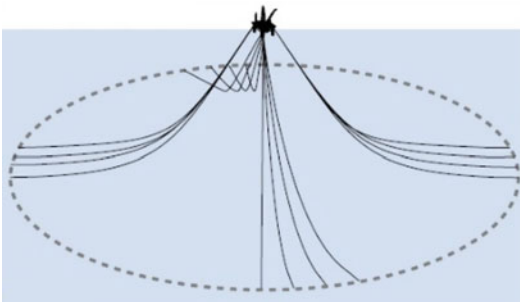
Definition

It gets its name from the shape of the free hanging line, one end of the mooring line is connected to the floating structure, and the other end is fixed at the anchor point of the seabed; the part that is suspended in the seawater will take a catenary shape, which is similar to the rope that is fixed at both ends and freely placed under the uniform force. The catenary forms can be seen everywhere in nature; when you wake up in the morning to see spider webs covered with water drops, when you

see the ropes on the suspension bridge, and when you see the wires between the two poles, you know what kind of shape are they? The answer is the catenary. As early as 1690, Jacobi Bernoulli proposed the famous “catenary problem,” but it was not until Newton and Leibniz invented the calculus to get the correct answer. The catenary is a magical curve that has been used extensively in buildings, bridges, ship, and ocean platform moorings (Wren et al. 1989).

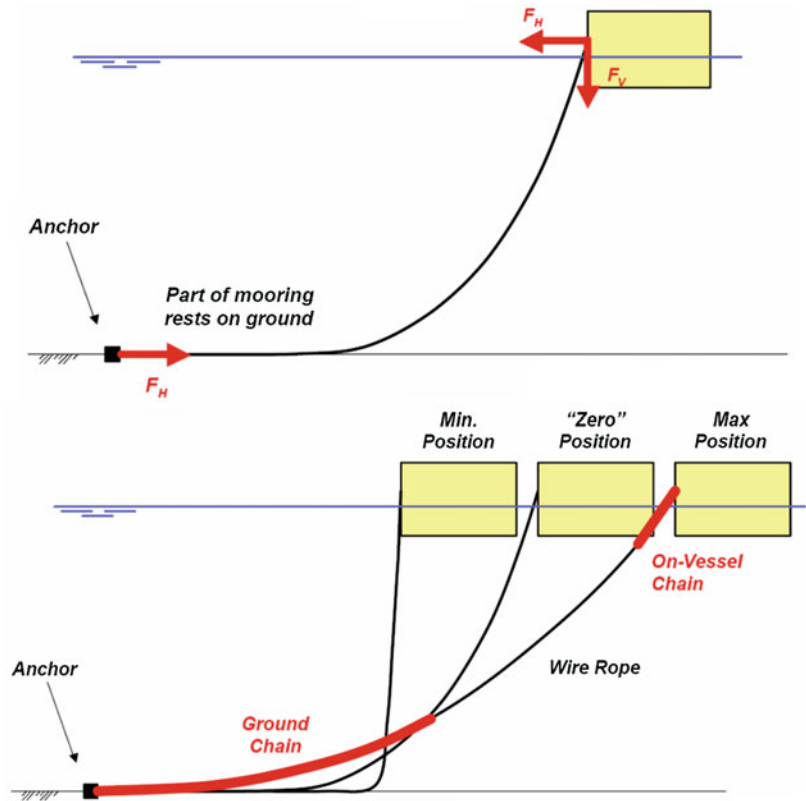
In marine engineering, spread catenary mooring systems and single-point catenary mooring systems are two common arrangements (Skop 1988). Mono-hull ship and semisubmersible platforms have traditionally been moored using a spread catenary mooring system, and the points are connected to different positions on the platform so that the platform cannot rotate freely, and the orientation of the platform remains basically unchanged. In some cases, excessive displacement caused by environmental forces may cause large loads on the mooring system. In order to overcome this drawback, a single-point mooring system has been developed which is characterized in that a plurality of mooring lines are connected at a point on the longitudinal centerline of the platform. The platform will have a wind vane effect to reduce the environmental loads caused by wind, current, and waves. Single-point moorings are used primarily for ship-shaped vessels. They allow the vessel to weather vane. This is necessary to minimize environmental loads on the ship-shaped vessel by heading into the prevailing weather (API 2005).

As can be seen from the above (Fig. 1), the catenary mooring system occupies a large range on the seabed, that is, the radius of influence of the mooring point will be large. In actual ocean engineering, the collision problem has to be considered with other structures, such as risers, conveyor cables, etc. At the seabed, the catenary mooring line lies horizontally, in general, which has a minimum length of 50 m; thus the mooring line has to be longer than the water depth, and the anchor points in a catenary mooring system only bear horizontal forces and are not subject to vertical forces (Fig. 2); the nonlinear restoring force generated by the catenary mooring system provides



Catenary Mooring, Fig. 1 The catenary mooring system of a semisubmersible platform. (From anchor manual 2005)

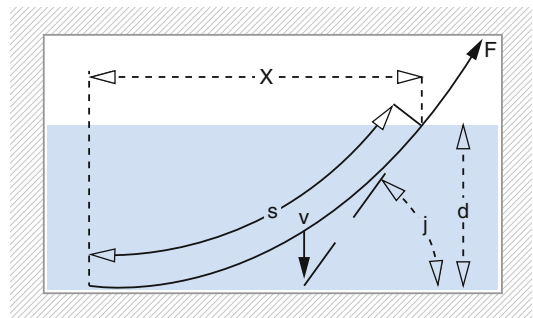
Catenary Mooring,
Fig. 2 The catenary
mooring line



the positioning function of the floating platform and balances the environmental load acting on the water platform (Samad 2009). Meanwhile, as the water depth increases, the length and weight of the mooring line also increase rapidly, which not only increases the tension in the chain but also increases the vertical load on the floating structure, further reducing the payload capacity of the floating structure. In that case, synthetic ropes are used because of conventional catenary systems become less and less economical (Banfield and Flory 2009). That is to say, the catenary mooring system is suitable for relatively shallow waters (<1000 m).

Theoretical Analysis Methods

For the mooring line of the general floating structure, it is not completely consistent with the theoretical catenary due to its own tensile, bending, and sea flow forces. However, for the convenience



Catenary Mooring, Fig. 3 Static analysis of catenary
mooring line

of calculation and analysis, the catenary theory is still used to describe its preliminary analysis and ignore the influence of sea current and wave, which is static analysis. Figure 3 shows catenary mooring line and its two-dimensional diagram of the force. The angle j is the angle between the mooring line at the fairlead and the horizontal; a

segment of arc microelement is selected from the catenary mooring line.

Depending on the water depth, the weight of the mooring line, and the force applied to the mooring line at the fairlead, the length of the suspended mooring line S in [m] can be calculated with:

$$S = \sqrt{d} \cdot \left(\frac{2 \cdot F}{w} - d \right)$$

with

d : the water depth plus the distance between sea level and the fairlead in [m]

F : the force applied to the mooring line at the fairlead in [t]

S : the unit weight of the mooring line in water in [t/m]

The horizontal distance X in [m] between the fairlead and the touchdown point of the mooring line on the seabed can be calculated with:

$$X = \left(\frac{F}{w} - d \right) \cdot \ln \left(\frac{S + \frac{F}{w}}{\frac{F}{w} - d} \right)$$

The weight of the suspended chain V in [t] is given by:

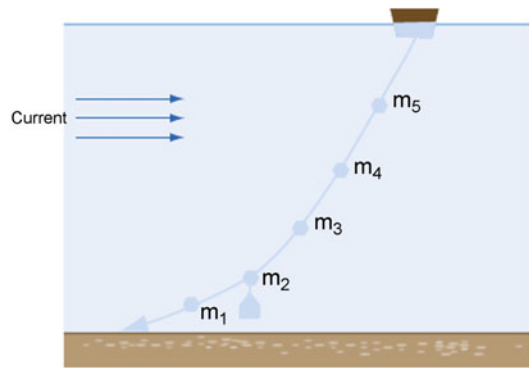
$$V = w \cdot S$$

The vertical distance of a catenary mooring line is given by the function:

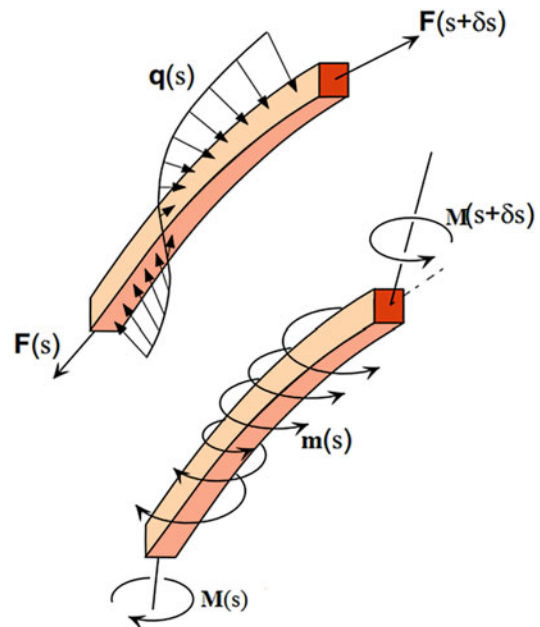
$$y = -\frac{T_0}{w} \left[\cosh \left(\frac{wS}{T_0} \right) - 1 \right]$$

According to the above formula based on single-component mooring line, the shape and force of the catenary mooring line can be preliminarily calculated. However, mooring lines are usually composed of materials with many different properties, the calculation principle is the same, but the equations of the individual components need to be integrated. As for the dynamic analysis of mooring lines, the lumped mass

method (Fig. 4) and the slender rod theory (Fig. 5) are commonly used. The lumped mass method is equivalent to simulating the slender structure into a series of lumped mass node systems connected by springs (Kato 1982). The finite difference method is used to solve the dynamic problem, which can simplify the analysis of the structure without losing accuracy. It has been widely used. The slender rod theory applies the overall coordinates and slope and establishes the



Catenary Mooring, Fig. 4 The lumped mass method. (Image by MIT OpenCourseWare)



Catenary Mooring, Fig. 5 The slender rod theory

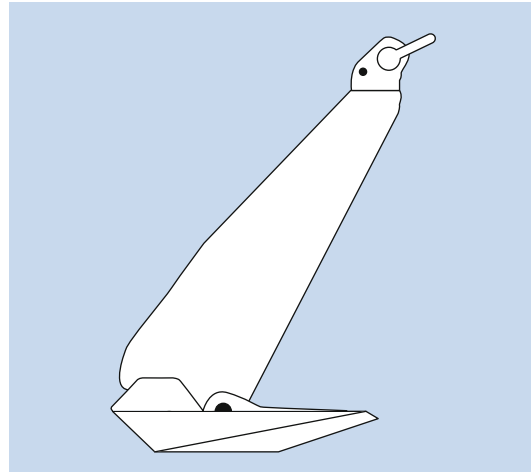
governing equations of the mooring line under tensile deformation and bending deformation (Webster et al. 2012; Ma et al. 2015). The finite element method is used to solve the dynamic problem. Its main advantage is that the nonlinear governing equations can be performed in the global coordinate system without coordinate system transformation. In comparison of the two methods, the equation computational speed of the former is faster, but the accuracy is smaller than the latter.

Composition Materials

The abovementioned mooring line can be composed of a single component material or a combination of multiple components on the basis of different needs. The most common product used for mooring lines is chain which is available in different diameters and grades. Although the steel chain is resistant to abrasion and is not easily damaged, the quality is high and the cost is high. Since the deep water mooring system has strict weight restrictions, the full chain system is generally not used. When compared to chain, wire rope has a lower weight than chain, for the same breaking load and a higher elasticity. The most common combination method is three components: top chain + steel cable + bottom chain. The use of multicomponent mooring lines needs to focus on the length and diameter of each component and the choice of buoys or weights. In this way, the top and bottom chains prevent damage to the top mooring line from long-term friction of the fairlead and subsea portion as well as undulating collisions (Smith and Macfarlane 2001).

Selection of Anchors

The restoring force of the catenary mooring line is mainly caused by its own weight. The bottom of the mooring line usually has sufficient length to contact the sea floor even if the floating structure system is in the worst sea conditions. Therefore, the seabed anchor is only subjected to horizontal forces and does not bear vertical forces (Fig. 2).



Catenary Mooring, Fig. 6 Drag embedment anchor. (From anchor manual 2005)

Drag embedment anchors (Fig. 6) are then employed to resist the horizontal loads, but not for large vertical loads; this is the most popular type of anchoring point available today. The drag embedment anchor has been designed to penetrate into the seabed, either partly or fully, and its holding capacity is generated by the resistance of the soil in front of the anchor (Ruinen and Degenkamp 2001).

Development and Challenges

In general, the traditional catenary mooring positioning system occupies an important position in the mooring positioning system of the mobile platform. However, the influencing factors to be considered in deep-water mooring systems are more complicated than those in shallow water, and their research needs more attention and consideration. On the one hand, the traditional steel anchor chain and wire rope system, with the increase of water depth, its self-weight increases, and horizontal stiffness decreases sharply, while the weight increase leads to the increase of the cost, and it will also cause the multifaceted impact about deployment and recovery of the mooring system. On the other hand, the new concept and new materials for the deep-water mooring system are continuously developed. For example,

polyester ropes with light weight and high strength and weight ratio have replaced steel cables in some cases. The application of the new material can not only reduce the weight of the mooring system, but also its mechanical properties are improved to suit deep water. The lightweight catenary mooring system developed in recent years overcomes many shortcomings of the traditional catenary mooring system in deep sea mooring, but neither the lightweight catenary line nor the traditional can't overcome the excessive deviation of the horizontal level under sea conditions; the greater the water depth, the more obvious this shortcoming. In addition, the use of such mooring systems in deep water requires a large number of mooring lines of larger diameter, which can take up a lot of space on the installation vessel. There are many problems that need to be solved, and the effectiveness and economy of the multiple component systems also need further study.

Cross-References

- [Catenary Anchor Leg Mooring](#)
- [Drag Embedment Anchor](#)
- [Floating Structure](#)
- [Single Anchor Leg Mooring](#)
- [Single-Point Mooring System](#)
- [Spread Mooring System](#)
- [Station Keeping System](#)

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Catenary Mooring Lines

- [Static Analysis Method](#)

Cavitation Test

- [Open Water Test](#)

CBL – Continuous Line Bucket

- [Underwater Mining System](#)

CCS – Collaborative Control System

- [Central Monitoring System](#)

CEM - Cation-Exchange Membrane

- [Salinity Gradient Power Conversion](#)

Center Pipe

► Fiber-Optic Cable

Centimeter (cm)

► Welding Technology

Central Monitoring System

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Synonyms

CCS – Collaborative Control System; OPC – OLE for Process Control

Introduction

The central monitoring system in underwater mining system is designed based on the OLE for process control (OPC) and the Ethernet technologies. The purpose of central monitoring system is to obtain promising performance on monitoring and controlling of all the related subsystems of the entire underwater mining system. In addition, the collaborative control subsystem (CCS) is also an important part of the central monitoring system.

Central Monitoring System

The concept of central monitoring system is the system which can obtain the monitoring and control of all the related subsystems, as well as all the parameters within each subsystem. The central monitoring system is a necessary part in the deep sea mining (Kirkham et al. 2020; Cormier and Lonsdale 2020; Hein et al. 2013; Wedding et al. 2015).

For ensuring the realization of ocean mining process, all the related subsystems, equipment and devices need to be monitored in real-time. That is to say, the central monitoring system is designed for monitoring and controlling all the related subsystems. The central monitoring system is composed of application layer, network layer, and device layer. It uses “OPC + Ethernet” mode to realize the communication connection between the upper computer and related subsystems, and then controls the main equipment of each subsystem to complete the relevant actions according to the instructions of the upper computer, and can realize the real-time monitoring and display of the main operating parameters of each subsystem.

Development of Central Monitoring System

In recent decades, the central monitoring system has developed from a single machine and functional control system to a multifunctional intelligent decision-making system. Generally speaking, the development of platform management system has experienced three stages (Han et al. 2004; Li et al. 2002).

First, the distributed control system with single function was applied before the 1970s. The sensors of the devices were directly connected with the controller or computer, which were directly controlled with simple function.

In the second stage, from the 1970s to the mid-1980s, the fieldbus system and supervisory control and data acquisition (SCADA) system have appeared, and the control system based on fieldbus network has been fully developed.

From the mid-1980s to now, with the development of computer, communication, and control technologies, the demand for comprehensive control of various functional subsystems is more and more obvious, and the technology is becoming more and more mature. Therefore the integrated platform management system emerges as the times require. At present, the integrated platform management system in developed countries has developed from a basic ship monitoring function system to an intelligent control system with large amount of information processing, fast processing speed, and complex decision support.

From the point of view of technical implementation, the traditional platform control system lays functional subsystem cable independently and collects the data by different functional systems. There are many cables in the system, and the wiring is very complex. The binding of control and human-computer interaction makes the console a special one.

The central monitoring and platform management system collects data nearby, which greatly reduces the number of system cables and makes wiring simple. The separation of control system and human-computer interface makes the console become a general one, reduces the number of spare parts, and makes the system secure and reliable. The shared information resources of the whole ship can realize complex closed-loop control and decision-making with high degree of automation. And the flexible control mode of central monitoring and platform management system reduces the staffing and the lifecycle maintenance cost.

Centralized Central Monitoring System

In the early application, the subsystems were discrete units and just simply connected. This kind of structure is also a solidified structure, lacking the ability of self-change and adapting to new products. Generally, the centralized structure has only one master node for information processing, and a master computer is used to complete all information processing and decision control.

The advantage of centralized structure is that only a few computers are needed, and each control module can access the global database. Because the master node grasps the global information and knowledge, the master node using certain intelligent technology may make the optimal system operation decision, and may be the global optimal decision. However, the key to the above advantages is that the scale of the system needs to be relatively simple and stable. For the system with complex structure and function, the speed of centralized processing of all information by the master computer may become a bottleneck that seriously affects the real-time performance of the system, so it cannot meet the requirements of resource scheduling and control for large complex

systems. One of the inherent defects of centralized structure is that once the main control node fails, the whole system will face the risk of collapse. In addition, the disadvantages of centralized control architecture include low fault tolerance, poor scalability, and long development cycle. However, some advantages of centralized control structure are still irreplaceable. In some small systems, this structure is very effective for global stability control, so that it is still applied in some cases. For example, Comau in Italy developed a set of software based on centralized control structure, which is used to deal with all control problems of flexible manufacturing system.

OLE for Process Control

OLE for process control is a set of standards pertaining to automated manufacturing process systems which facilitates industry-wide ease of communication between system controllers and plant equipment. The OLE identifier stands for Object Linking and Embedding, and the whole term is generally shortened to Object Linking and Embedding for Process Control, or OPC (Janke 1999; Li et al. 2005). The standards are of relevance to manufacturing processes based on the Microsoft Windows Operating System (OS) and are governed by the OPC Foundation. OLE for process control was developed in 1996 in an attempt to create a standardized set of process communication protocols based on the Microsoft Corporation's OLE, DCOM, and COM technologies for their Windows OS. The enforcement of the OLE for process control standards system allows ease of integration of automated manufacturing systems in a wide range of industries.

The OLE for process control system, or OPC, as it is more commonly known, has at its core a set of common objects, protocols, and interfaces. Together, these form a standardized communications bridge that ensures that control and execution systems can be moved about or added to a process seamlessly. As technologies in manufacturing and industry advance, so the OPC standards are upgraded and amended accordingly. The management of this process is handled by a group known as the OPC Foundation, which is responsible for its maintenance and implementation.

Prior to the advent of the OLE for process control system, a dedicated application had to be written for each new automated process. This was obviously time-consuming and expensive, and created an environment where industry players and machinery manufacturers could not develop or implement common systems. This led to systems being developed in isolation with commensurate losses in streamlining and efficiency. The introduction of the OLE for process control standards can, thus, be seen as one of the most significant developments in the history of automated process control.

Ethernet

Ethernet is a standard communications protocol embedded in software and hardware devices, intended for building a local area network (LAN). It was designed by Bob Metcalfe in 1973, and through the efforts of Digital, Intel, and Xerox (for which Metcalfe worked), “DIX” Ethernet became the standard model for LANs worldwide. The router with ethernet jacks is shown as Fig. 1.

A network interface card (NIC) is installed in each computer and is assigned a unique address. An Ethernet cable runs from each NIC to the central switch or hub. The switch or hub will act as a relay (though they have significant differences in how they handle network traffic), receiving and directing packets of data across the LAN. This type of networking, therefore, creates a communications system that allows the sharing of data and resources, including printers, fax machines, and scanners.

These networks can also be wireless. Rather than using a cable to connect the computers, wireless NICs use radio waves for two-way communication with a wireless switch or hub. In lieu of

Ethernet ports, wireless NICs, switches, and hubs each feature a small antenna. Wireless networks can be more flexible to use, but also require extra care in configuring security.

Ethernet and OPC-Based Central Monitoring System

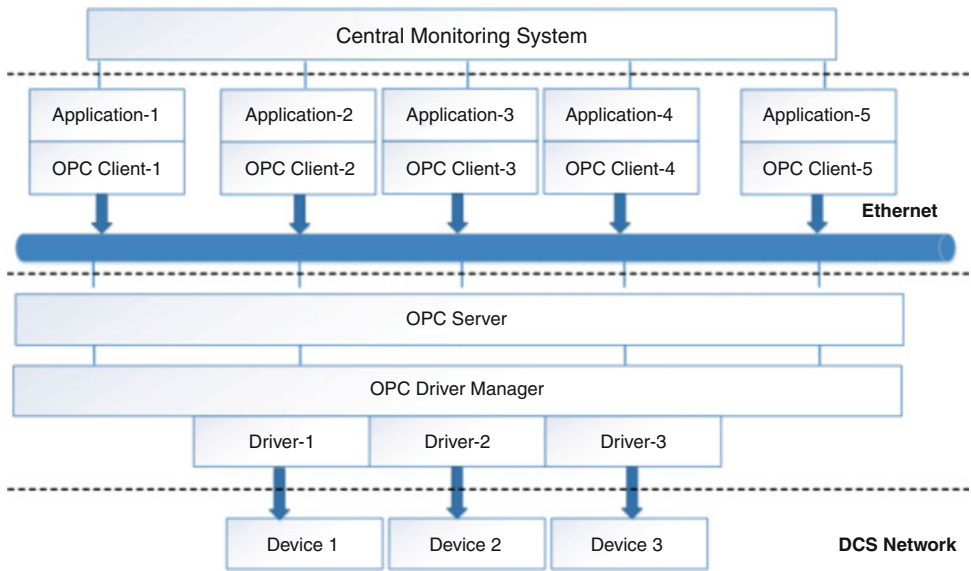
The development of central monitoring system is based on the object linking and embedding for process control (OPC) and Ethernet technologies, that is, all the monitored subsystems, equipment, and devices are regarded as the OPC servers, and all these objects are communicating based on the Ethernet. The schematic of the OPC/ Ethernet based subsystem is illustrated as the Fig. 2. The detailed structure of the system is shown in Fig. 3.

Collaborative Control

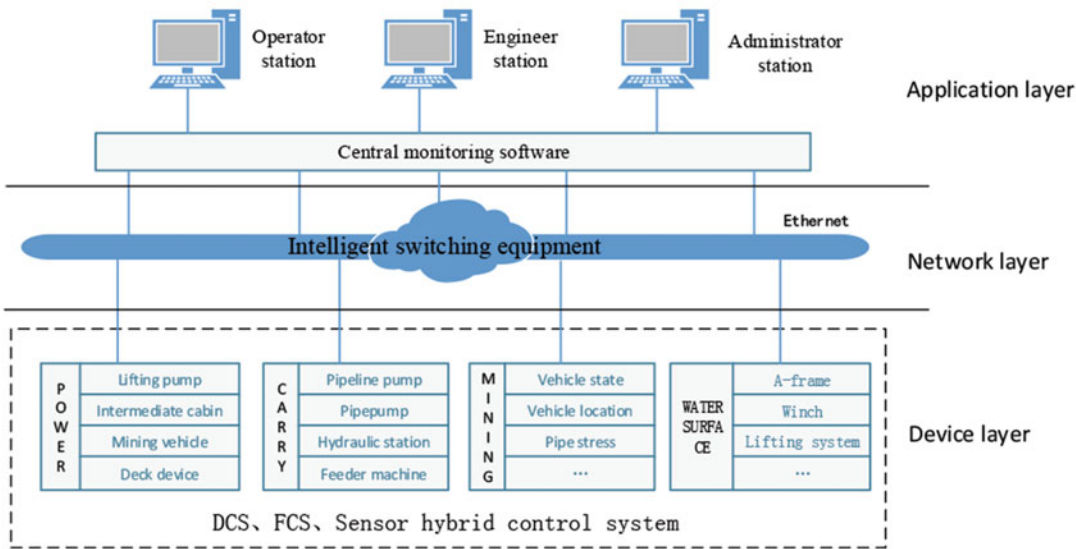
Collaborative control system (CCS), also known as distributed control, is characterized by the fact that there is no master-slave relationship between entities in the system. They coordinate through mutual communication to achieve the goal of the system. The form of collaborative control is common in nature. To ensure the operation of ocean mining system, the collaborative control system is also of great importance. That is because the underwater environment is always very complex and unpredictable. So the descending and lifting processes of the mining vehicle are very complex. This will cause serious consequence for the entire ocean mining system. The collaborative control system obtains the relevant monitoring parameters from the central monitoring system in real time, and transmits the upper computer control instructions to the controlled equipment of A-frame system to control it to complete the relevant actions according to the control instructions. In addition, the collaborative control system has

Central Monitoring System, Fig. 1 Router with ethernet jacks





Central Monitoring System, Fig. 2 Schematic of OPC/Ethernet-based monitoring system



Central Monitoring System, Fig. 3 Detailed structure of OPC/Ethernet-based monitoring system

the function of emergency response, which should be able to realize emergency braking of the main equipment in case of communication failure.

Human Machine Interface

Human-machine interface, also called user interface or human-computer interface, means by

which humans and computers communicate with each other. The human-machine interface includes the hardware and software that are used to translate user (i.e., human) input into commands and to present results to the user.

The human machine interface (HMI) of ocean mining central monitoring system is designed in

detail in this section. The developing environmental is KINGVIEW v7.5. Some main HMIs are listed as Fig. 4.

As shown in Fig. 4, the main operation parameters of all the subsystems, equipment, and devices can be collected and communicated



Central Monitoring System, Fig. 4 (continued)



Central Monitoring System, Fig. 4 HMI of ocean mining central monitoring system based on KINGVIEW v7.5. (a) Login HMI. (b) Main HMI. (c) Monitoring HMI. (d) Mining vehicle control HMI. (e) Lifting pump monitoring HMI. (f) Intermediate cabin monitoring HMI

through the OPC/Ethernet mode. For example, in the main HMI shown in Fig. 4b, the operation mode can be shifted using the button above the HMI. In the monitoring mode, the operation parameters of all the related subsystems can be displayed within each window. However, in the control mode, the control parameters like the lifting speed of mining vehicle, the lifting mode of mining vehicle, and so on, can be set using the corresponding windows. In addition, the alarm function is also integrated into the system. That is to say, when the operation parameters are greater than the upper bound or less than the lower bound, the system will warn the operator to check and eliminate the fault.

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CFD – Computational Fluid Dynamics

- [Analysis of Renewable Energy Devices](#)

Chain Connector

- [Mooring Connector](#)

Change for Reuse

- [Modification for Reuse](#)

Christmas Tree

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Synonyms

[Blowout Preventer \(BOP\)](#); [American Petroleum Institute \(API\)](#); [Tension Leg Platforms \(TLPs\)](#); [Net Present Value \(NPV\)](#); [Internal Rate of Return \(IRR\)](#); [Enhanced Proliferation Control Initiative \(EPCI\)](#); [Down Hole Pressure and Temperature \(DHPT\)](#)

Definition

In petroleum and natural gas extraction, a Christmas tree, or “tree,” is an assembly of valves, casing spools, and fittings used to regulate the flow of pipes in an oil well, gas well, water injection well, water disposal well, gas injection well, condensate well, and other types of wells. A tree provides numerous additional functions including chemical injection points, well intervention means, pressure relief means, monitoring points (such as pressure, temperature, corrosion, erosion, sand detection, flow rate, flow composition, valve and choke position feedback), and connection points for devices such as down hole pressure and temperature transducers (DHPT). ([http://en.volupedia.org/wiki/Christmas_tree_\(oil_well\)](http://en.volupedia.org/wiki/Christmas_tree_(oil_well))).

Background

The subsea tree and wellheads act as important roles in underwater production system. The subsea wellhead system has the same general functions with a usual surface wellhead, which seals and supports casing strings, and simultaneously, the subsea tree after completion and BOP (blowout preventer) stack during drilling are both supported with it.

A Christmas tree system is mainly composed with a group of valves located on an underwater wellhead for providing an effective interface between the underwater production system and well, as shown in Fig. 1. It is also called an X-tree, cross tree, or subsea tree. Christmas tree includes various valves applied to servicing, regulating, testing, or choking the stream of produced gas, oil, and liquid spilling over from the well below.

Christmas tree has various types, which can be applied to water/gas injection or underwater

production. Components of Christmas tree can be different in term of field developments and different projects.

Christmas tree is usually designed mainly according to the codes and API standard below:

- API 6A, Specification for Wellhead and Christmas Tree Equipment
- API RP 17A, Design and Operation of Subsea Production Systems
- API 17D, Design and Operation of Subsea Production Systems
- API RP 17G, Design and Operation of Completion/Workover Risers
- API RP 17H, Remotely Operated Vehicle Interfaces on Subsea Production Systems
- API 5L, Specification for Line Pipe
- DNV RP B401, Cathodic Protection Design
- ASME B31.3, Process Piping
- ASME B31.8, Gas Transmission and Distribution Piping Systems
- ASME BPVC VIII, Rules for Construction of Pressure Vessels
- AWS D1.1, Structural Steel Welding Code
- NACE MR-0175, Petroleum and Natural Gas Industries



Christmas Tree, Fig. 1 The structure of subsea tree
<http://www.zszc.com.cn/product/25.html>

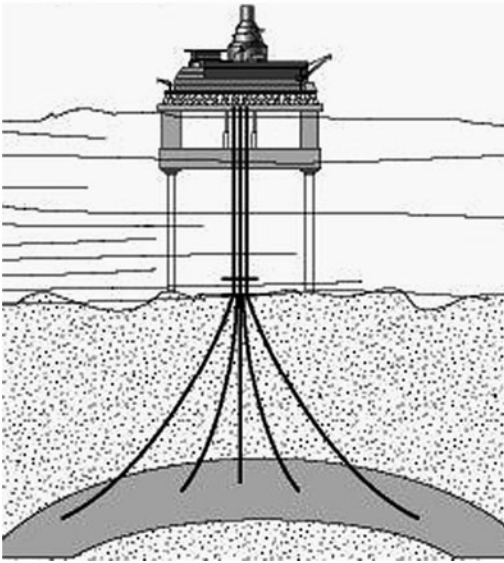
Dry Tree and Wet Tree System

According to the structure form, there are two types of Christmas tree are used in deep-water fields: dry tree and wet tree systems, which are illustrated in Fig. 2.

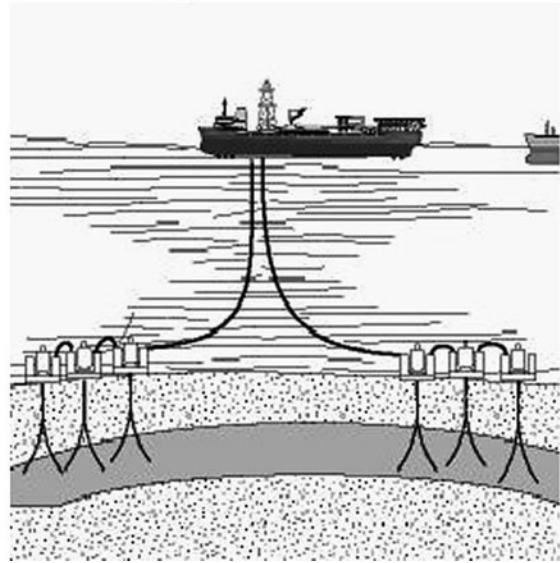
For the dry tree system, tree is close to or installed on the platform, nevertheless, wet tree can be anywhere in a field referring to tie-back method, cluster, and template. Dry tree platform has a central well bay for the surface tree, supplying convenient access to the wells for recovery and workover. Spars and TLPs (tension leg platforms) are usually used in a dry tree system.

The size of the central well bay of dry tree system is depended on spacing and well count, which is placed around by topside equipment. The surface tree is projected for full reservoir shut-down pressures. A large subsea manifold is

Dry Tree System



Wet Tree System



Christmas Tree, Fig. 2 Dry tree and wet tree system (Yong Bai and Qiang Bai 2019)

required to install on deck, and a skiddable rig is required for individual well interventions.

For wet tree system, trees and other related parts are exposed to the seabed environment. In deep-water field, for well completions, the wet tree system usually uses remotely controlled Christmas tree installation tools, but in shallow water, divers can deal with installation and operation. The wet tree platform has a central “moon-pool” for the installation of Christmas tree and marine riser. If its size is suitable, the wet tree system also can install other equipment, such as the BOP and subsea manifold. Wet tree system is suitable for widespread reservoir structures. They provide field expansion flexibility and a degree of vessel with simplified riser interfaces, but at a high cost of drilling and workover.

In recent years, operators have to reassess their strategies of rapid expansion in ultra-deepwater areas due to the technical difficulty and commercial competitiveness between dry tree and wet tree system. Globally, more than 70% of the wells in deepwater projects that are wet tree system. The data shows the industry’s confidence in the wet tree system.

Compared with wet tree system, the dry tree system requires a motion-optimized hull to adapt

the riser system. They are deemed to be limited in development flexibility and depth. Globally, most underwater oil field developments employing wet tree system. The dry tree system is not adapted to deep-water and ultra-deepwater situation in spite of widespread development in shallow to medium depth.

Systems Selection

In the design process, an objective and thorough assessment of dry and wet tree system option should be considered, including cost, risk, and flexibility, to ensure that the development concept that best matches the reservoir characteristics is selected.

The problem that operators often face is commercial metric, such as NPV (net present value) and capital cost, which are inconclusive. In the basic case of development examined, the commercial metric is displayed to generally prefer the wet tree system over the life cycle of the system, but the advantage to the dry tree system is not obvious. In this case, the operator must thoroughly evaluate the other significant different factors before making the final selection.

Through experience and technical analysis, the optimal Christmas tree system matching reservoir characteristics can be selected. The following are the basic selection points:

- Extrinsic factor: These factors are in the form of operator preferences, innovative thinking, project risk, project management, and people (each person's evaluation method may be different).
- Economic factor: Estimated net present value, IRR (Internal Rate of Return), project schedule, project cash flow, and possibly EPCI (Enhanced Proliferation Control Initiative) proposals (if any options are available at the time of selection) will undoubtedly be the key drivers of this choice.
- Technical factor: These factors are driven primarily by reservoir depletion plans and means, operating philosophy, concept maturity, field worldwide location and feasibility, reliability, and industry readiness.

References

<http://www.zsxc.com.cn/product/25.html>

Yong Bai, Qiang Bai (2019) Subsea engineering handbook, 2nd edn. Elsevier, Cambridge, MA, pp 697–729

CIMS (Computer Integrated Manufacture System)

- [Introduction to Shipbuilding \(Shipyard\)](#)

Closed-Containment Systems

- [Modern Aquaculture Structures](#)

CNC (Computer Numerical Control)

- [Introduction to Shipbuilding \(Shipyard\)](#)

Coastal and River Engineering (NRC-OCRE)

- [Ice Tank Test](#)

Cold Regions Research and Engineering Laboratory (CRREL)

- [Ice Tank Test](#)

Collector and Crusher

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Synonyms

[OCTM – Ore Collecting Test Machine](#)

Definition

Collector: Collecting attached ore or broken ore particles from the seabed.

Crusher: Crushing the collected ore or cutting the ore from the surface of the sea bottom to a specified size.

Introduction

In the process of seabed mining, the collector and crusher of the deep-sea mining vehicle is responsible for correspondingly collecting and crushing the ore covered in the seabed. The process and technology of deep-sea mineral resources collection and crushing mainly depend on the occurrence state of mineral resources in the seabed and the characteristics of seabed topography and geomorphology. It is easy to be collected for the

polymetallic nodules on the surface of sediments. Usually, the polymetallic nodules are first collected to the mining vehicle and then broken to required particles by crusher in the mining vehicle. Up to now, there are three representative methods that have been used in deep sea experiments, including mechanical-type collection, hydraulic-type collection, and mechanical-hydraulic-type hybrid collection. The conventional single toothed roller crusher is used to crush the polymetallic nodule. For the polymetallic sulfides and cobalt-rich crusts consolidated in the hard bedrock, the ore body shall be first excavated and stripped from the bedrock and then collected to the mining vehicle. In general, the ore body shall be broken to a certain particle size in the excavating process. As of this date, the representative methods that have been applied to the deep sea test mainly include mechanical mining crushing and hydraulic collection.

Collector and Crusher of Seabed Polymetallic Nodules

Polymetallic nodules occurred on the surface of seabed sediments with the depth of 4000–6000 m in Pacific Ocean are usually in a semi buried state (Kaufman 1980). Most of polymetallic nodules are spherical or ellipsoidal with a particle size of 2–10 cm and a density of about 2100 kg/m³.

It needs a certain degree of stripping force for the collection to strip the polymetallic nodules from seabed sediments. According to the measurement, the stripping force from the sediment with a shear strength of 3 and 6 kPa is, respectively, 25 N and 32 N for a nodule with the weight of 0.14 kg and the diameter of 6.4 cm. While the maximum force per unit area is 0.02 MPa when the total burial depth is doubled. Considering the insurance factor of 1.5 times, the stripping pressure is 0.03 MPa in the design (Wang 2015).

Collection Technology of Seabed Polymetallic Nodules

Up to now, hundreds of collection principles and institutional patents are invented, but only

mechanical collection principle, hydraulic collection principle, and hydraulic-mechanical combination collection principle are valuable in economy and technology.

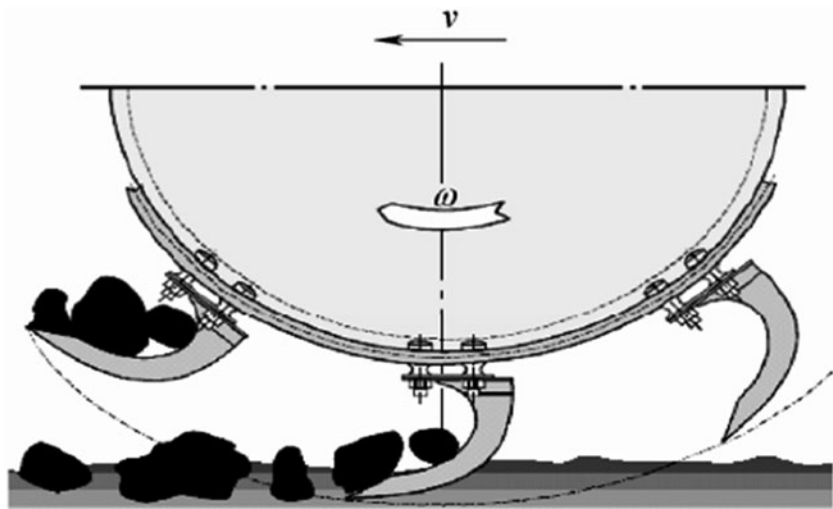
Mechanical Collection Technology

Mechanical devices are used to collect the polymetallic nodules in the mechanical type, which is originated from agricultural potato harvesting devices. The principle and structure are relatively mature and efficient. However, the mechanism of the mechanical collector is complex, and its digging teeth are easy to be damaged result in blockage, and it is difficult to unload quickly.

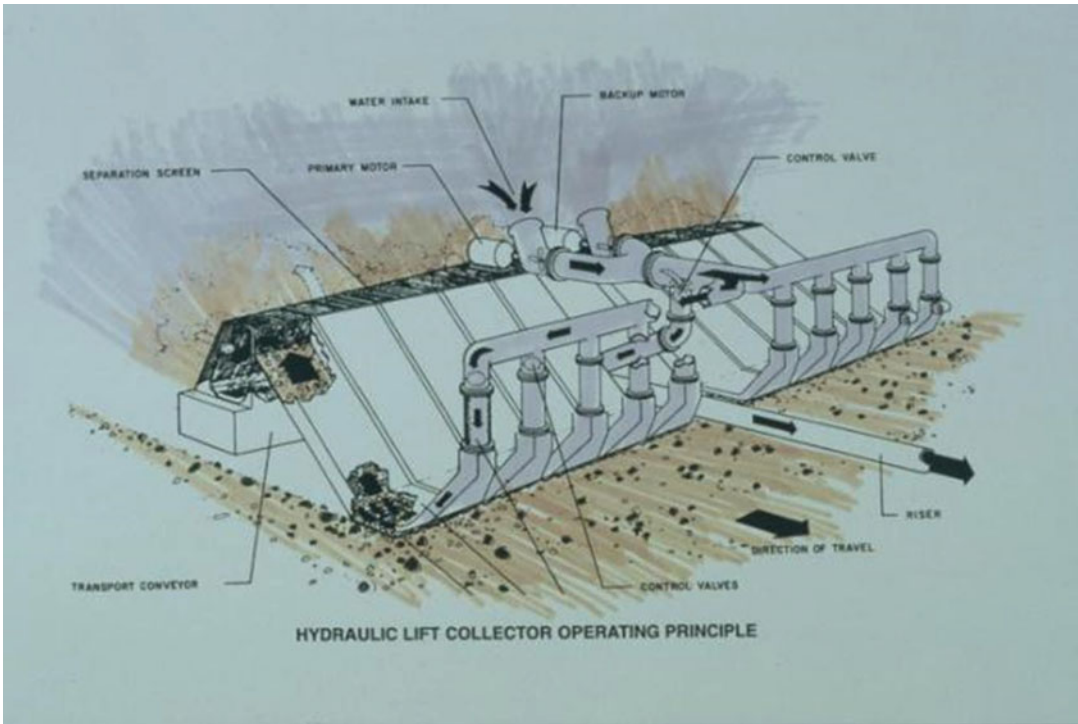
In addition, the mining resistance of the collector is large to increase the walking load of the seabed mining vehicle. A mechanical nodule collector invented by Ocean Minerals Company (OMCO) is shown in Fig. 1. The nodule is excavated from the seabed and transported to the seabed mining vehicle by means of the rotary chain-tooth collector (Welling 1981).

Hydraulic Collection Technology

The hydraulic collection technology is researched and developed to separate and move the nodules on the surface of seabed sediments with water flow. Three main types of hydraulic collection are developed that are jet hydraulic collection, suction hydraulic collection, and lifting type. The principle of water jet hydraulic collection is typically based on the wall attached nozzle effect of the water jet. Under the negative pressure generated by the low-pressure water jet of the flat wall attached nozzle, the nodule is loosened from the sea floor and the surrounding seawater is pumped simultaneously, the nodule is lifted and transported to the inclined conveying pipe. The hydraulic suction collection uses the pump to produce suction in the pipeline, sucks the sediment around the pipe orifice and the nodule together, and lifts to a certain height. The principle of hydraulic lifting collection is that two rows of water jets are used from the front and back inclined to the seabed nozzle at a certain height from the bottom to wash and lift the nodule, and then the nodule are input into the



Collector and Crusher, Fig. 1 Mechanical collection technology (Welling 1981)



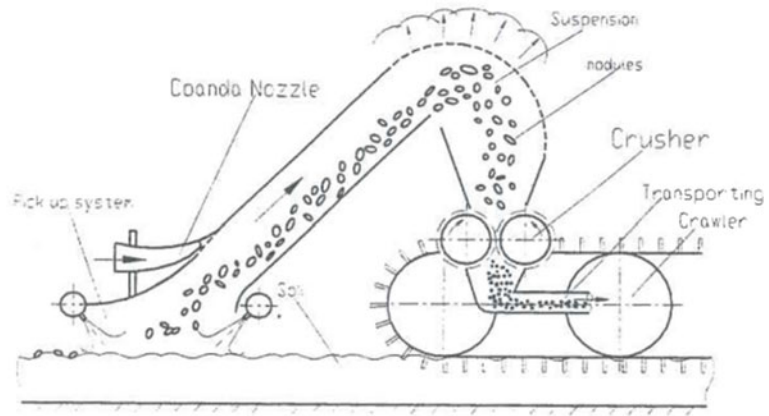
Collector and Crusher, Fig. 2 Hydraulic collector of nodule used in sea pilot mining system test hold by OMI (Brockett 1999)

transmission pipeline. This collection type was proposed by Berlin Institute of underwater engineering in Germany.

Figure 2 is the schematic diagram of the hydraulic nodule collection device used in Ocean Management, Inc. (OMI) seabed mining

Collector and Crusher,

Fig. 3 Hydraulic collector of nodule used in Chinese lake test mining vehicle (Liu and Yang 1999)



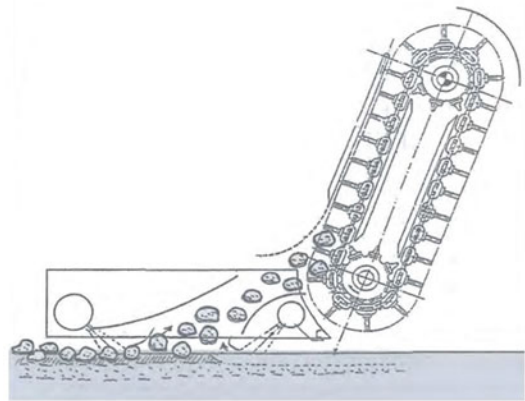
system test. Its working principle is to use high-pressure water jet to blow the nodule from the seabed and collect it on the seabed mining vehicle by using water flow (Brockett 1999).

The collector is a hydraulic collection mechanism with double row of nozzles for washing and mining and negative pressure conveying of the wall attached nozzles of the 1000 m class lake test polymetallic nodule mining vehicle. The schematic diagram is shown in Fig. 3 (Liu and Yang 1999). Structurally, two rows of nozzles are arranged inclined to the sea bottom at a certain height from the sea bottom. Firstly, the water jet is generated by nozzles to flush the nodule out of the sedimentary layer. Secondly, the jet washes off part of the sediment and lifts the nodule under the action of the formed rising water flow. Thirdly, the lifted nodules are sent to the crusher feed port under the action of the forward movement of the ore collecting device and the negative pressure generated by the nozzle attached to the wall.

Hydro-Mechanical-Type Collection Technology

Hydro-mechanical-type collector combining the double-row nozzle blanking mechanism with the gear chain conveying mechanism is to avoid the disadvantages of mechanical tooth digging and high energy consumption.

The working principle diagram of hydro-mechanical-type connector developed by Siegen University (Handschuh et al. 2001) is shown in Fig. 4. Hydraulic power is used to blow up the nodules, and mechanical chain-plate is used to



Collector and Crusher, Fig. 4 The working principle diagram of hydro-mechanical-type connector (Handschuh et al. 2001)

transport the nodules to the seabed mining vehicle.

Mechanical-hydraulic-type connector, which combines mechanical-type connector and hydraulic-type connector, has the advantages of low acquisition resistance, clean flushing and small power of tooth chain transmission, etc. However, it is difficult to determine the bottom height of the collector, hydraulic system parameters and channel shape, so it needs to be corrected through repeated tests.

In the series of sea trials carried out in the 1970s, the mechanical collection mechanism adopted by OMCO failed to continue for only 10 minutes due to the jam of the crusher. The hydraulic collection mechanism adopted by OMI

and Ocean Mining Associates (OMA), respectively, with the principle of water jet collection and the principle of suction and lift collection has a better effect, which has a higher collection efficiency than mechanical collection. Currently, the mining vehicles developed by China, Belgium, and South Korea all adopt hydraulic collection mechanism, and the test results show that they have achieved good results (McFarlane and Brockett 2008; Kim et al. 2019; GSR 2019).

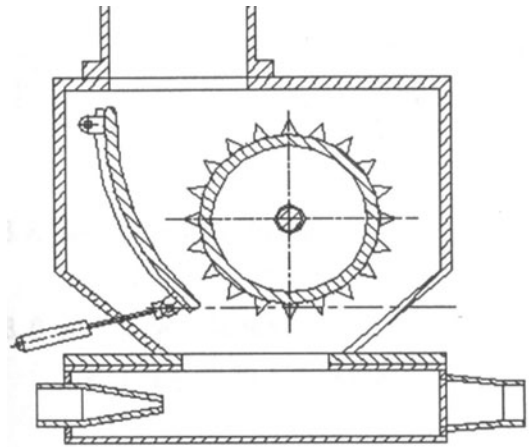
These collecting devices can collect the nodule from the seabed to the mining vehicle. But for industrial application, a good design of nodule collecting mechanism should not only have high collection rate and low energy consumption, but also be able to minimize the disturbance of seabed sediment. Thus, the collecting device is required to minimize the damage to the seabed ecosystem and meet the requirements of the environmental protection regulations. Due to the lack of sufficient test and application verification, it is still a long way to obtain the most effective scheme for commercial application.

Crushing Technology of Seabed Polymetallic Nodules

According to the requirements of simple structure, light weight, and low crushing object strength (average compressive strength 5 MPa) (Wang 2015), it is more suitable to choose single toothed roller crusher. A typical crusher is shown in Fig. 5, with hydraulic overload jaw opening plate and linkage bottom door opening discharge mechanism.

Collector and Crusher of Seabed Polymetallic Sulfides

Seabed polymetallic sulfides originate from the mixing of hydrothermal fluid and cold sea water that erupts from the seabed, which is large in the seabed with the thickness of up to 10 meters or tens of meters. According to the mechanical properties of a large number of samples, the fracture characteristics of seabed polymetallic sulfides are similar to coal. Toughness and plasticity of seabed polymetallic sulfides are similar to salt and

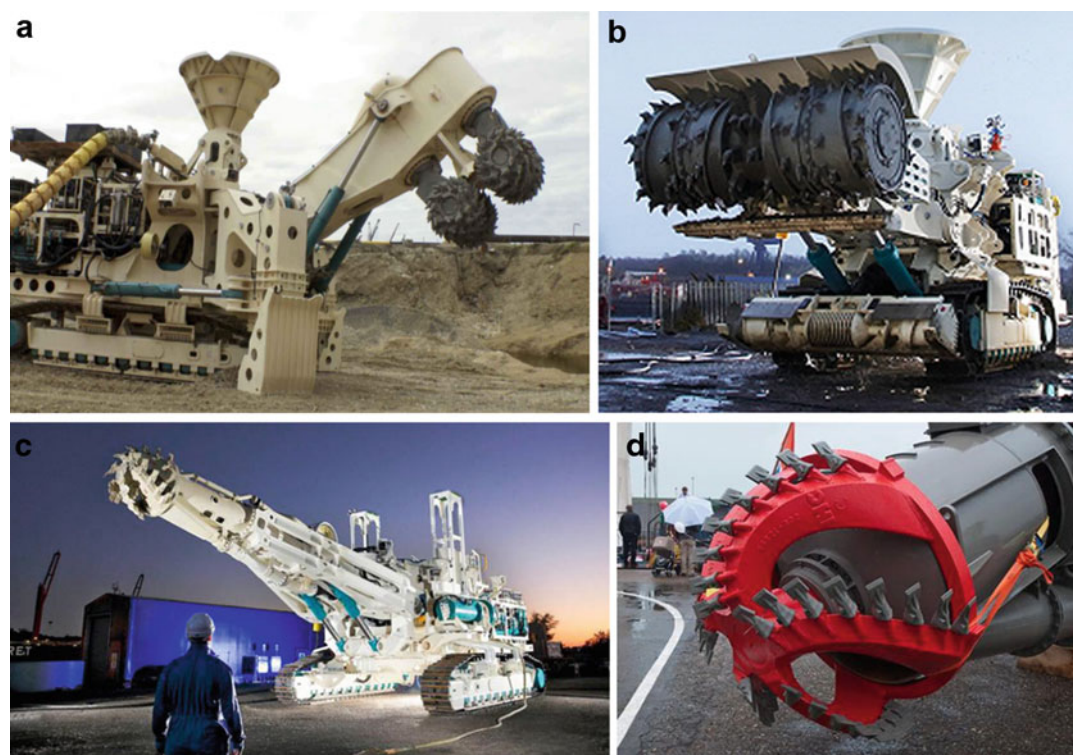


Collector and Crusher, Fig. 5 Single tooth roller crusher used in China Lake test mining vehicle (Wang 2015)

potassium carbonate, and the axial compression strength of seabed polymetallic sulfides is less than 40 MPa.

Polymetallic sulfides deposit on the sea bottom surface can be collected easily by traditional mechanical or hydraulic machinery. In view of the distribution characteristics and the mechanical properties of polymetallic massive sulfide are similar to those of coal, the existing mature technology and equipment of land hard coal can be used for reference for its collection. Mechanical or other acoustic devices need be used to break the sulfide ore block and then collect it on the process flow.

According to the commercial mining system proposal of seabed polymetallic sulfide from Nautilus Minerals, the subsea operating device consists of three operating vehicles, namely, auxiliary miner (Fig. 6a), bulk miner (Fig. 6b), and gathering miner (Fig. 6c). The auxiliary miner opens up the mining belt, prepares the mining working face, and transports the excavated and broken ore grains to the vicinity of the mining vehicle. The bulk miner excavates and crushes ore bodies on the working surface. The gathering miner collects the ore grains excavated by bulk miner, and the ore grains cut by auxiliary miner, and then the gathered ore grains are pumped to the flexible hose by an underwater pump to the surface mining ship through riser (Nautilusminerals 2015).



Collector and Crusher, Fig. 6 Auxiliary miner (a), bulk miner (b), gathering miner (c), and cutter suction dredger cutter head (d) (Nautilusminerals 2015)

In view of the characteristics of polymetallic sulfide deposits in the waters of Okinawa, an ore collecting test machine (OCTM) was developed to verify the mining feasibility of polymetallic sulfides. OCTM adopts the integrated technology of crushing and collecting with the integrated mining head so as to reduce the mining time and the cost related to the auxiliary ship equipment and improve the productivity of offshore mining operations.

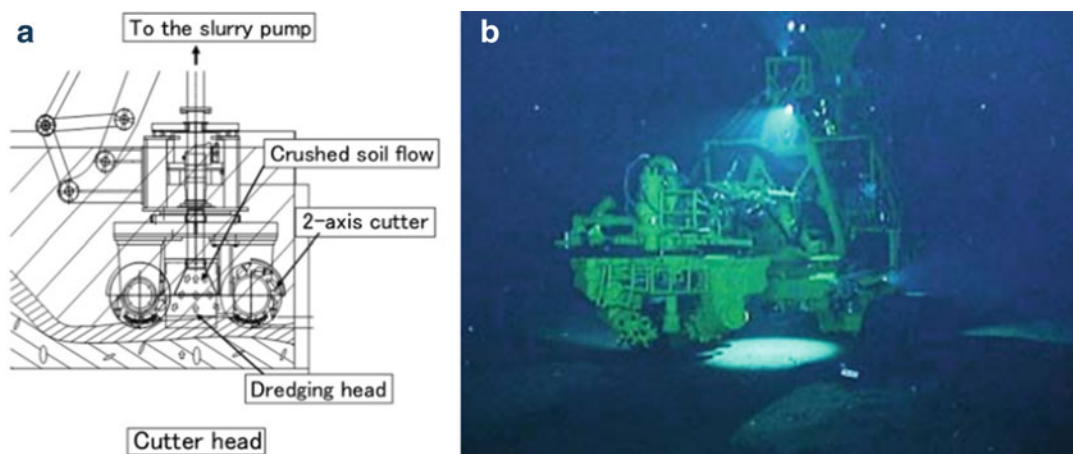
The spiral drum cutting technology and mechanism by mature coal mining on land is used for reference to cut and crush the polymetallic sulfide ore body in auxiliary miner, bulk miner, and OCTM. The collector of gathering miner and OCTM is based on the corresponding principles and schemes of river dredger and seabed Diamond miner, which is shown in Fig. 6d.

OCTM with two single-arm drum spacing reversing arrangement combining mining and collecting was developed through laboratory

principle tests and submarine collection head tests. A dredging head is arranged vertically in the center of the two drums. The dredging head is driven by slurry pump through the pipe. When it works, the cutting drum cuts the ore material, and the ore particles are collected in the center of the two cutting drums (Masuda et al. 2014). The schematic diagram of the mining head is shown in Fig. 6a and the mining head underwater operation diagram is shown in Fig. 7b. In 2017, the polymetallic sulfide mining test at a depth of 1600 meters was completed, and a total of 16 t polymetallic sulfide ore was collected (Ishiguro et al. 2013; Masuda et al. 2014; Okamoto et al. 2018).

Crusher of Seabed Cobalt-Rich Crusts

Cobalt-rich crusted deposits are mainly distributed on the slopes and top surfaces of seamounts, mid-ridge and platforms in depth of 400–4000 m,



Collector and Crusher, Fig. 7 Polymetallic sulfide mining head of ore collecting test machine (Okamoto et al. 2018)

attached to bedrock with crusts of 2–20 cm thickness. Their mechanical properties are similar to terrestrial hard coal. Regardless of the terrain, the coal cutter method can be used to peel and break the crust from the bedrock. However, the geological conditions for the occurrence of cobalt-rich crust deposits are extremely complicated. Some of the mining areas are relatively flat in the 2–3 km zone at the edge of the mountain, and most of the cobalt-rich crust mining areas are on slopes with slopes of 10–20 degrees (Qiu 2002; T. Yamazaki 2004). The micro-topography fluctuates greatly and the crust thickness is very thin with an average thickness of less than 10 cm. The key problem of collecting cobalt-rich crusts is how to remove the crusts effectively from the bedrock. The mining of cobalt-rich crust is limited by two key technologies. Firstly, the mining mechanism will fully excavate the crust ore body attached to the bedrock to ensure a high recovery rate. Secondly, the bedrock should not be cut off as much as possible to reduce the dilution rate. Generally, the recovery rate is generally higher than 80% and the dilution rate is lower than 25%.

According to the current information, the cobalt-rich crusts with commercial mining value are mostly in the form of thin-layer plate-like crusts covered by hard basalts and other types of hard bedrock. At present, researchers in the United States, Japan, Russia, and other countries have carried out the research on the principle



Collector and Crusher, Fig. 8 Cobalt-rich crust mining head (CIMR 2018)

scheme of the mechanical crushing and stripping method for cobalt-rich crusts, but there is no report on the in-depth design, manufacturing, and test work.

A seabed cobalt-rich crusts mining equipment for sampling test was developed in China. In the test equipment, the crusher is designed to be double arm single spiral drum from the successful experience of the spiral drum cutting technology of land mining, and the slurry pump is used as the collection mechanism according to the river dredging technology. When the equipment worked on the seabed, the spiral drum (as shown in Fig. 8) excavates the crust, and then the ore particles, crushed by spiral drum, are collected into the sampler by the slurry pump.

In September 2018, the mining sampling equipment was tested at the depth of 2019 meters in the cobalt-rich crusts contract area of COMRA. About 150 kg of crust particles were collected in four tests (CIMR 2018). The sampling equipment verifies the feasibility of collecting cobalt-rich crusts, and provides a self-adaptive collection method of cobalt-rich crusts. However, it is still necessary to solve the technical problem of efficient cutting, stripping, and acquisition of the crusts which are thin and crispy, distributed in hard bedrock.

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Combined Vertical, Horizontal and Moment Loading

► Shallow Foundations

Complete and Partial Dismantling

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Synonyms

[Ship dismantling](#); [Ship recycling](#); [Vessel dismantling](#); [Vessel recycling](#)

Definition

Nowadays, there are different ways of dismantling ships and offshore structures, i.e., complete dismantling and partial dismantling. For ship recycling, most the end-of-life vessels are usually sold to the international markets for complete

dismantling. It is a common way to export obsolete vessels to shipyards for recycling in both developed and developing countries as a way of resource recycling and reusing. However, some vessels, such as cargo ships and tankers, are partially dismantled and transformed into fishing farming, which would be beneficial to the environment. For offshore structure recycling, when an offshore field becomes unviable, the offshore infrastructures require decommissioning, by using one of the following alternatives – complete removal, partial removal, or remote reefing (Chandler et al. 2017).

Scientific Fundamentals

Historical Development

Currently, the core ship dismantling locations have been moved to developing countries, including India, Bangladesh, Turkey, Pakistan, and China (Mikelis 2008; Deshpande et al. 2012; Ormond 2012; Hiremath et al. 2015; Rahman and Mayer 2015; Du et al. 2017, 2018). However, the ship dismantling industry usually has severe negative implications on the environment, human safety, and health because of the hazardous materials and chemicals involved (Sonak et al. 2008; Neşer et al. 2008; Harris and Kahwa 2003). Recycling end-of-life vessels in an environmentally friendly manner is a major challenge faced by ship owners, shipbreaking yards, as well as government agencies worldwide (Du et al. 2018). Moreover, problems generated by the insufficiencies of current ship dismantling practices have consequences for not only the environment but also for occupational safety and the health of the workers.

Meanwhile, there are thousands of offshore oil and gas facilities and platforms all over the world, as well as subsea infrastructure, pipelines, wells, etc. Many of those offshore engineering infrastructure have been in service for several decades and are due or will soon be due to be decommissioned (Chandler et al. 2017). Typically, offshore structure decommissioning procedure includes the following steps – decommissioning planning and engineering, permitting and regulatory compliance, platform preparation, well plugging and abandonment,

conductor removal, mobilization and demobilization of derrick barges, platform removal, pipeline and power cable decommissioning, materials disposal, site clearance and verification, and post-job activities. Herein, this part aims to introduce complete and partial dismantling of ships and offshore structures that are end-of-life.

Complete Dismantling of Ships

The Sixth Meeting of the Conference of Parties to the Basel Convention in 2003 had adopted the Technical Guidelines for the Environmentally Sound Management of the Full and Partial Dismantling of Ships (hereafter as The Guidelines) as the technical guidelines for the hazardous wastes and their disposal during ship dismantling. The Guidelines had also been considered and adopted by IMO for IMO Guidelines on Ship Recycling in 2004. Meanwhile, the Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships, 2009 (the Hong Kong Convention), also aims at ensuring the environment, safety, and health issues. These guidelines aim to enable ship dismantling facilities attain environmental sound management (ESM) by providing information and recommendations on procedures, processes, and practices for the countries during ship dismantling processes. Therefore, it is a major challenge to recycle end-of-life ships in both environmentally sound and safe manner ways.

As mentioned by the Basel Convention, ESM is defined as “taking all practicable steps to ensure that hazardous wastes or other wastes are managed in a manner which will protect human health and the environment against the adverse effects which may result from such wastes.” The Guidelines provide an overview of the elements to be considered for the complete dismantling of ships in order to achieve an environmentally sound ship dismantling, including preparations on the ship, ship dismantling facility, environmental management plan, etc. In this section, the procedures for different methods of ship dismantling are introduced and studied. The procedures include pre-demolition removal and preparations prior to ship dismantling operations, arrangement of ship dismantling facility, main works of ship dismantling operations, and the environmental management plan.

Environmental Management Plan

At the beginning of ship dismantling, an environmental management plan (EMP) is usually required to achieve the aim of environmental sound management for ship dismantling. The EMP includes the assessment of the potential environmental impacts during ship dismantling by conducting an environmental impact assessment, which helps to identify the environmental aspects and the environmental goals to be set for the facilities. The EMP, which is an all-encompassing document, covers all aspects of environmental issues at a macroscopic range, including the assessment of potential environmental impacts, the potential preventative measures, and environmental management system (hazardous materials management plan, contingency preparedness plan, monitoring plan, etc.).

Pre-demolition Removing and Preparations

Prior to ship dismantling operations, firstly, whatever method the yard uses to dismantle the ship, the ship should be fixed or moored well to minimize displacement of the ship. Thereafter, some preparatory procedures should be implemented on the ship, such as overall checking onboard, collecting related information, and planning of ship dismantling; deployment of access gangway and warning signs; open and free any gas; oxygen and explosion tests in sealed cabin; deployment of oil booms; emergency facilities equipment; etc. More details are shown in Table 1.

Ship Dismantling Facility Arrangement

It is very important to make sure that all of the dismantling facilities and equipment should be working properly and the dismantling yard should be cleaned and comprises certain key

Complete and Partial Dismantling, Table 1 Preparatory procedures before main work of ship dismantling

Item	Action	Purpose
Prepare inventory list of hazardous materials and polluting wastes	Make a detailed ship dismantling plan Identify, quantify, and locate different types of hazardous materials and wastes on board	To arrange the sequence and nature of the work to be executed and ensure human health and safety and ESM
Remove and clean the liquids, e.g., fuels, oils, etc.	Remove the hazardous materials, e.g., asbestos, Polychlorinated biphenyl (PCBs), tributyltin (TBT) paints, etc. Clear all the residual materials before cutting process Remove crew's furniture, storage, tools, and other sundries Pump out and drain off the oil in oil pipes, oil tanks, and lubricant systems, and clean the tanks Remove harmful and harmless insulation materials Remove other dangerous (toxic, flammable, and explosive) items such as dump oil, paints, high-pressure cylinder, etc.	To ensure that the ship is ready for dismantling in safe and clean conditions
Removal of equipment	Remove consumable and loose equipment Remove reusable components	To ensure that the cutting operations are more efficient
Securing	Deploy safe access to all areas, compartments, tanks, etc. ensuring breathable atmospheres Create safe working conditions for hot work, including cleaning/venting, removal of toxic or highly flammable paints from areas to be cut, and testing/monitoring before any hot work is performed Deploy emergency facilities equipment	To ensure that working procedures and operations are undertaken in a safe manner

Note: Adapted from the Basel Convention (2003)

functionalities. The equipment includes different kinds of cranes, forklift trucks, loaders, transport vehicles, cutting equipment, etc. Whichever method used for dismantling, the shipbreaking yard should include various zones for different purposes during dismantling. For an environmentally sound design of a yard, it is important to discern what activities take place in which zone, and what associated hazards are to be considered and prevented through environmentally friendly design. The zones shall comprise certain key functionalities, including containment for spills and leakages of hazardous materials/liquids, certain areas for dismantling blocks and sections into small pieces, specially equipped warehouse and appropriate containment for hazardous and toxic materials removal (such as asbestos), temporary regions for benign materials and hull blocks/sections, secure storage rooms/areas for hazardous wastes, certain regions for fully processed materials and equipment that can be reused and recycled, and proper disposal facilities for persistent organic pollutants.

Main Works of Ship Hull Dismantling

The main work of ship hull dismantling follows the pre-demolition removal and cleaning, and the layout of dismantling zones and facilities/equipment. Various techniques applied for ship dismantling have different costs and different degrees of environmental and social impacts. Currently, there are mainly four methods for ship dismantling. These include beaching, slipway or landing, alongside or pier breaking, and dry dock, as shown in Table 2.

During the first dismantling, the ship is usually broken down into blocks or sections. Piece cutting is needed to cut the blocks into pieces on site for secondary dismantling, especially for alongside or pier breaking method. The workers should clean up the residual oil and wastes on the blocks so that they can be disposed of separately. As a requirement, the blocks are cut into steel sheets, profiled materials, and waste steel and, respectively, placed in a certain place (Du et al. 2018). A procedure for dismantling a LNG (liquefied natural gas) vessel, which applies pier breaking method, can be seen in Fig. 1.

Complete Dismantling of Offshore Platforms

Offshore oil and gas platforms are aging gradually around the world. Those end-of-life offshore structures are decommissioned and removed by ways of complete dismantling and partial dismantling. For complete dismantling of offshore platform, the removal procedures include platform preparation and cleaning, conductor removal, removal of deck and equipment, complete removal of jacket or hull, site clearance, etc.

Platform Preparation

For platform preparation and cleaning, the main procedures are as follows.

- Prepare the facilities for removal.
- Shut down the facilities and equipment.
- Topside inspections.
- Underwater inspections, conducted by divers and remotely operated vehicles (ROVs).
- Plug and abandon the wells permanently, and abandon the pipelines.

When removing the topside of the platform, the following steps shall be carried out.

- Flush and clean the tanks, process equipment and pipes, and dispose of the residual hydrocarbons on the platform.
- Remove the equipment.
- Cut the pipes and cables.
- Divide the platform modules into individual units.
- Weld padeyes and remove obstructions for deck module lifting.
- Reinforce local structure for lifting.
- When removing the jacket, marine growth can be removed from the structure at sea if possible.

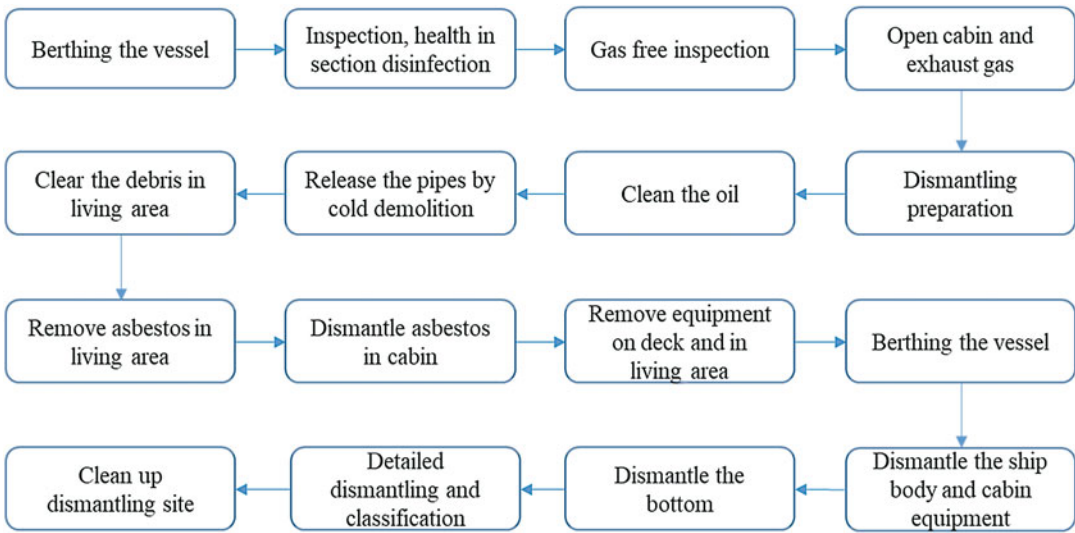
Conductor Removal

The conductors can be removed during platform decommissioning operations or prior to the arrival of heavy-lift vehicle. The conductors can be severed by using one of the following alternatives: explosive, abrasive cutting, mechanical cutting, and diamond wire cutting. When pulling the conductors, the following methods can be used:

Complete and Partial Dismantling, Table 2 Comparison of different ship dismantling methods

Ship dismantling method	Procedures	Advantages/disadvantages	Key locations
Beaching	The ship is driven to a tidal flat on a high magnitude tide and then pulled up the beach as far as possible The items and steel are removed from the ships by gas or oxygen cutting All operations take place directly on the beach in a relatively small and congested area The winches drag the steel pieces up the beach Large blocks may also be cut from the ship, released onto the mudflats, and dragged individually by the winches Once onshore, everything is cut into smaller pieces progressively and then taken from the yard by lorry or truck	Does not require investments in infrastructure or technology The moving cranes and motorized winches can be reused Labor-intensive Unsuitable for the use of heavy lifting gear Not allow for containment of hazardous substances Any spills of oil or cargo remaining would be swept out to sea and cause environmental issue	The regions with high tidal variation and wide mudflats Chittagong in Bangladesh Alang in India Gadani in Pakistan
Slipway or landing	A modification of beaching method essentially, with some crucial differences The ship is driven onto the shore or on a concrete slipway connecting shore and sea Ship sections are removed by a mobile crane working from the shore, and the ship is pulled onto the shore gradually Steel pieces are also removed from the ship by the mobile crane After the ship is lightened, it will be dragged up the shore for further dismantling A temporary quay or semipermanent jetty may also be added	Comparing to beaching method, this method is comparatively safe and environmentally friendly Any accidental spillages can be contained due to the lack of racing tides	Typically used in areas where the tidal flow is low and easily predictable Aliaga Turkey Some other EU Countries
Alongside or pier breaking	The ship is immobilized on a wharf or quay in a sheltered harbor or river The ship is secured alongside in the sheltered waters The superstructure is dismantled from top to bottom and from outside to inside The cutting usually follows the welding lines Usually uses cold dismantling method for oil pipes It is mandatory to drain the remaining oil, gas, and water inside the equipment and conduct gas-free check before cutting during engine room dismantling Dismantle the connection pipe and parts with the equipment at first Maintain the ship's balance at all times A crane removes the hull from the top to the bottom The last piece (such as the double bottom) may be lifted or sent to a dry dock for final cuttings	Results in pollution on alongside, which can be monitored and controlled It is a safety and environmental sound way	Usually applied in calm water China Myanmar Few places in Europe
Dry dock	The ship is driven to an enclosed, flooded dock, and the water is pumped out The ship is then dismantled piece by piece On completion, the dock is cleaned and flooded again for the next ship	With a minimum risk of environmental pollution The cost is the highest Rarely used presently due to its high cost	The USA and Europe practice this method Leave Sley International's facility in Liverpool, UK

Note: Adapted from Hossain (2017), Demaria (2010), and Du et al. (2018)



Complete and Partial Dismantling, Fig. 1 Ship recycling procedure for LNG (Du et al. 2018)

casing jack, drilling rig, platform crane, and the crane on heavy-lift vehicles.

Deck and Module Removal

The removal of topsides is in reverse sequence of the installation procedures. The modules shall be removed and placed on a cargo barge, which is fixed on the cargo barge by welding. The deck and module removal can also be carried by floatover method, which requires a specially designed vessel that can be partially submerged by ballasting and raised by deballasting. The cargo barge is then sent to the onshore disposal yard.

Complete Removal of Jacket and Hull

Complete removal of jacket can be conducted by separating the structure from its foundation and transporting it in whole or in pieces to onshore yard for further dismantling. Explosive method is usually to separate the structure from its foundation. Prior to the explosive detonation, certain surveys need to be carried out to ensure that the marine mammals are not near the platform. Usually, explosives are placed in the main piles and skirt piles at least 15 ft below the mudline. If the lifting capacity of the heavy-lift vehicle is insufficient, buoyant can be added to the served jacket to reduce

the weight. Large jacket can be towed to shallower water at first, then cut to sections, and sent to the onshore disposal yard.

After the complete removal of the platform, both the site and any location where the jacket is cut have to be cleared of debris and verified that they have been cleared. The site can be cleared by trawlers dragging nets across the bottom in a recorded grid pattern of specified dimensions or by the use of other authorized methods.

Partial Dismantling of Offshore Platforms

When offshore platforms require decommissioning, there are usually different ways, including complete removal, in situ decommissioning (leaving the infrastructure in place either completely intact or with the topsides removed and legs toppled), removal and relocation offshore platforms (for example, as a dive site or fishery), as well as partial removal (removing some parts of the infrastructure while leaving others in situ) (Ekins et al. 2005; Chandler et al. 2017). Partial dismantling usually happens in the decommissioning of offshore platforms. Platform jackets or hulls may be decommissioned using one of the following alternatives – complete removal, partial removal, or remote reefing. These alternatives vary only in the methods used

to remove the jacket or hull. For any given platform type, the platform removal preparation, well P&A, and pipeline decommissioning are the same for all removal alternatives.

The advantages and disadvantages for dealing with offshore platforms in different ways, i.e., leave in place, partial removal, and complete removal, are shown in Table 3.

Complete and Partial Dismantling, Table 3 Advantages and disadvantages for different methods of platform removal

Option	Advantages	Disadvantages
Leave in place	- Poses no harm to marine life - Immediate cost savings - Provides recreational fishing, diving habitat - Provides emergency safe havens - Maintains status - Structure remains visible - Requires no research and development - Requires no site clearance - Provides migratory animal habitat (surface) - Provides reef habitat (subsurface) - Suitable for structures at water depth more than 400 feet	- Maintains unnatural habitat - Maintenance costs escalate with age - Requires protective coating above water - Requires cathodic protection under water - Requires navigation-aid lights and horns - Remains susceptible to storm damage - Continues conflicts with other users - Potential liabilities - Unauthorized boarding - Collisions - Surface and subsurface navigation hazards - May require eventual removal with - Reduced structural integrity - Increased safety risk - Increased cost - Negatively affects construction/removal industry - No recycling of steel - Requires changes in regulations and laws - Not suitable for structures at water depth less than 400 feet
Partial removal	- Potentially reduces harm to marine life during removal and maintains some reef habitat - Potentially cost effective - Requires no maintenance - Requires no site clearance - May provide recreational fishing and diving habitat - Operators released from liability - Encourages innovative removal methods	- Does not return habitat to natural state - Eliminates habitat structure in upper range of water column - Must maintain buoys - Useful only in water depths allowing sufficient clearance - Potentially increases diver risk during removal - Decreases shrimping access - Liability attaches to regulatory agency - Court test inevitable - Creates navigational hazards (surface and subsurface) - Loss of resources - Eliminates surface habitat - No recycling of steel
Complete removal (at 15 feet below mudline)	- Meets shrimper requirements - Maintains clearance for trawlers - Requires no changes in regulations or laws - Poses no navigational hazards - Eliminates liability and site maintenance - Allows reuse and recycling	- Environmental impacts - Relocates or eliminates reef habitat - Fish kill from explosives - Expensive to operators - Explosives require an observer program - Restricts use of explosives - Discourages development of nonexplosive techniques - Requires transportation to shore or reef site - Requires site clearance - May require backfill - Hazardous to divers - Potential removal problems from soil skin friction at 15 feet below mudline

(continued)

Complete and Partial Dismantling, Table 3 (continued)

Option	Advantages	Disadvantages
Complete removal (at 5 feet below mudline or less)	Immediate cost savings Requires less jetting Minimizes problems from soil skin friction Encourages use of nonexplosive methods Less hazardous to divers Easier to clean for access by mechanical or abrasive tools Meets shrimpers requirements Nothing remains above mud line Reuse or recycling possible Poses no navigational hazards Requires no backfill Eliminates liability and site maintenance	Requires changes in regulations and laws Explosives may still be necessary in some cases although advanced techniques using smaller charges could be used Site clearance required Environmental impact Relocates or eliminates reef habitat Requires disposal

Closing Remarks

Different from partial dismantling of offshore platforms (such as offshore jacket platform), generally, the ships are completely dismantled in different ways after a life span of around 30 years. Specifically, the environmental management plan for ship dismantling, pre-demolition removing and preparations before ship dismantling operations, ship dismantling facility arrangement, and the dismantling procedures for different methods of ship recycling have been introduced in this paper. The end-of-life ships usually contain many kinds of hazardous materials which may result in health, safety, and environmental issues if they are not disposed properly. Therefore, an environmental management plan is needed; advanced techniques remain to be developed to enhance the prevention and protection of the environment, human health, and safety. The steps for the complete dismantling of offshore platforms have also been discussed. “Design for recycling” concepts are meaningful ways of improving the health, safety, and environmental conditions during ship dismantling.

Cross-References

- [Decommissioning of Subsea Facilities](#)
- [Recycling Regulations](#)
- [Ship Recycling](#)
- [Ship Recycling Facility Plan](#)

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Compliant Tower Platform

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Synonyms

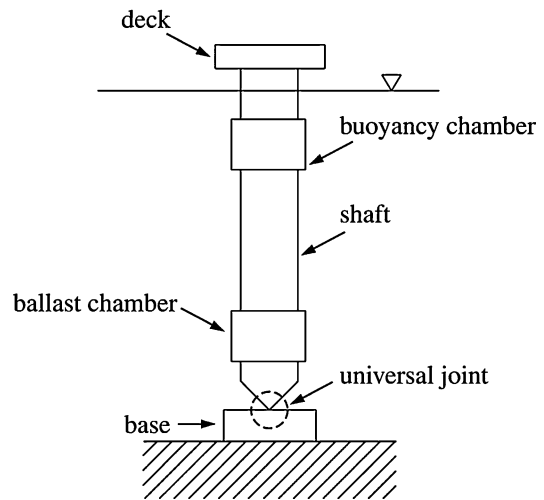
Single anchor leg mooring (SALM); Single point mooring (SPM)

Definition

A *compliant tower (CT)* is a fixed rig structure normally used for the offshore production of oil or gas. The rig consists of narrow, *flexible (compliant) towers* and a piled foundation supporting a conventional deck for drilling and production operations. *Compliant towers* are designed to sustain significant lateral deflections and forces and are typically used in water depths ranging from 1,500 to 3,000 ft. (450–900 m). At present the deepest is the *Chevron Petronius tower* in waters 623 m deep (https://en.wikipedia.org/wiki/Compliant_Tower).

Basic Characteristics of Compliant Tower Platform

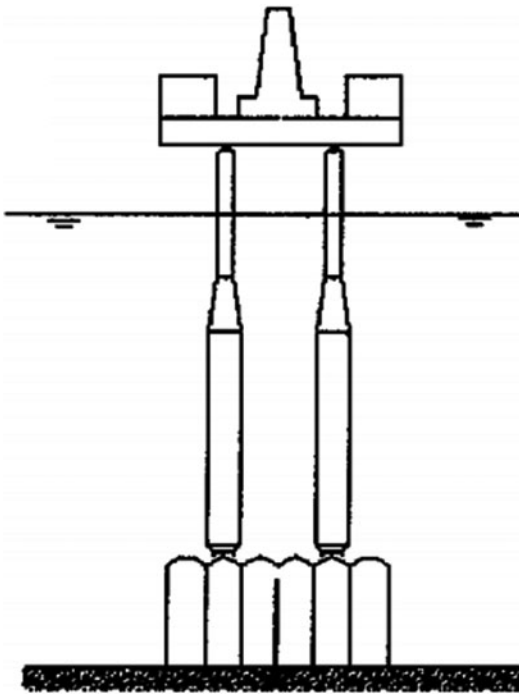
The compliant tower platform consists of the tower shaft, buoyancy chambers, ballast



Compliant Tower Platform, Fig. 1 A compliant tower platform diagram (Benaroya and Gabbai 2008)

chambers, the deck, and the base, as shown in Fig. 1. The shaft is connected to the seafloor through a universal joint. The base can be either piled or gravity type. The compliant tower platform is among the first elaborate compliant design to be used in the ocean service. “It serves as the first link between the sea-keeping analysis of the free body, traditionally the domain of the naval architect, and the environmental load analysis of the permanent structure, the traditional task of the ocean engineering” (Young 1983). The extension of the concept of single leg compliant tower platform has led to the development of the multi-hinged compliant towers (see Fig. 2). In the design of the multi-hinged compliant towers, there are several columns which are parallel to each other. The columns connect the deck and the foundations. The universal joints ensure that the columns are always parallel to each other, and the deck is always in horizontal position. This type of multi-hinged compliant tower can be used at large water depth.

The compliant tower platform is suitable for seawaters between 60 and 200 m. Also, shafts can be series connected, which could allow larger water depth. In 1977, the single anchor leg mooring (SALM) with multi-hinged articulated tower in the Thistle field in the North Sea was installed at the water depth



Compliant Tower Platform, Fig. 2 A multi-hinged compliant tower platform diagram

of 174 m. The compliant tower platform can be used for single point mooring (SPM), loading terminals, control towers, and early or full production facilities. In some marginal fields where reserve size does not warrant large facilities, it may also be used as a production platform.

History of Compliant Tower Platform

In the year 1963, due to the needs of industries, the first compliant tower platform was designed. In 1968, the first full-scale compliant tower platform to be used for experimental purposes was built. This platform worked at a depth of 110 m. This platform existed for 3 years at the site, during which researchers carried out many tests demonstrating that the compliant tower platform can be used in offshore engineering. In 1977, Burns and O'Amorim proposed the design and construction of two compliant tower platforms (Burns and O'Amorim 1977). The two

compliant tower platforms worked at a depth of 140 m. The design took into account the dynamic response and the strength of the base under the wind, wave, and current loads. The main structures of these two compliant tower platforms include base, universal joint, ballast chambers, buoyancy chambers, and the shaft. In the year 1990, the world's largest SPM compliant tower platform was built in 1989 in the Timor Sea in the northwestern of Australia. This compliant tower platform can withstand waves up to 9 m, wind up to 47 m/s, and current up to 2 m/s.

Main Characteristics of Compliant Tower Platforms

The tower shaft is the main structural component and is hinged on the seabed. The tower shaft can be a tubular cylinder or a steel lattice column. There is normally buoyancy chamber near the water surface. At the bottom of the shaft, there is always a ballast chamber. The oil production system is set up on the deck. The buoyancy chamber near the water surface provides buoyancy, and the buoyancy forces ensure the compliant tower platform is in an upright position. The larger the buoyancy, the smaller the tower shaft's motion caused by the wave force.

Besides, the damping effects of compliant tower are of great importance for the dynamic response of the whole platform. The influence of frictional damping, structural damping, and viscous damping on the vibration response amplitude is different. Frictional damping has the greatest influence on the amplitude attenuation of free vibration response without wave force, followed by structural damping, while viscous damping of fluid is the least (Hao 2008). In other words, the control of frictional damping can well control the response of the platform and enable the platform to quickly return to the equilibrium position after being stressed.

In addition, the difference between the compliant tower platform and the traditional steel structure platform is that there is no pile inserted into the seabed in the compliant tower platform, while

the pile of the traditional steel structure needs to insert into the seabed to resist loads of various forces at sea. Therefore, it is more convenient to construct and install a compliant tower platform and cause little hazard to the seabed environment. The compliant tower platform relies on its own huge moment of buoyancy to adapt itself to the waves without affecting drilling and production operations. In the case of not using, the ballast water in the base can be drained, and the base drifts up a little. So the tower platform can be easily transported by tugboats.

Furthermore, because the compliant tower platform connects to the seabed, the capability to resist the seismic load deserves great attentions. Some researchers (Han et al. 2006) analyzed the natural vibration characteristics and the seismic response characteristics of the platform by establishing the nonlinear governing equations of the compliant tower platform under earthquake excitation. The influence of different depths and the friction coefficient of the bottom universal joint on the platform response were also discussed. It is found that the seismic load acts on the platform can cause vibration of the compliant tower platform; when only the average water depth is changed, while keeping other parameters unchanged, the seismic response decreases with the increase of water depth; and when the friction coefficient of the bottom universal joint is increased, the amplitude of the seismic response decreases, and increasing the friction coefficient properly can effectively reduce the seismic response of the compliant tower platform.

The compliant tower platform has its own advantages and disadvantages. Compared with the traditional steel structure platforms, this type of platform can save a lot of steel. Because of the small size of this type of platform, small wind and wave force, simple structure, and low cost, the compliant tower platform is an ideal platform for deep water. The natural frequency of this type of platform is much lower than the wave frequencies. A salient feature of this type of platform is that the buoyancy chamber is near the sea surface, so the wave load is principally caused by inertia.

Although compliant tower platform has quite a lot of advantages, there are also some

disadvantages. The disadvantage of this platform is that due to practical and economic reasons, the buoyancy of the compliant tower platform is not enough, and the platform may swing too large in some cases. Sometimes the base may slide along the seabed. Besides, in severe sea state, the motion of the compliant tower platform can be too large to work normally.

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Computational Fluid Dynamics (CFD)

- [Aquaculture Structures: Numerical Methods](#)
 - [Hydrodynamics for Subsea Systems](#)
-

Computational Fluid Dynamics in AUV/ROV/HOV Hydrodynamics

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Definition

Hydrodynamic performance is one of the most important performances of AUV/ROV/HOV; the

economy, practicability, and safety of submersible can be affected directly by its advantages and disadvantages. Computational fluid dynamics (CFD) is an important method to research the hydrodynamic performance of submersible. It is a systematic analysis of physical phenomena including fluid flow through computer numerical computation and image display. The flow can be numerical simulated by the CFD under the control of the basic equation of the flow. The distribution of physical quantities at various positions in the flow field of extremely complex problems and the changes of these physical quantities with time can be obtained through the numerical simulation. The flow field of submersible can be obtained in CFD results, laying a good foundation for the further optimization of submersible designs.

Scientific Fundamentals

The Basic Concept of CFD

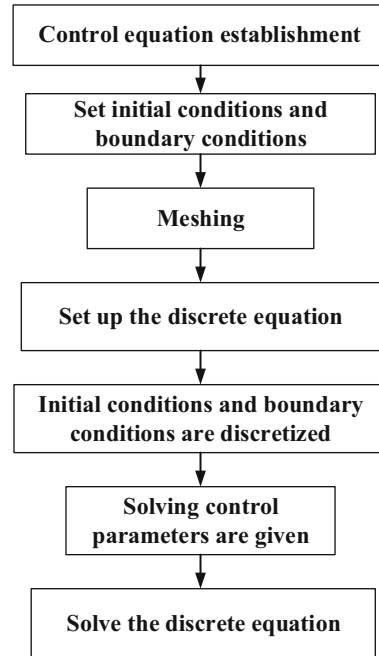
The fields of the continuous physical quantities in the spatial domain, such as velocity field and pressure field, are replaced by a series of sets of variable values at finite discrete points. Through a theoretical relation, the algebraic equations of the relation among the variables at these discrete points are established, and then, the approximate values of the variables are obtained by solving the algebraic equations (Renilson 2015).

The Basic Procedure of CFD

Actually, there is no standard CFD procedure. A general procedure can be expressed as below (Hu and Li 2015) (Fig. 1).

Characteristics of CFD Method

There are usually three methods to research the hydrodynamic performance of submersibles: theoretical formula calculation method, experimental design method, and CFD method. The theoretical formula calculation method can only be used for simple problems and qualitative analysis. Compared with the theoretical formula, the experimental method has the characteristics of strong authenticity and high reliability. However, due to



Computational Fluid Dynamics in AUV/ROV/HOV Hydrodynamics, Fig. 1 The flow chart

the constraints of site and environmental conditions, the experimental research is difficult to be widely used. Compared with the traditional theoretical and experimental methods, CFD method has the following characteristics (Wang 2013):

1. Advantages:

1. Strong adaptability to physical problems and wide range of application.
2. Not subject to conditions, with more flexibility.
3. The underwater flow field of the submersible can be visually displayed.

2. Disadvantages:

Compared with the other two methods, CFD method has the characteristics of unrestricted conditions and strong adaptability, but it also has a few limitations.

1. The CFD numerical solution is an approximate calculation method, which can only get the numerical solution of a finite number of discrete points, with certain calculation errors;

2. It involves a large number of numerical calculations and requires high computer configuration.

Key Applications

CFD Software Structure and Common Commercial Software

To solve different types of engineering problems, complex processes are integrated in CFD software, which basically include three steps: pre-processing, solution, and post-processing. The corresponding modules are preprocessor, solver, and postprocessor.

1. Preprocessor

The preprocessor is used to complete the work of preprocessing. In the preprocessing phase, the following tasks should be completed by engineers:

- Establish the geometric computational domain of the physical problem.
- The computational domain is meshed into nonoverlapping subregions.
- Select the control equation.
- Define the property parameters of the fluid.
- Set boundary conditions for geometric computational domain.
- Set the initial conditions for transient physical problems.

At present, preprocessing software which is currently used includes ANSYS ICEM CFD, Pointwise, Gridgen, Hypermesh, GridPro, etc. (Wang 2013).

2. The solver

The task of the solver is to read into the grid file (e.g., “.msh”), boundary conditions, control parameters, etc., by the preprocessor, and then, the results can be outputted and calculated according to the solution algorithm specified by the user. For the convenience of users, pre-processing functions are attached to some software solvers.

At present, CFD solution software which is commonly used includes ANSYS Fluent, ANSYS CFX, STAR CCM+, Phoenix, etc. (Wang 2013).

3. Postprocessor

The data obtained from the calculation is presented in an intuitive way through the post-processing, which is convenient for users to analyze and engineering application. The data is presented in some ways which includes:

- Data table: resistance, lift, moment, speed, and other physical quantities
- Curve diagram: reflect the changing rule of each physical quantity
- Nephogram: reflects the spatial distribution of physical quantities, such as pressure contour, velocity contour, vortex contour, etc.
- Vector diagram: reflect the vector spatial distribution, such as the velocity vector diagram
- Animation: reflect physical quantity changes with time in the form of animation

At present, the commonly used software of CFD postprocessor includes ANSYS CFD-POST, Tecplot, EnSight, ParaView, etc. (Wang 2013).

In addition to commercial software, users can also compile and modify the code to form their own computing software according to their requirement. Commonly used CFD open-source software (OSS) includes Open Foam, SU2, etc. (Wang 2013).

CFD Applications in Submersible Design

1. Prediction of submersible resistance and effective power

State-of-the-art CFD techniques can be used to estimate the resistance of an underwater vehicle, either deeply submerged or close to the surface. In the preliminary design stage of AUV/ROV/HOV, it is necessary to configure the system power unit. Therefore, resistance and effective power prediction are necessary. The advantage in CFD method is short calculation period and low cost. By simplifying the model of the designed submersible firstly and then simulating the flow field around the submersible with CFD method, the resistance of the submersible in different speeds can be obtained. Finally, the effective power of the designed submersible can be predicted through a certain processing of results. In order to

ensure the effectiveness of submersible numerical calculation, it is usually necessary to compare the calculated results with the experimental results.

2. Shape or molded lines optimization of AUV/ROV/HOV

The shape or molded lines optimization is more conducted for AUVs or HOVs rather than ROVs, since the power of ROVs is supplied from the surface ship through umbilical. However, AUVs or HOVs normally carry the battery themselves to navigate or perform certain operations under the sea. So for AUVs or HOVs, shape or molded lines optimization are usually conducted to achieve as small resistance as possible to better utilize energy. Since CFD can be used effectively to determine the effect of small changes in the hull form, it makes CFD a useful tool in the molded lines optimization.

3. Maneuverability analysis of submersible

During the design stage, it is necessary to research the ability and performance of the submersible when changing its motion direction and attitude through the control device. For the maneuverability analysis of submersible, it is necessary to establish six degrees of freedom equation in space to determine the external forces on the submersible (including gravity, buoyancy, thrust, and hydrodynamic force). Among them, the hydrodynamic force on the submersible is generally represented by the hydrodynamic coefficient. Therefore, the CFD method to calculate the hydrodynamic coefficient of submersible is the precondition to research the maneuverability.

Cross-References

- [AUV/ROV/HOV Propulsion system](#)
- [AUV/ROV/HOV Resistance](#)

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Computational Fracture Mechanics

- [Numerical Simulation Floe Ice–Sloping Structure Interactions](#)

Concept Design

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Synonyms

[Design of submersibles](#); [Design spiral](#); [Detailed design](#); [Empirical design](#); [Hydrodynamic design](#); [Multidisciplinary design optimization \(MDO\)](#); [Optimal design](#); [Preliminary design](#); [Reliability based design \(RBD\)](#); [Structural design](#); [Technical design](#)

Definition

Concept design is the first part of basic design, followed by preliminary design. It turns the mission requirements into the characteristics of the systems, developing two or more design alternatives from which a design is selected in accordance with specified criteria.

Scientific Fundamentals

Concept design is the first part of basic design, followed by preliminary design. In general,

according to E. E. Allmendinger (1990), it often involves the following stages:

1. Development of performance requirements, or system capabilities, from the mission requirements.
2. Determination of the principal characteristics of the systems required to achieve these capabilities, which enable new construction systems to be designed and existing systems to be selected and altered if necessary.
3. Estimation of capital and operating costs of the systems.
4. Identification of one or a few mission systems from among a series of alternatives which optimally meet performance requirements to the extent possible considering design constraints acting.

The system discussed here includes the total system which is also called the mission system and its supporting systems. The supporting systems provide the submersible with shore-based facilities, housing maintenance, repair and administrative activities, transportation, launching and retrieval services, navigation, and even with land-transport vehicles, air-transport vehicles, seafloor-based transponders, and submarines. The submersible itself and other systems should coordinate with each other in order to achieve the mission in the most effective manner. Therefore, it is important to consider the series of individual systems as a whole and take their interaction into account.

It is often convenient to describe a system as isolated from its environment by a boundary for the purpose of studying its internal behavior and interaction with its environment. A mission system is designed to serve one or more underwater missions. The boundary of the mission system encloses individual or systems, each enclosed by its own boundary, illuminating the fact that the mission is completed by a submersible supported by other systems.

Many mission systems are designed in the beginning with most of the potential systems already existing and some nonexistent. Existing systems require from no to extensive alterations to

convert them to systems, within equivalent amounts of design effort, construction, and costs involved. As for nonexistent systems, they must be designed and constructed “from scratch” and are expensive among all the systems.

Design Constraint

There are many design constraints imposed on the mission system and its individual systems. These constraints can be categorized into three kinds including mission external design constraints, mission internal design constraints, and design criteria.

Mission external design constraints (MEDC) come from outside and are external to the boundary of the mission system, which means that they are independent of all mission considerations. The sources of MEDCs may include environmental, technological, economical, legal, sociological, physiological, and psychological. Some of MEDCs cannot be changed by designers, for example, environmental factors like the speed of current and the height of wave cannot be determined by the engineers. Some of them, however, can be controlled by designers. For example, during the process of designing the hull, the material, one of the technological factors, can be selected by the designer. Once the material has been determined, the following stages will be constrained by its characteristic.

Some of sources of MEDC will change as time goes by. For instance, the technology or the policy of the law at the stage of construction may be different from which when being designing. Some of these changes will have a great impact on the procedure of design. Therefore, it will be better for a designer to take it into account.

Mission internal design constraints (MIDC) come from inside and are internal to the boundary of the mission system or any lower-level system. The sources of MIDC are the interaction between the systems that influence each other. Some constraints between two systems are weak, but some are strong.

Design criteria (DC) afford standards for judgments regarding design procedures and for selecting system components (Allmendinger

1990). The use of specific structural formulas and the requirements of critical system components based on state-of-the-art technology are examples. The examples of design criteria also include the basis for optimization, cost-effectiveness, or minimum weight. And these design criteria may be formulated by the owner, the designer, or the classification society.

Concept design turns the mission requirements into the characteristics of the systems. It includes feasibility studies and completion of the optimum designs. Feasibility studies are concerned with the development of those performance requirements and associated system characteristics, which have significant impact on the cost-effectiveness optimization process. Completion of the optimum design, however, involves developing other major performance requirements and characteristics which are essential in defining systems but do not have significant impact on the optimization process (Allmendinger 1990).

Feasibility studies produce a series of mission system alternatives, all of which meet the demand of mission requirements and are technically feasible. One or some of these alternatives will be chosen to be the input of preliminary design stage. The number of the alternatives depends on the complexity of the task and the implication of optimizing techniques. Each alternative is composed of a unique combination of systems, each of which has its unique capabilities, characteristics required to achieve these capabilities, and cost associated with providing these characteristics. Certain unique systems may appeal in more than one combination, but individual combinations are not duplicated in other alternatives. Combinations of systems that make up the mission system alternatives may vary in number, capability, and state, all of which are related to the functions required to fulfill the mission tasks, the time for accomplishing these tasks, and even the economic benefits.

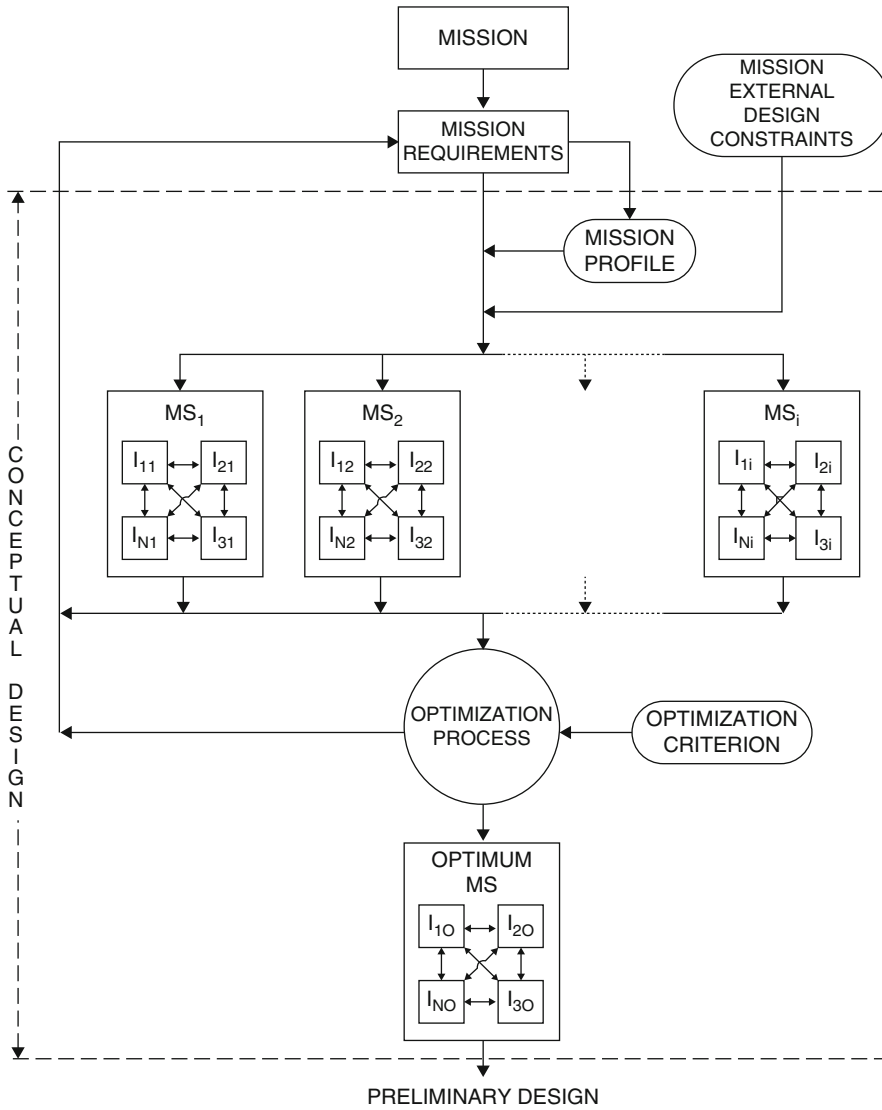
Figure 1 shows a “feedback line” from concept design to the mission requirements which indicate that certain changes in performance requirements may lead to changes in certain mission requirements, a procedure which involves

communication between owner and designer. According to Fig. 1, there are two inputs to the “feedback line”: one from the existing line of the mission system alternative and another from the optimization process. At first, it will be recalled that all of these alternatives must meet performance requirements and be technically feasible from the availability and operability points of view. As a result, in forming these alternatives, certain performance requirements cannot be met for technical feasibility reasons. On this occasion, they must be altered to make the alternatives at issue feasible. In the second instance, cycling the alternatives through the optimization process in searching for the optimum mission systems may reveal that changes in some performance requirements will issue in a better solution to the design problem from the cost-effectiveness view. In other cases, these changes may also require changing mission requirements to meet these requirements through a changed set of performance requirements. The other existing line from the optimization process leads to the optimum mission systems identified by this process in a manner suggested by the cost-effectiveness equation. By now, the concept design can be accomplished by developing the primary characteristics of the systems not considered in the feasibility studies. The identification of more than one optimum concept design means that the “optimization curve” is reasonably flat over a limited range of alternatives – that there is little to choose among them.

Approaches and Techniques

There are two approaches of concept design: one is the empirical approach which is modeled by the design spiral and another is the systematic parametric analysis approach. Figure 2 shows a typical design spiral.

As is shown in Fig. 2, the design spiral consists of spokes and loops. The spokes stand for design considerations including performance requirements, arrangement, geometry displacement, hull structure, propulsion plant, electrical plant, command and surveillance, auxiliary systems, outfit and furnishings, energy summary, weight displacement, cost-estimate, and so on.



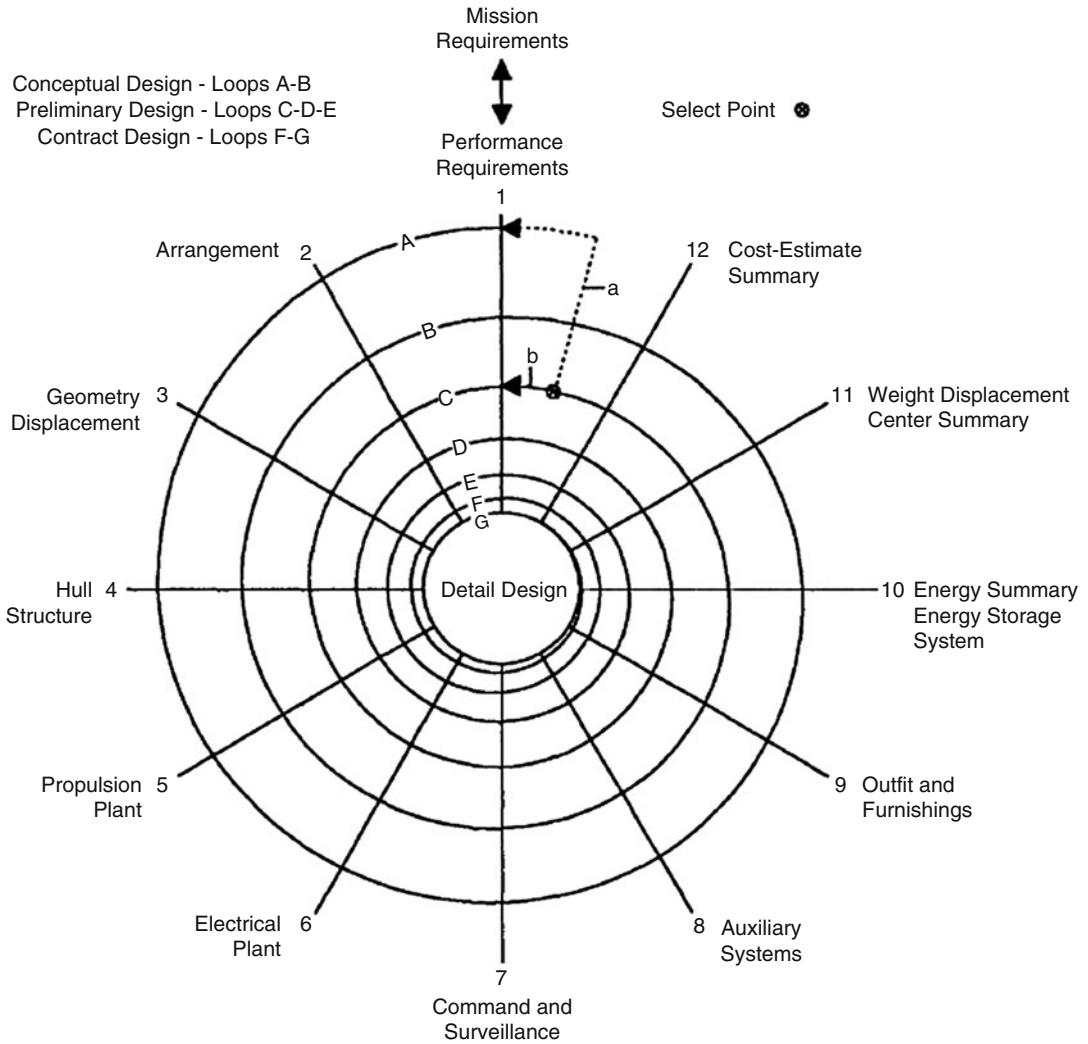
Concept Design, Fig. 1 Concept design of mission system (Allmendinger 1990)

The loops represent design iterations with the refinement of the design increasing as the loops spiral inward. The points identified alpha-numerically stand for certain stages. For example, 2B means the arrangement in the second loop.

Concept design feasibility studies are somewhat confined to the A and B loops of design spiral in the Fig. 2. The A loop is mainly considered with the approximate sizing of the submersible and utilize empirical formulations and similar design data extensively. Three major steps are

involved in the sizing process. Firstly, select the geometry and the major elements of the SWBS (Ship Work Breakdown Structure) systems. Secondly, obtain first estimate of sizing data for these systems. Thirdly, summarize these data to estimate the size parameters of total weight and displacement in the neutral buoyancy condition and the total cubic.

Concept design procedure from 1A to 12B based on Fig. 2 will be discussed in the following part.



Concept Design, Fig. 2 Design spiral (Allmendinger 1990)

A loop Step 1 – Performance Requirements

The basic inputs to the design process are those over-all mission requirements that are relevant to the design of the submersible. It includes task site and task work, the details of which are as following (Allmendinger 1990):

1. Task site

a. Location

1. Area of the world
2. Seafloor or water column site
3. Maximum operating depth

b. Type

1. Point or separated points – site at one point or more than one point separated by certain distances
 2. Line – site along lines of specified distances, as would be the case in a pipeline inspection mission
 3. Area – site over an area of specified dimensions, as would be the case in a seafloor area mapping mission
- ##### 2. Task work
- ###### a. Category

1. Inspection
2. Maintenance
3. Other
- b. Work objects
 1. General – a general description of the work object such as a buried pipe, an open-frame fixed platform, or a wet-well head
 2. Specified – as extensive information as possible needed to formulate adequate maneuvering and mission subsystems performance requirements facilitating proper vehicle positioning, vehicle orientation, and mating of tools and other equipment – a critical mission requirement in affecting the productive time of a submersible
- c. Task description
 1. Work to be performed – data to be acquired, nondestructive inspection to be performed, values to be replaced, and so forth
 2. Loads to be transported – from water surface to site or site to water surface or both. These loads are part of the overall payload which also includes weights of all items used exclusively in performing mission task work
- d. On-board personnel – required to accomplish certain work
 1. Number
 2. Type – one atmosphere or ambient pressure types
- e. Response time – time allotted for the submersible to reach to the work site, if critical in performing mission tasks

A loop Step 2 – Arrangement

Firstly, the submersible may be considered as consisting of the pressure hull, envelope, and all other forms of structures comprising it. In this stage, the designer should determine the type and number of the pressure hull which includes one-atmosphere type and ambient pressure type, the type of envelope which includes enclosed one and non-enclosed one. And the location of other items should also be considered at this stage. Inside the pressure hull, there are personnel

(in HOV) and all items of equipment to which they must have ready access. The items outside pressure hull include propulsion unit, high-pressure air bottles, main ballast tank, and so forth.

A loop Step 3 – Geometry

Geometries including shape and dimensions should be determined.

1. Pressure hull
 - a. Shape
 1. If the designed submersible works in shallow depths, then its shape is governed by hydrodynamic and arrangement considerations.
 2. If the designed submersible works in deep depths, then its shape is governed by structural considerations.
 - b. Dimensions

Dimensions are estimated from volume which pressure hull must enclose.
2. Envelope
 - a. Shape
 1. Enclosed envelope – envelope streamlined in direction of motion for relatively high forward or vertical speeds or both. If not streamlined, it should be a fair form to reduce the potential for entanglement and fouling
 2. Open-frame envelope – shape usually approximates a rectangular prism
 - b. Dimensions – the dimensions of length L, breadth B, and depth D, which can initially be estimated from the similar designs
 - c. Envelope volumes
 - d. Displacement of submersible

Steps 4–9 in A loop are categorized as SWBS (ship work breakdown structure) groups. Two of the sizing procedures should be followed in this portion of the A loop: (1) select major components of systems to be used with the particular design alternative under study and (2) obtain first estimates of sizing data for the systems selected.

A loop Step 4 – Hull structure

Fixed ballast and buoyancy are included. These systems can provide for neutral buoyancy,

zero trim, adequate stability, and a weight margin in design. The pressure hull is the main, pressure-resisting shell structure of the submersible. This hull is usually a ring-stiffened cylinder, closed with hemispherical heads, for shallow-depth submersible although other geometries may be used. An unstiffened sphere, or combination of spheres, is used for deep-depth submersibles.

A loop Step 5 – Propulsion plant

The propulsion plant is composed of all systems required to move the submersible in the ahead and astern directions under normal or emergency conditions. The propulsion power source which consists almost all of the batteries for untethered submersibles is included in this group.

A loop Step 6 – Electrical plant

The propulsion plant is composed of systems necessary to provide and distribute electric power to meet all of the submersible's needs, excluding propulsion. It is the nonpropulsion power sources for auxiliary, including electrical power generation system, power distribution system, lighting systems, and some special purpose systems.

A loop Step 7 – Command and Surveillance

The nature and the extent of systems composing command and surveillance system vary greatly depending on demands placed on them by mission performance requirements. It includes command and control system, navigation system, communication system, surveillance system, and special purpose system.

A loop Step 8 – Auxiliary Systems

- (a) Human systems (especially in HOV) – It is composed of crew and noncrew and personal effects brought on board.
- (b) Air, gas, and miscellaneous fluid systems, which are required to expel seawater from ballast tank and to provide hydraulic power.
- (c) Static control systems – Care must be taken that items associated with these systems which vary during the dive, such as seawater in the main ballast tank, are accounted for as loads under load to submerge.

- (d) Dynamic control systems – Only thrust-producing systems used exclusively for maneuvering are included in this subgroup.

A loop Step 9 – Outfit and furnishings

The weight and displacement are minute in these systems compared with others. And the space requirements of them should also be carefully considered. The size of ladders, lockers, seats, and hull insulation has great influence on the pressure hull volume and, hence, has a cascading effects on other design considerations.

A loop Step 10 – Energy Summary

The power-energy requirements for the SWBS systems should be summarized and the sizing data for the electrical and pneumatic energy storage systems should be estimated. At this point, the following should be considered: dive profile, bottom-work time, endurance, power users, power requirements, power totals and time, reserve energy, emergency energy storage, and energy storage capacity.

A loop Step 11 – Weight-Displacement-Center Summary

At this point, summarize weights and displacements of SWBS systems and their loads and achieve the submerged design condition criterion of neutral buoyancy, in which total weight is equal to the displacement.

A loop Step 12 – Cost Estimate Summary

Summary of cost estimates is deferred until Point 12B.

Above is the complete procedure of the A loop. The B loop procedures begin to develop the submersible and its SWBS systems in detail in order to refine sizing data, obtain cost estimates, and obtain assurance that design condition criteria of neutral buoyancy, zero time, and adequate stability can be acquired.

B loop Step 1 – Performance Requirements

Examine performance requirements according to sizing data created in the A loop. The requirements should remain the same unless these data indicate that certain requirements are excessive and may lead to unfeasible design. In this case,

the designer should consult with the owner to make some changes to the requirements.

B loop Step 2–3 – Arrangement and Geometry

At this stage, the first definitive view of the submersible is obtained, and the major components of the SWBS systems are located for the first time. These procedures are promoted by considering the two points together.

B loop Step 4–9 – The SWBS System Groups

These steps develop systems in sufficient detail to permit refinement of sizing data and close estimates of cost data to be made. More precise formulations and direct calculations are used in this portion of the B loop, which begin to analyze each system, breaking it down into its parts for study, increasing in detail as the design progress through the inner loops of the spiral.

B loop Step 10 – Energy Summary

It refines electrical power-energy profile and pneumatic storage system based on more precise B loop SWBS data.

B loop Step 11 – Weight-Displacement-Center Summary

Using the more refined sizing data, it obtain a more accurate estimate of fixed ballast or buoyancy, a check on the vertical distribution of weights and displacement necessary for adequate stability, and a check on the longitudinal distribution of weights and displacements required for zero trim.

B loop Step 12 – Cost Estimate Summary

It summarizes estimates of first and annual operating costs associated with the SWBS systems selected for a particular design alternative (Allmendinger 1990).

Systematic Parametric Analysis

Approach – Optimization Programs

These programs permit a very large number of concept design alternatives to be investigated, which increases the probability that the optimum concept design will be identified. Design procedures are formalized by establishing mathematical relationships between dependent design variables and independent design variables

or parameters. These relationships or parameter equations in the program permit rapid evaluation of design parameter changes on the dependent variables. And the explicitness of optimization program is useful in facilitating discussion between owners and designers regarding any aspect of the mission requirements or the design in question.

The above introduction mostly came from Allmendinger (1990). More detailed introduction can be found from Busby (1990) and Funnel (1999).

Key Applications

Concept design is an irreplaceable step of the design of submersible. Christ and Wernli (2014) dis reviewed the technology of ROV and then delve into the theoretical basis for ROV operations (including stability and drag computations) and also reviewed the standards surrounding the industry in general. Correa et al. (2015) introduced architecture for the conceptual design of remotely operated vehicles (ROV). Guo and Lin (2011) developed an underwater vehicle which can adjust its attitude freely by changing the direction of propulsive forces. Li et al. (2011) introduced the design of P-SURO AUV which is a test-bed for developing underwater technologies. The team of biologists and engineers considered the concept design of a bionic autonomous underwater vehicle (AUV) that uses the pectoral fins to achieve high maneuverability (Fish et al. 2003). ARV, a hybrid kind of submersible, can be considered as the combination of ROV and AUV. The design of *Nereus* (Bowen et al. 2008) is an important attempt of the design of ARV.

Cross-References

- [Design of Submersibles](#)
- [Design Spiral](#)
- [Detailed Design](#)
- [Empirical Design](#)
- [Hydrodynamic Design](#)
- [Multidisciplinary Design Optimization \(MDO\)](#)

- [Optimal Design](#)
- [Preliminary Design](#)
- [Reliability Based Design \(RBD\)](#)
- [Structural Design](#)
- [Technical Design](#)

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Conceptual Design

- [Structural Design](#)

Concrete Foundation Template

- [Shallow Foundations](#)

(Concrete Gravity-Based Structures) CGBS

- [Concrete Platform](#)

Concrete Platform

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Synonyms

[\(Concrete Gravity-Based Structures\) CGBS](#)

Definition

A *CGBS* is generally a very large and extremely heavy reinforced concrete structure which is placed on the seabed. It withstands the extreme environmental forces by virtue of its own weight and inherent strength.

CGBS platforms are among the largest and most impressive structures ever built and moved by man. To date the *Troll platform* is installed in the deepest water (303 m), and the *Hibernia platform* is the heaviest weighing 1.2 million tons on land. The designs of *CGBSs* vary considerably, and their weights range from 3000 to 1.2 million tons with corresponding topsides weighing between 650 and 52,000 tons.

A typical *CGBS* has a concrete base (often called a caisson) with one or more shafts to support the topside platform. When empty, voids within the base known as cells (along with the hollow shafts) provide buoyancy during the latter stages of construction, tow-out, and installation. When a *CGBS* is in operation, the cells are flooded with seawater or act as storage and in some cases separation facilities for crude oil (<https://www.iogp.org/bookstore/product/decommissioning-of-offshore-concrete-gravity-based-structures-cgbs-in-the-ospar-maritime-area/>).

Basic Characteristics of Concrete Platform

The concrete platform is a type of fixed platform which stands on the seabed by means of its huge weight. The bottom of a concrete gravity platform usually consists of many huge concrete caissons that support the deck structure with three or four hollow concrete legs. Caissons in the bottom are usually cylindrical and can be used as oil tanks or ballast tanks, as shown in Fig. 1. The weight of this type of platform is up to hundreds of thousands of tons. Table 1 shows the dimensions and weights of typical concrete gravity platforms. Thanks to its huge weight, the platform can directly sit on the seabed. However, due to the heavy weight of concrete platform, the condition of the seabed must be strong enough to bear the huge weight. So the concrete gravity platform is generally located on the seabed with dense sand.

Concrete platforms can be used for drilling, oil extraction, oil storage, and so on. The water depth that concrete platforms can work well is within 200 m and the best from 100 to 150 m.

Concrete platform has its own advantages and disadvantages.

The advantages of concrete platforms are:

- (a) Save steel and economical. The total cost of constructing a concrete gravity platform is lower than that of the steel jacket platform.
- (b) The installation workload at the sea site is small.

- (c) Anti-seawater corrosion, low maintenance costs, and long service life.

The disadvantages of concrete platforms are:

- 1. Require good seabed conditions.
- 2. The structural analysis and manufacturing process are complicated.
- 3. Reuse is difficult.

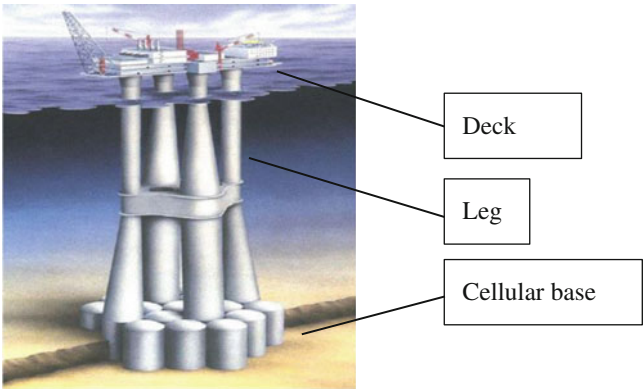
History of Concrete Platform

At the end of 1969, industrial oil was discovered in the Ekofisk field of the North Sea. However, due to the harsh sea conditions in the North Sea and the surging price of steel at that time, it is difficult to adopt the traditional steel jacket platform. Then, the Norwegians built the first concrete gravity platform. In the design, the Norwegians used their experience of building concrete breakwaters and offshore gravity light-houses, as well as the abundance of concrete in

Concrete Platform, Table 1 The dimension and weight of typical concrete gravity platforms (Sharp 1979)

Water depth	141 m
Production capacity	150,000 barrels/day oil
Storage capacity	550,000 barrels
Caisson base area	10,340 m ²
Caisson height	57 m
Weight of substructure in air	283,000 tons
Total height of platform	168 m

Concrete Platform, Fig. 1 A concrete gravity platform



Norway. This concrete gravity platform was designed in 1970 and started to be built in 1971 and was finished in June 1973. This concrete platform with 92 m diameter, about 100 m high, has an oil storage capacity of 158,900 m³. The successful construction of this platform provided valuable experience for the future use of concrete gravity platforms in the North Sea and other parts of the world.

Concrete platforms can be used for drilling, oil extraction, and oil storage and transportation. It has the advantages of saving steel, anti-seawater corrosion, low maintenance costs, and long service life. So 17 concrete platforms were built in the North Sea and off the coast of Brazil in just 5 years from 1973 to 1978. The maximum depth of these platforms' operating depths is 150 m and the minimum depth 15 m.

However, some shortcomings of the concrete platform appeared gradually, and in the early stages of development, some problems emerged. For example, the concrete gravity platform requires good seabed condition during working and deepwater depth during construction. These problems made it difficult to promote the concrete gravity platform to other waters in the world. At the same time, due to the improved techniques of steel jacket platforms, steel jacket platforms became popular again. This reduced the demand for concrete platforms.

After 1984, the advantages of concrete platforms were recognized by some oil companies, and orders started to increase gradually. The concrete platform at this time has the following characteristics:

- (a) From medium water depth to shallow and deepwater depth.
- (b) Safer and more reliable concrete gravity platform appeared.

Structural Components of Concrete Platform

Concrete platform contains a cellular base, columns, and deck structure. Figure 1 also shows the

structures of a concrete platform. The cellular base is the foundation of the entire building. In order to resist the huge storm force, the platform requires a large base structure. To prevent the base from sliding on the surface of the sea floor, skirt plates could be inserted into the seabed around the base. Large base structure can be used to store crude oil, which provides the concrete platform the advantages of oil storage.

The deck provides a workplace for production. The deck is fitted with various production and living facilities.

The column is connected between the base and the deck, to support the deck.

Construction and Installation Process of Concrete Platform

Gravity platform construction process is complicated, compared to those of other types of offshore platforms. Basically, there are two construction methods for concrete platform. One is that the building site can be located at the low-lying water area that is drained. The other is that construct the base on the land to a certain extent and then launched into the water.

The construction of concrete platform normally consists of the following steps:

- (a) In the dry dock, construct the base to a predetermined height and pour water into the dry dock.
- (b) Tug the base together with lifting equipment to the deepwater storm-sheltered area, and moor it firmly.
- (c) Continue building the upper part of the base and the legs until the concrete base and concrete legs are constructed completely.
- (d) Ballast the base, and the platform goes down. Then the prefabricated platform deck is transported by barges right up to the legs. Drain the base, and the legs float up a little until the legs support the deck at the right positions.
- (e) Connect the legs and the deck firmly together, forming a concrete gravity platform.

The Strength of Concrete Platform Main Structures

The strength of a concrete platform is highly important, and it is a basic problem needed to be considered when engineers design a concrete platform. However, sometimes it was still neglected and an accidental example is provided here.

On August 23, 1991, the loss of the Sleipner A concrete gravity platform caused huge economic losses (Collins et al. 1997). The concrete base structure of the Sleipner A platform was lowered into the Gandsfjord for deck-mating. When deck-mating, a Condeep structure was about 20 m deeper than it is during operation. The Condeep structure experienced the critical hydrostatic pressure. This high pressure caused the cell wall to break, allowing water to rush into the drill shaft. Later, the structure sank. As the structure went deeper into the fjord, the buoyancy cells imploded. Finally, this structure of \$180 million dollars was totally lost. Only the pile of rubble at the bottom of the fjord remained.

This accident gave a big lesson on the structural design for concrete platform. The design of concrete gravity platforms should consider the buckling or crushing of the cellular base bulkhead. When the base is also used as a storage tank, temperature stress due to internal and external temperature differences should be considered. In addition, when the concrete gravity platform works in the area with high seismicity, the seismic performance of the platform has to be considered. The water-structure interaction and soil-structure interaction have much importance in the analysis of offshore concrete gravity platform. Besides, the inelastic behavior of the hollow concrete members deserves attention (Whittaker 1987).

Furthermore, because concrete platforms mostly work in the North Sea, where ice may be expected in the winter, the ice-concrete interactive effect has also to be taken into account if the concrete gravity platform works in the ice zone. Level ice on offshore concrete platform is the main concern for general concrete abrasion, and

the other type of ice, for example, iceberg and freshwater ice, should not be forgotten (Tijssen 2015). Concrete abrasion mainly occurs below water level, because level ice adheres to the concrete above water level.

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Conductivity, Temperature, Depth (CTD)

- [Underwater Information Sensing Technology](#)

Conductor

- [Cable](#)

Confinement

- [Ship-Iceberg Interactions](#)

Connectors of Mobile Offshore Base (MOB)

- [Connectors of VLFS](#)

Connectors of VLFS

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Synonyms

Connectors of mobile offshore base (MOB); Elastic connectors; Flexible connectors; Hinge connectors; Rigid connectors; Rigid Module and Flexible Connector (RMFC); Semirigid connectors

Definition

A very large floating structure (VLFS) system is usually composed of a number of modules.

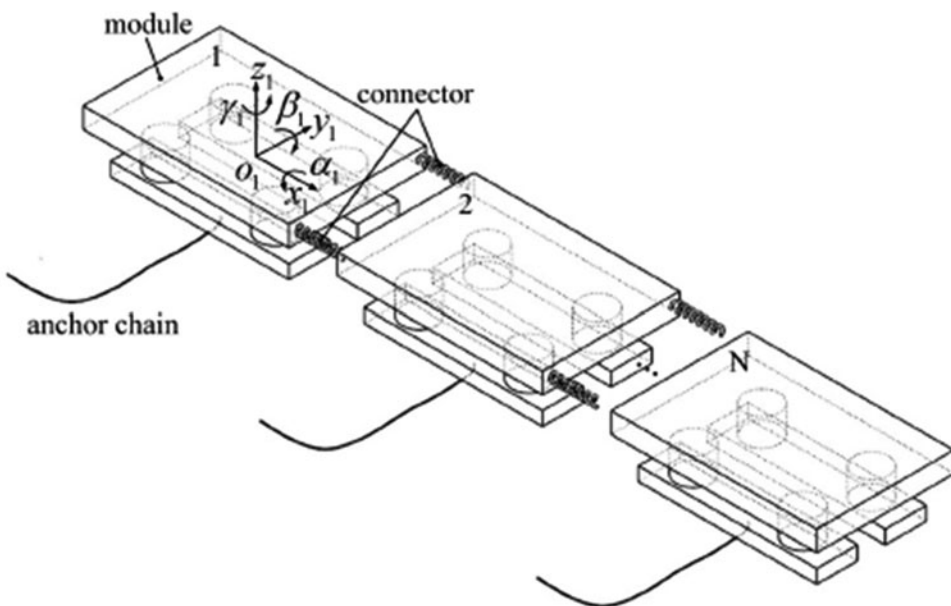
Connectors of VLFS are used to connect different modules of a VLFS system (see Fig. 1). For multimodular floating systems, the connector between modules becomes the weakest component in the whole system, which is subjected to large loads and is easily damaged.

Scientific Fundamentals

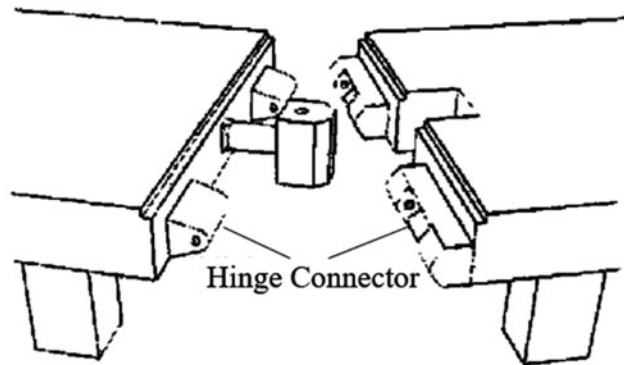
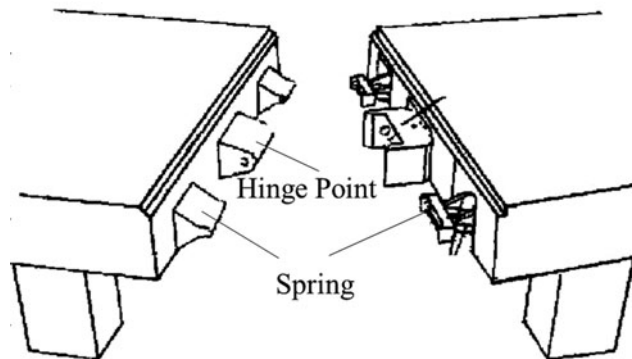
Connector Types

The connectors could be designed as different types for different purposes and ocean conditions, as well as beneficial for the fabrication, transportation, installation, and even potential expansion of the practical VLFS system. In general, there are the following types of connectors:

1. The rigid connectors (the fixed connectors): Adjacent modules are fixed together with no relative motion in all degrees of freedom.
2. The hinge connectors (the pin joints); see Fig. 2: Adjacent modules can move **freely in pitch mode** and move together (with no relative motion) in the other five degrees of freedom.



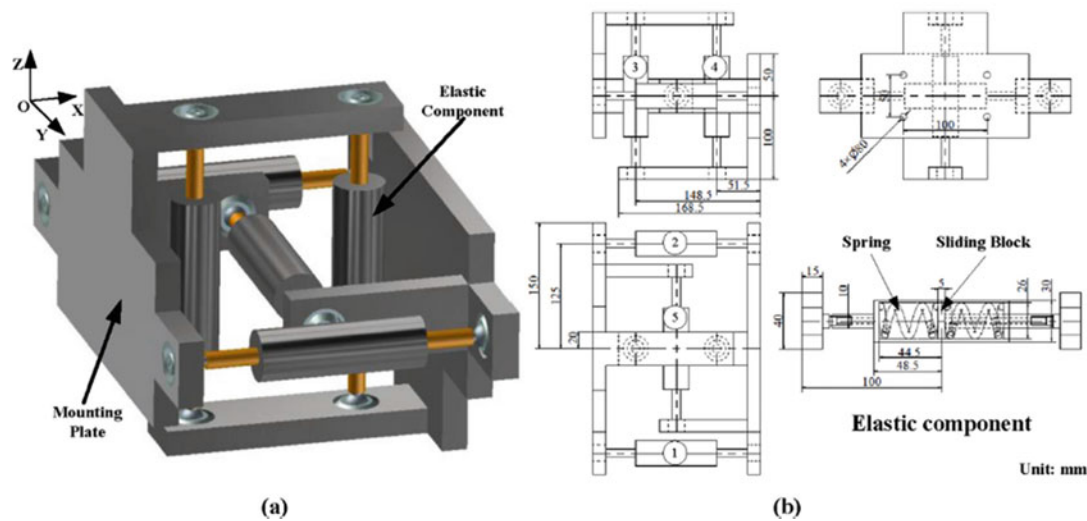
Connectors of VLFS, Fig. 1 Sketch of multimodular floating systems (Shi et al. 2018)

Connectors of VLFS,**Fig. 2** The hinge connector (Yu et al. 2003)**Connectors of VLFS,****Fig. 3** The semirigid connector (Yu et al. 2003)

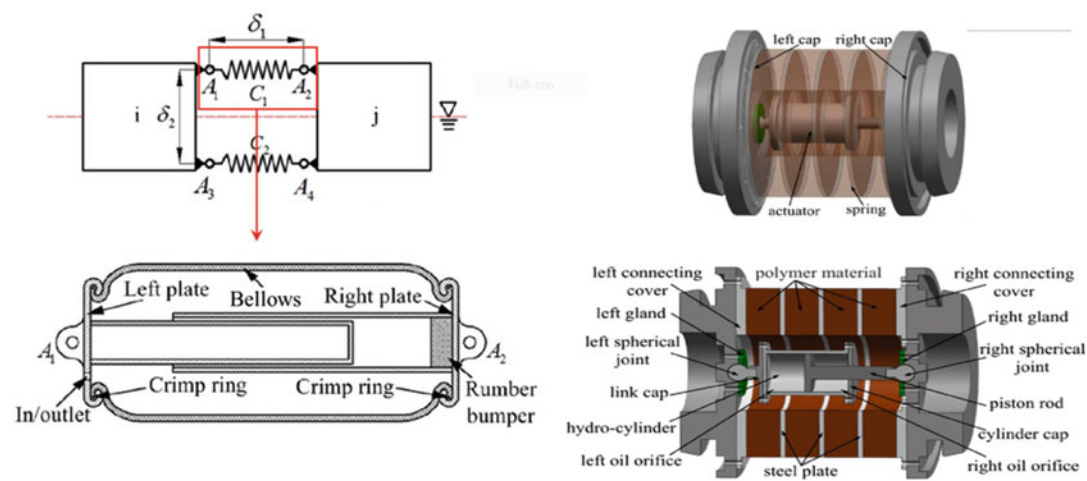
3. The semirigid connectors (the hinge connectors with an additional linear rotational spring), see Fig. 3: Compared with the hinge connectors, as for the semirigid connector, the relative pitch motion of adjacent modules will be **affected by the additional rotational spring**.
4. The hinge connectors with an additional rotational damper: Adjacent modules can move in **pitch mode affected by the damper** and move together (with no relative motion) in the other five degrees of freedom.
5. The flexible connectors (the chain-type connectors and the compliant connectors) shown in Fig. 4: The connectors could be elastic cables, which can only provide a relative constraint between the connection points of the adjacent modules. The adjacent modules have considerable relative linear displacement (surge, sway, heave) that is affected by the elastic properties of the connectors and can move **freely in three angular** displacement degrees (roll, pitch, yaw).
6. The elastic connectors (see Fig. 5): The connectors could have **one or several degrees of elastic deformation** under outside loads. The adjacent modules have relative smaller motions that are affected by the elastic properties of the connectors.
7. The fixed connectors **with connection gaps** (see Fig. 6): Adjacent modules can freely have smaller amounts of movement in several degrees because of the gaps. However, after a small amount of movement, the connectors work as rigid connectors, hinge connectors, elastic connectors, etc.

Simplification of Connectors in the Hydroelasticity Analysis of VLFS

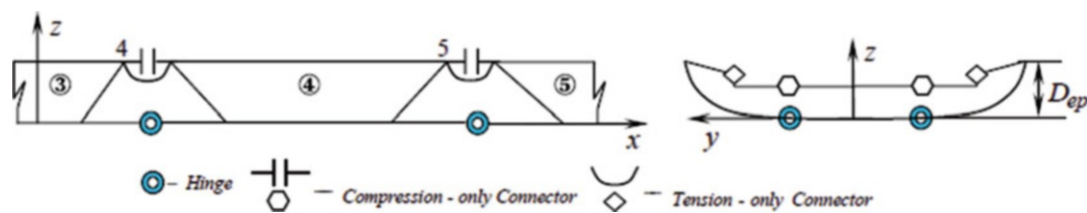
The Very Large Floating Structure (VLFS) is a unique concept of oceanic structures proposed for potential applications, such as floating airports, storage facilities for oil and natural gas, military and emergency bases, floating farms, and so on (Wang and Tay 2011). Because of the large scale



Connectors of VLFS, Fig. 4 The flexible connector (Shi et al. 2018)



Connectors of VLFS, Fig. 5 The elastic connector (Xia 2018)



Connectors of VLFS, Fig. 6 The fixed connectors with connect gaps (Fu et al. 2005)

of the horizontal directions of a VLFS, they are usually composed of a number of standardized or different types of modules with different types of connectors.

For the flexible/elastic connectors, if we simplify each module as a rigid body, the flexibility of the whole modular VLFS system will mainly occur in the connectors (Riggs et al. 2000). However, for the rigid or the fixed connectors, we could not simplify each module as a rigid body because the elastic deformations of the whole structure will be significant, occurring not only in the connectors but also in all the modules. Many research results show that the stiffness of the connectors strongly influences the hydroelastic responses of the VLFS and the loads on the connectors (Khabakhpasheva and Korobkin 2002; Fu et al. 2007; Wang et al. 2009; Gao et al. 2011; Michailides et al. 2013; Ren et al. 2019).

A comparative study of rigid-module-flexible-connector (RMFC) and finite-element-analysis (FEA) models for the hydroelastic analysis of a mobile offshore base (MOB) VLFS has been performed by Riggs et al. (2000). The results indicate that the simplified RMFC model with proper corresponding parameters can predict as well as the refined FEA model. Therefore, the RMFC model may be a suitable selection for hydroelastic analysis of the modular VLFS system for its short time-consuming. A generalized network model based on the RMFC model has been proposed and developed for modeling the modular VLFS system with arbitrary structure topology (Xu et al. 2014; Zhang et al. 2015b, 2017b; Xia et al. 2016).

Effects of Different Types of Connectors

The types of VLFS are usually the pontoon-type and semisubmersible type. From a dynamic point of view, the VLFS is different from the conventional floating structures. For the large scale of the horizontal directions, as well as the existing of elastic or flexible connectors, the stiffness is relatively small and its behavior on the water surface is similar to that of an elastic thin plate. Hydroelasticity theories (Betts et al. 1977; Bishop and Price 1977; Jensen and Pedersen 1981; Wu

1984; Price and Wu 1985; Bishop et al. 1986; Wu et al. 1991a, b, 1997; Jensen and Dogliani 1996; Xia and Wang 1997; Chen et al. 2003a, b, 2004, 2006a, b, 2010, etc.) were employed to check the conceptual design of a kind of very large floating airport of a length from hundreds to thousands meters or larger (Wang et al. 1991; Du and Ertekin 1991; Ertekin et al. 1993) during the last century and in the early part of this century.

A research project of Mobile Offshore Bases (MOB) executed by the Office of Naval Research (ONR) of the USA and a Japanese national project by the Technological Research Association of Mega-Float (TRAM) are two landmark projects with great influence on the development of VLFS technology. For the purpose of deducing the loads in the structure, as well as for convenience of construction and transportation, the types of multimodular floating structures with connectors are preferred. Thus, the connector becomes a key component for the multimodular floating structures, and the connection load becomes a critical issue for the design of modular connection.

Different kinds of connectors have been proposed for modular connections. The simple hinge connector is the most common one first proposed for VLFS. This type of the connector completely releases the pitch motion among modules so that the bending stress in the structure is greatly reduced but translational loads of the hinge connector are still large.

Compared with hinge connector, the compliant connector or the so-called flexible connector can significantly reduce the connector loads with some tolerances on structural flexibility. In this regard, the compliant connector, enhanced compliant connector, and modified compliant connector were sequentially proposed (Haney 1999). Riggs et al. (1999) studied 13 different connector stiffness cases, in which the longitudinal and vertical stiffness varied and the transversal stiffness was fixed at a large value. The results indicated that different stiffness cases might have different impacts on the system dynamic responses and connector loads. Xia et al. (2000) investigated the hydroelastic behavior of two-dimensional articulated plates connected by idealized connectors that were treated as a series of flexural

rotational springs and vertical linear springs. It indicates that the hydroelastic response strongly depended on the stiffness of the springs and the incoming wave frequency.

A prototype design of a flexible connector is associated with the stiffness configuration and the strength analysis. Since the stiffness of flexible connectors has a great impact on the connector loads and the responses of multimodular floating systems, the stiffness configuration is therefore a precondition of strength analysis. Some efforts have been made to understand the effect of the stiffness on the responses of multimodular floating systems. A multiplate floating system connected with torsion springs was studied, indicating that the torsion spring stiffness greatly influences the vibration amplitude of the main structures (Khabakhpasheva and Korobkin 2002). The effect of the stiffness of semirigid connectors on the hydroelastic response of VLFS was studied by Fu et al. (2007). Their results show that the stiffness of the connectors and the modules is very important in determining the hydroelastic response of the structure as well as the characteristics of the bending moment distribution.

Riyansyah et al. (2010) and Gao et al. (2011) investigated the effect of the location and rotational stiffness of the connector for reducing the hydroelastic response and stress resultants of the VLFS. They found that appropriate position and stiffness of connectors could improve the response of the floating system.

A modular multipurpose floating structure (MMFS) conceptual system utilizing a number of standardized modules is proposed by Ren et al. (2019). Different typical outermost connector designs have been proposed and investigated for the MMFS system. Different connector types of the outermost module can strongly influence the hydrodynamic responses of the MMFS system. The more rigid outermost connector tends to induce much larger responses of the shear force and the bending moment of the inner connectors. Hinge type outermost connectors can significantly reduce the responses of both the shear force and the bending moment of the most loaded connectors. By adding a proper linear pitch damper to the

outermost hinge connector, it can effectively reduce the motion of the outermost module, and produce considerable wave energy. The results indicate that the longer outermost module with the hinge connection may be an effective way to reduce both the loads on connectors and the motion of the outermost module.

Designation of Different Types of Connectors

Rognaas et al. (2001) put forward a flexible connector design with tolerance in the pitch and yaw motions, where elastomeric bearings, hydraulic jacks, and fenders were utilized to provide elastic supports, and steel cables undertook axial tensile forces. Except for utilizing rubber cushions or cables as compliant components, springs are generally applied to provide elastic connection. Xu et al. (2014) proposed a kind of flexible connector with trapezoidal rubber cushions and cables, where the rubber cushion is mainly used to constrain the longitudinal and transversal motions between adjacent modules when the compression occurs, and the cable is used to constrain the longitudinal motion between adjacent modules when the tension occurs. Four orthogonal linkages equipped with springs were used to restrain the linear displacements where angular rotations were allowed (Wu et al. 2016).

Xia et al. (2016) designed a special elastic connector with air-springs, which could alter the stiffness of connector to reduce the oscillation of floating structures by changing the air pressure of the air-springs. Although different designs exhibit their own distinguishing features, what is common is that these flexible connectors provide elastic constraints so that the load level in structure can be ameliorated. Zhang et al. (2017b) proposed three types of flexible connector configurations to represent anisotropic stiffness models and investigated the effect of the different stiffness on the global stability of the multimodular floating system with amplitude death mechanism (Zhang et al. 2015a) where a preferred parameter domain for the stiffness and wave period was suggested.

A variety of connections (Tan et al. 2018; Zhang et al. 2017a) utilized in ocean production systems have been widely investigated to provide guidance for structural optimization. Zhao et al. (2019) studied a general strategy for resolving the stiffness configuration for flexible connectors in various wave conditions. In numerical studies of an eight-module floating system, the optimal stiffness configuration is found in a full scenario of weight combinations. It indicates that the combination of small longitudinal stiffness, large transverse and large vertical stiffness is favorable for most wave conditions.

Cross-References

- ▶ [Floating Bridge](#)
- ▶ [Hydroelasticity Theory](#)
- ▶ [Mega-Float](#)
- ▶ [Station-Keeping System for VLFS](#)
- ▶ [Structural Analysis and Design of VLFS](#)
- ▶ [Very Large Floating Structures \(VLFS\): Overview](#)

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Conservation Biology

► Biodiversity Conservation

Conservation of Biological Diversity

► Biodiversity Conservation

Construction Design

► Structural Design

Constructive Solid Geometry (CSG)

► Obstacle Avoidance Technology for Underwater Vehicle

Continuous Plankton Recorder (CPR)

► Jellyfish Bloom

Contract Design

- ▶ [Design Spiral](#)
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Control Approach

- ▶ [AUV/ROV/HOV Control Systems](#)

Control Device

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Control Module

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Control Unit

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Controller

- ▶ [AUV/ROV/HOV Control Systems](#)

Conversion for Reuse

- ▶ [Modification for Reuse](#)

Cooperative Coevolutionary Artificial Bee Colony Algorithm and Tabu Search (CCTS-ABCA)

- ▶ [Obstacle Avoidance Technology for Underwater Vehicle](#)

Copper Alloy Net

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Coupled Eulerian-Lagrangian (CEL)

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CPT – Cone Penetration test

- ▶ [Pile Capacity](#)

CPT(Cone Penetration Test)

- ▶ [Offshore Pile Driving](#)

Cross-Layer Design

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Crushing Strength of Sea Ice

- ▶ [Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications](#)

CUSS I (Continental, Union, Superior and Shell Oil Companies' Project Mohole)

► Semi-submersible Platform

Cutting Technology of Steel

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Definition

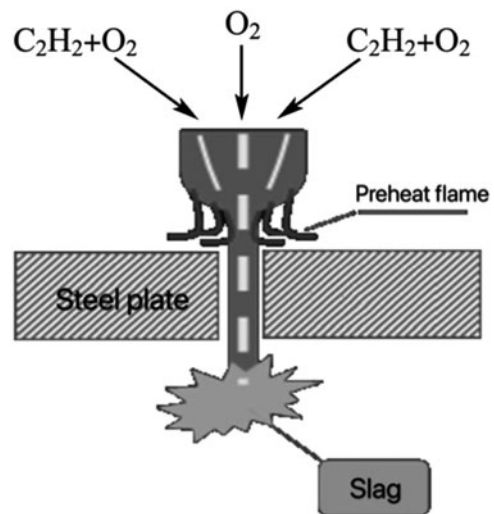
Ships and marine structures are made of thousands of welded and assembled members, which are made of ship steel. The technological process of making steel plates and profiles into hull members is called hull steel processing, and cutting technology of steel is also a part of hull steel processing, including mechanical cutting, flame cutting (gas cutting), plasma cutting, and laser cutting. With the development of hull construction technology, the processing technology of hull members tends to the direction of highly efficient processing equipment, numerical control, and operation assembly-line (Liu and Wang 2011).

Mechanical Cutting

Mechanical cutting refers to the process of applying to the material a mechanical cutting force exceeding the ultimate strength of the material, which causes the extrusion and the fracture separation of the material. Helical sword square shears and pressure shearing machines are the two main kinds of machines for cutting straight edge components in hull processing workshops, while the disc shearing machine is mainly used for mechanical cutting of curve edge components (Enomoto et al. 2012).

Flame Cutting (Gas Cutting)

Flame cutting usually refers to the oxy-acetylene cutting or oxy-propane cutting, the essence of which is the chemical process of metal burning in oxygen (Huang 2013). Generally, it can be divided into three stages: preheating, combustion, and slag removal. In gas cutting, first of all, the metal is heated by the adjusted preheating flame, so that the temperature of the starting point of the cutting seam gradually rises until it reaches the burning point of the metal. Next, a high-pressure flow of pure oxygen is released, causing the metal to burn (vigorous oxidation) and rapidly blowing off the slag (metal oxide) generated by the combustion. By continuing the above process, a smooth slit could be formed in the cut metal to separate the metal, as shown in the Fig. 1. In the hull processing workshop, manual gas cutting torch, semi-automatic gas cutting machine, portal automatic cutting machine (Fig. 2), and other equipment are often used for cutting the edge of components. The portal automatic cutting machine is equipped with many cutting torches, which are mainly used for the straight-line cutting of steel plates, and those cutting torches can be operated simultaneously with high cutting efficiency.



Cutting Technology of Steel, Fig. 1 Gas cutting

Cutting Technology of Steel, Fig. 2 Portal automatic cutting machine



Plasma Cutting

Plasma cutting is a processing method using high temperature plasma arc heat to partly melt (and evaporate) the metal at the incision of the work-piece, and forming the incision through the use of the momentum of high-speed plasma to exclude molten metal. It produces a temperature five to six times that of the average arc, and a flow rate of up to above 1500 m/s. Therefore, this method can be applied to cutting copper, aluminum, stainless steel, and other high temperature refractory metals which cannot be cut by gas flame. It should be noted that the method is especially suitable for cutting sheet, with a cutting speed three to six times that of flame cutting. Because of its high cutting speed and excellent cutting quality, plasma cutting has a great potential to replace gas flame cutting for cutting steel plate in shipbuilding (Xu 2005).

Semi-dry plasma cutting, dry plasma cutting, and underwater plasma cutting are three main ways of plasma cutting. Semi-dry plasma cutting (also known as water-cooled cutting) uses the force of the high-speed ionized airflow ejected from the torches to separate the molten metal to form a cut. Dry plasma cutting (also known as air-cooled cutting) uses the air compressor to inject air to the seam of the plate to achieve the cooling

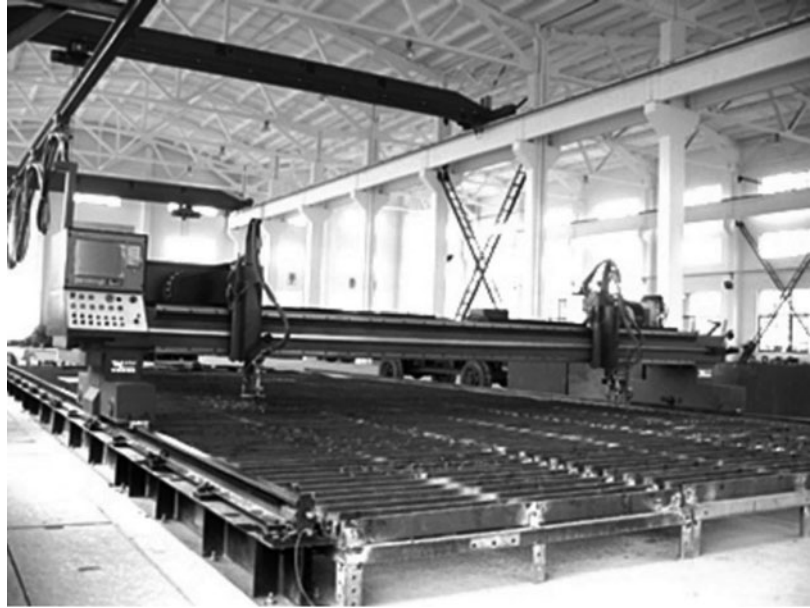
effect, which has a lower cost, but a greater pollution. Underwater plasma cutting uses the plasma arc to perform cutting in the water curtain or water, and uses the water to separate the plasma arc column from the air. Most of the metal dust generated by underwater plasma cutting is deposited in the water, and the frequency and intensity of noise are also greatly reduced, so the pollution of such cutting to the environment is limited (Li and Wei 2006).

Plasma cutting is not only faster, but has higher production efficiency and better incision quality. To be specific, the plasma arc incision is narrow and flat, with relatively small heat-affected area and deformation. Thus, numerical control plasma cutting machine is increasingly used in shipbuilding (Fig. 3) (An et al. 2020).

Laser Cutting

The application of laser cutting technology originated from the late 1970s. There are two forms of industrial laser machines, namely gas (such as CO₂) laser machines and solid (such as neodymium and yttrium aluminum garnet) laser machines, with different design and structure. The energy generated by laser can reach up to between 1×10^5 and 1×10^6 W/cm², which is fully adequate to melt metal. Therefore, laser can be

Cutting Technology of Steel, Fig. 3 Plasma cutting



used for metal cutting, and has many advantages – high cutting precision, good surface quality, high cutting speed, small thermal deformation, narrow notch width – compared with flame, plasma, and other cutting technology (Yilbas 2004).

Laser cutting can be generally divided into oxide assistant cutting and fusion cutting, of which the former is the most widely used. Oxide assistant cutting uses laser as the heat source for preheating, and oxygen and other active gases as the cutting gas. The gas ejected, on one hand, reacts with the metal being cut, oxidizes it and releases a large amount of oxidation heat; on the other hand, blows the molten oxides and fusion products out of the reaction zone to form a cut in the metal. The faster oxide assistant cutting speed is associated with a smaller thermal penetration and a better cutting quality. During laser fusion cutting, the metal is melted by laser heating; non-oxidizing gases (argon, helium, nitrogen, etc.) are ejected through the torch coaxial with the beam; the liquid metal is discharged by high pressure of the gas; thus an incision is formed.

Laser fusion cutting is mainly used for cutting some nonoxidizing materials or active metals. In its applications, oxygen is used as cutting gas for cutting non-alloy steel, while nitrogen is used for

cutting stainless steel, aluminum, and aluminum alloys (Shemshadian et al. 2020).

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D

Damage Stability

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Synonyms

[Damaged ship stability](#); [Stability of damaged ship](#)

Definition

Damage stability of a ship mainly refers to the ability of the ship to resist capsizing and restore the equilibrium position in calm water and waves, and it requires that the ship can still have sufficient buoyancy and stability after one or several damaged compartments. The water flow into the ship through the damaged compartment will change the ship's buoyancy, the ship's motion response, and dynamic stability in waves, and even cause the ship to capsize or sink. Damage stability is one of the most basic and important ship performances which is related to the safety of ship navigation at sea. Due to the complexity of the marine environment, the catastrophic events caused by the damaged instability often occur, which cause huge property losses in the light, a large number of casualties, and serious environmental pollution. Therefore, improving the ship stability under

damage conditions, especially the environmental adaptability and survivability of ships under severe sea conditions, has always been the main goal of the ship overall performance research.

Introduction

In the past, the ship damage stability problem had received little research attention, mainly because the numerical as well as experimental treatment of progressive flooding and capsize in a random seaway represents a very difficult undertaking. As a result, quantitatively specific standards of residual stability have been introduced for the first time in the 1960 SOLAS Convention which only address the metacentric height (0.05 m) (22nd ITTC).

However, a number of major disasters involving a large loss of life have forced development in international regulations based either on deterministic or probabilistic approaches to assessing damage stability but ignoring in the main the physics of the problem at hand (22nd ITTC).

Subdivision and Stability

Ship subdivision and layout are two of the most important issues affecting ship performance, functionality, and safety. The subdivision of a ship is considered sufficient if the attained subdivision index A is not less than the required subdivision index R according to the subdivision and

damaged stability provisions of SOLAS'90. For the calculation of required subdivision index R and attained subdivision index A, there are several key factors as follows:

Subdivision length (L_s) of the ship is the greatest projected molded length of that part of the ship at or below deck or decks limiting the vertical extent of flooding with the ship at the deepest subdivision draught.

Mid-length is the midpoint of the subdivision length of the ship.

Aft terminal is the aft limit of the subdivision length.

Forward terminal is the forward limit of the subdivision length.

Length (L) is the length as defined in the International Convention on Load Lines in force.

Freeboard deck is the deck as defined in the International Convention on Load Lines in force.

Forward perpendicular is the forward perpendicular as defined in the International Convention on Load Lines in force.

Deepest subdivision draught (ds) is the waterline which corresponds to the summer load line draught of the ship.

Light service draught (dl) is the service draught corresponding to the lightest anticipated loading and associated tankage, including, however, such ballast as may be necessary for stability and/or immersion.

Partial subdivision draught (dp) is the light service draught plus 60% of the difference between the light service draught and the deepest subdivision draught.

Permeability (m) of a space is the proportion of the immersed volume of that space which can be occupied by water.

Machinery spaces are spaces between the watertight boundaries of a space containing the main and auxiliary propulsion machinery, including boilers, generators, and electric motors primarily intended for propulsion. In the case of unusual arrangements, the administration may define the limits of the machinery spaces.

Watertight means that in any sea conditions water will not penetrate into the ship.

Watertight means having scantlings and arrangements capable of preventing the passage of water in any direction under the head of water likely to occur in intact and damaged conditions. In the damaged condition, the head of water is to be considered in the worst situation at equilibrium, including intermediate stages of flooding.

Deadweight is the difference in tonnes between the displacement of a ship in water of specific gravity of 1.025 at the draught corresponding to the assigned summer freeboard and the lightweight of the ship.

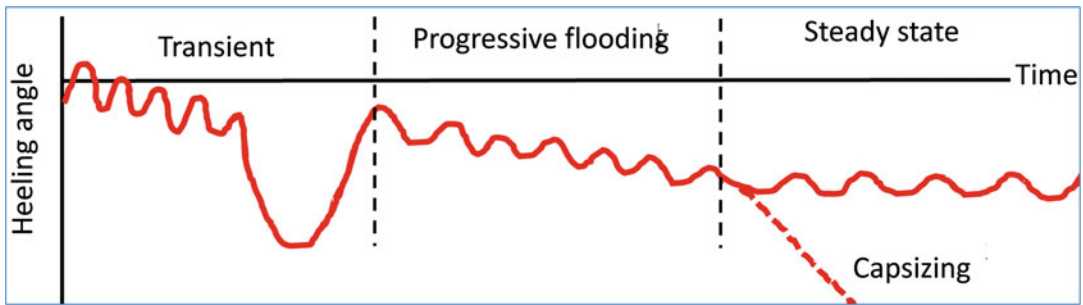
Lightweight is the displacement of a ship in tonnes without cargo, fuel, lubricating oil, ballast water, fresh water and feedwater in tanks, consumable stores, and passengers and crew and their effects.

Stability and Motion

When the side or bottom of one or more cabins are damaged due to collision, grounding, and other accidents, the external fluid will flow into the cabins through the breach holes, and then the fluid inside the breached cabins will flow into other adjacent cabins through the connection openings.

The performance of a damaged ship in waves is affected by both the external wave excitation and the hydrodynamics load generated by the liquid sloshing in the cabin, and the time domain motion of the damaged ship also affects the dynamic characteristics of the flooding. When the external waves enter the damaged cabin, the parameters of the ship (mass, buoyance, etc.) will change, causing the problem of liquid sloshing inside the damaged ruined cabin. The load generated by liquid sloshing will further affect the motion characteristics of the ship in turn. In this process, the motion of the damaged ship and the flooding water is coupled with each other. The research on the floodwater and the motion of the damaged ship in waves is extremely complex, involving many influencing factors.

In general, the flooding process that follows the creation of the damage opening can be divided into three main phases (IMO SLF46/INF.3 2003).



Damage Stability, Fig. 1 Main phases of the flooding process

Due to the large pressure difference, water will rush into the damaged compartment, and this phase can be called as transient flooding phase. After that, the water will flood to undamaged internal cabins through internal connection openings, and this phase can be called as progressive flooding phase. At the end of flooding, the ship will capsize/sink or reaches a final steady state. A schematic representation of these phases is given in Fig. 1.

The establishment of the prediction method has consistently been the focus of ITTC wave stability research. According to the physical phenomena involved in the damaged flow, ITTC divides it into three problems (ITTC 2005):

Damaged vessel dynamics: It generally refers to the ship with zero forward speed drifting on the free surface under the excitation of waves. At present, the research on the motion of a damaged ship with forward speed is also a hot topic. Nonlinear 6DOF seakeeping model that allows the vessel to drift as well as allows for changes with time in its mass, the center of mass, mean attitude, environmental excitation, and hydrodynamic reaction forces can be used for these problems (25th ITTC).

Water ingress/egress: This refers to the flooding phenomenon itself, namely the process of water inflow and outflow through the damaged openings and the progressive flooding through internal space. An adequately accurate water ingress/egress model that allows for multiple-compartment flooding in the presence of oscillatory flows and at time of shear flows in

extreme wave conditions is a prerequisite to undertaking any investigations on damage survivability. CFD method is the main method to study these problems at present (25th ITTC).

Floodwater/Vessel interaction: It refers to the behavior of the accumulated floodwater inside the ship's compartments and its interaction with the ship. A study of damage survivability involves two distinct but intrinsically interrelated and highly interacting processes, namely ship motion and flooding. Vessel motion influences considerably and directly the flooding process and conversely, flooding affects both the vessel motion and her attitude. It is essential, therefore, to take both phenomena into consideration when studying the evolution of either. However, the nonstationarity in the vessel motion introduced by the water accumulation coupled with the intermittence of the flooding process itself and the severe nonlinearities in the ensuing dynamic system demand great care in dealing with the various issues of this complex problem (25th ITTC).

Typical Accident

Damage stability is usually caused by collision, grounding, and contact. According to EU statistics of ship accidents from 1958–2009, it is found that there are 1016 accidents caused by collision, accounting for 66.54%.

In September 1994, the Baltic ferry MS Estonia sank, due to rapid loss of stability after loss of her hull integrity and ingress of seawater on the car deck through the bow ramp. More than

900 people were killed in one of Europe's worst ferry tragedies. The ship sank in about 30 min, just after midnight when most of the passengers were asleep. This vessel complied with damage stability requirements of SOLAS'74 (launched in 1980), while compliance with the later SOLAS'90 is uncertain (the ship was at the time of operation treated as an *existing ship* in terms of compliance with latest regulations) (Christianson and Engelberg 1999).

Key Numerical Method for Damages Ship Stability in Waves

The dynamic performance of damaged ships in waves is constantly changing, which leads to high nonlinearity of the dynamic system. Therefore, an effective method to reproduce the extreme nonlinearity of the ship motion and the process of the damaged flow is to solve it in the time domain. Up to now, many works have been devoted to the study of the damaged ship motions in waves.

The assessment of damaged ship motion is normally based on potential theory. The wave forces and hydrodynamic forces are usually calculated by potential theory, the same way as the intact ship; the internal water surface is assumed to be horizontal or a free-moving plane, and the inflow and outflow of water through the damaged opening are calculated by the modified empirical formulas (Vassalos et al. 1997; Lorkowski et al. 2014; Qian et al. 2000; Manderbacka et al. 2015; Lee 2015; Acanfora et al. 2019). Furthermore, roll damping coefficients due to floodwater are usually calculated by the empirical method. One degree of freedom (roll motion) to six degrees of freedom can be utilized to the simulation of damaged ship motion in waves.

With the development of technology, a fully viscous method is used for the capture of the dynamic characteristics of the flooding process (Shen and Vassalos 2009; Strasser 2010; Touzé et al. 2014; Gao and Vassalos 2015). However, we should also be aware of the simulation of the damaged ship motion coupled with the damaged flow by a fully viscous method which is unrealistic for the engineering application due to the high computational costs at present.

The coupling between the damaged ship motion in waves and the damaged floodwater is a very complex problem. The in-waves calculations need to consider not only the motions of the damaged ship but also the strong nonlinear floodwater dynamics as well as progressive flooding. Particularly, despite several studies that focused on the relationship between flooding water and the motion response of a damaged ship, the effects of flooding dynamics on the motion of a damaged ship are not yet clearly understood.

Considering that CFD method can simulate most of the flow characteristics and parameters, and the potential flow method has a strong advantage in solving seakeeping problems. Therefore, the coupling method of CFD and potential theory can be utilized to reduce the computational cost on one hand, and to simulate the flow problem more efficiently on the other hand. Cho et al. (2006) developed a numerical method that combines the three-dimensional frequency domain panel method and modified VOF method, which can take the internal flow and sloshing into account. Gao et al. (2013) combines strip theory and RANS equation for the simulation of damaged ship motion. Hashimoto et al. (2015) simulated the transient behavior of ships by coupling the three-dimensional MPS method with the strip method. Bu and Gu (2020), Bu et al. (2020, 2021) proposed a unified viscous and potential prediction method (UVPP) that combines CFD and potential theory to investigate the hydrodynamics of floodwater and its effects on the damaged ship motions. In this method, the three-dimensional time domain hybrid source method is employed to the calculation of damaged ship motion in waves, and

Damage Stability, Table 1 Main ship particulars

	Full scale	Model scale
L_{OA}	179 m	4475 mm
L_{BP}	170 m	4250 mm
B	27.8 m	695 mm
T	6.25 m	156.3 mm
$D_{CARDECK}$	9 m	225 mm
$Disp.$	17300ton	270.3 kg
KG	12.89 m	322 mm
GM	2.63 m	65.8 mm

the fully viscous method is employed to the simulation of the flooding process. They have proved the accuracy and efficiency of this method both in calm water in waves.

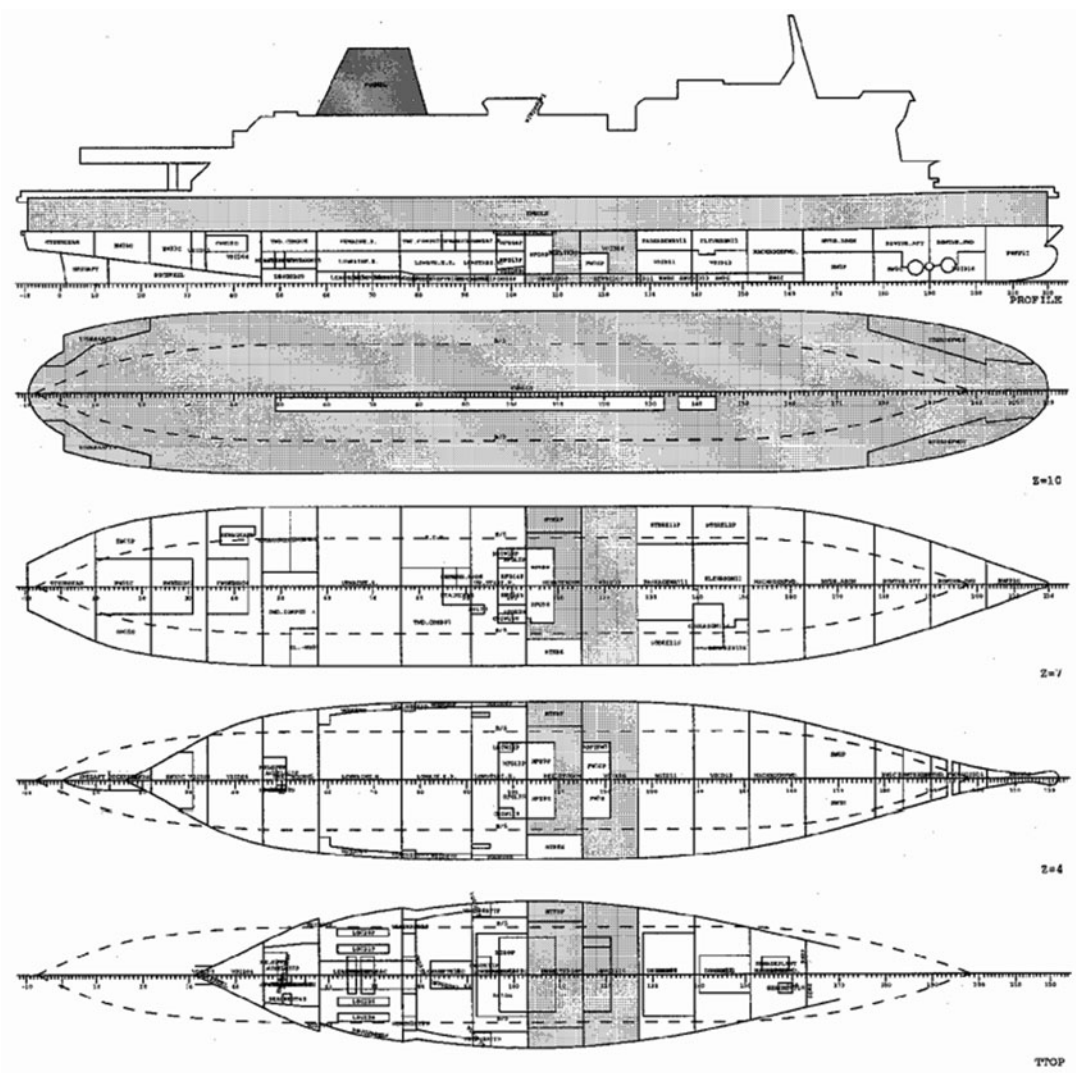
Influencing Factors on Numerical Simulation Results

The damaged Ro-Ro passenger vessel is adopted for the investigation of influencing factors on numerical simulation results (ITTC 2002), which was once used for the benchmark study of

damaged ship stability. The model tests were conducted in the Denny tank according to the model test method of Imo SOLAS’ 95 Resolution 14 (ITTC 2002). The main ship particulars are listed in Table 1, and the body lines are shown in Fig. 2.

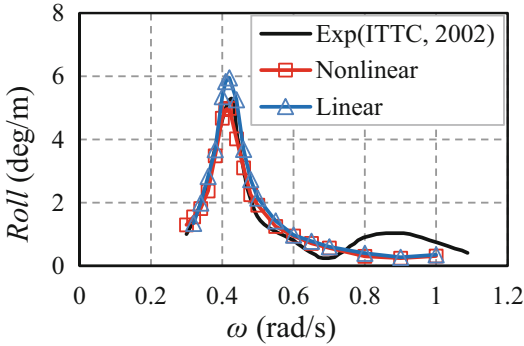
Effects of Calculating Method for Froude-Krylov and Hydrostatic Forces

Four degrees of freedom (sway-heave-roll-pitch) is adopted to calculate the motion response, and the wave height of 1.2 m is selected. Two methods were used for the calculation. In one method, FK

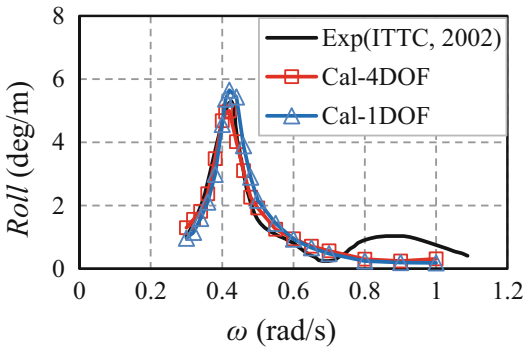


Damage Stability, Fig. 2 Ship body lines and midship damaged case (ITTC 1999)

and hydrostatic forces are integrated along the instantaneous wet surface of the hull, while FK and hydrostatic forces are integrated along the average wet surface of the hull in the second method. The comparison results are presented in



Damage Stability, Fig. 3 Effects of calculation method for FK and hydrostatic forces on roll motion



Damage Stability, Fig. 4 Effects of degree of freedom on roll motion

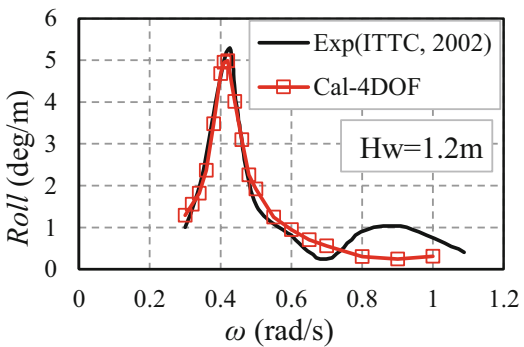


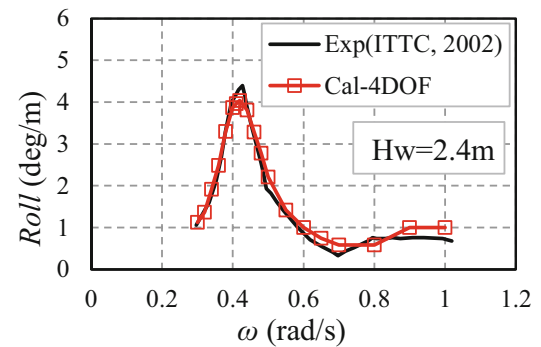
Fig. 3. It can be seen that the calculation results using the average wet surface integration are larger than the model test results, and the calculation results considering the instantaneous wet surface are in good agreement with the model test. Thus, it can be inferred that the instantaneous wet surface needs to be considered in the simulation of damaged ship motion.

Effects of the Degree of Freedom

Differences between the 1DOF roll motion mathematical model and the 4DOF mathematical model are compared, as shown in Fig. 4. The current calculation results show that there is little difference between the calculation results of the 1DOF mathematical model and the 4DOF mathematical model, which are in good agreement with the model test results.

The Effects of Wave Height

The 4DOF mathematical model and the method based on the instantaneous wet surface are used to calculate the roll motion at different frequencies and compared with the model test results. The comparison results are shown in Fig. 5. It can be seen that the calculation results are in good agreement with the model test. The difference of unit roll amplitude calculated by wave height 1.2 m and wave height 2.4 m explains the nonlinearity of the problem. Therefore, the linear method is no longer suitable for calculating roll motion. It can also be seen from the calculation results that at nearly twice the roll resonance frequency, the roll response of the damaged ship measured in the



Damage Stability, Fig. 5 Effects of wave height on roll motion

model test appears the second natural frequency, which is analyzed to be caused by the sloshing of the tank.

Cross-References

- [Fishing Ships](#)
- [Intact, Damage, and Dynamic Stability of Floating Structures](#)

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Damaged Ship Stability

- [Damage Stability](#)

DART

- [Deep-Ocean Assessment and Reporting of Tsunamis \(DART\) BUOY](#)

DART: Deep-Ocean Assessment and Reporting of Tsunamis

- [Tsunami Warning Buoy](#)

Data Analysis

► Big Data-Based Decision Support Systems

DCOR – Dynamic Characteristics of the Riser

► Buffer Station

Dead Reckoning (DR)

► Doppler Velocity Log for Navigation System in Underwater Vehicle

Dead Ship Condition

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Synonyms

Resonance roll; Stability in beam sea and wind

Definition

Dead ship condition is one mode of stability failure. This scenario assumes that a ship has lost its power and has turned into beam seas, where it is rolling under the action of waves as well as heeling and drifting under the action of wind. (SDC 3/WP.5/Annex 6 [2016](#)).

Figure 1 shows the process by which dead ship stability failure develop. Firstly, the ship moves with drift-related heel under wind aerodynamic force and hydrodynamic reaction caused by transverse motion. Next, the ship encounters a sudden and long gust of wind. When the ship rolls at the

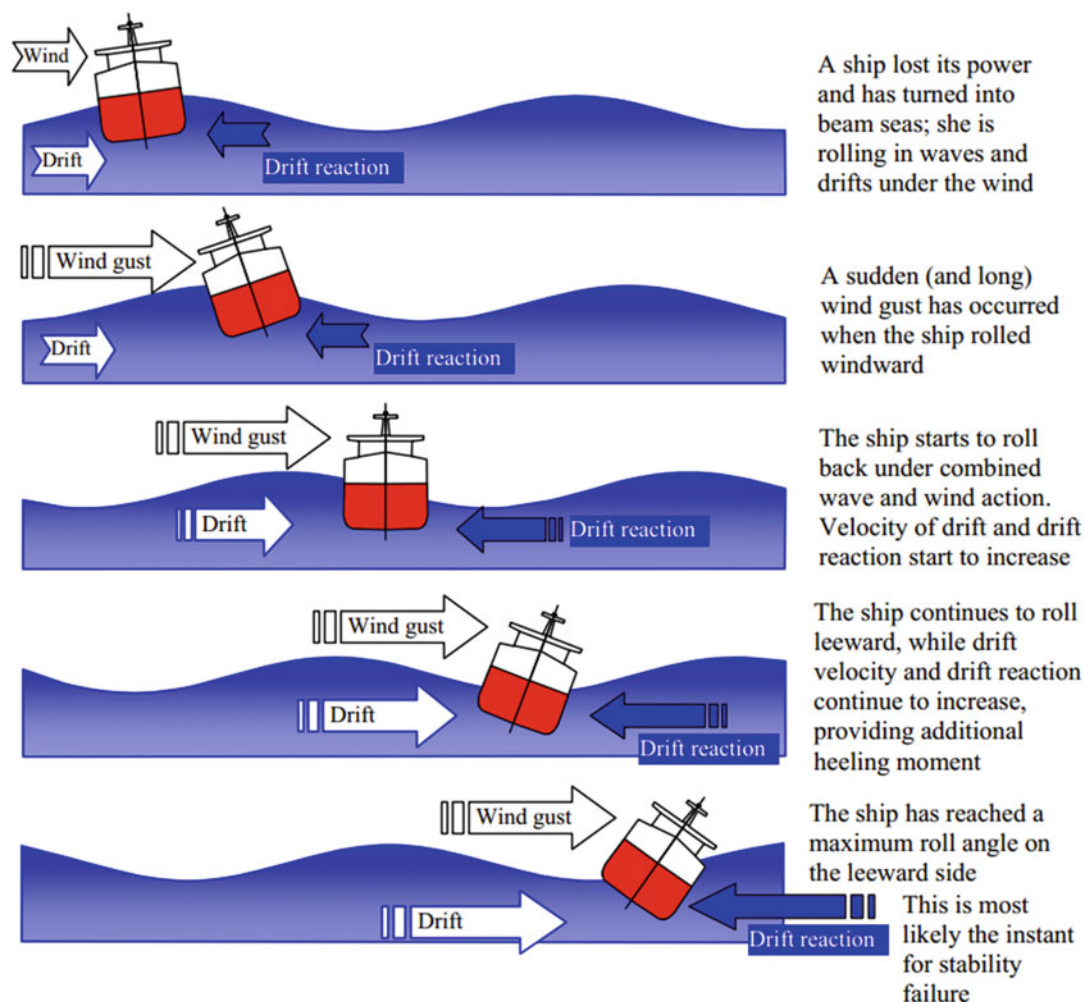
maximum windward angle, this is assumed to be the worst possible instant, action of wind is added to the action of waves. The strengthening wind increases drift velocity and this leads to an increase of the hydrodynamic drift reaction. The increase of the hydrodynamic reaction leads to the increase of the heeling moment. The gust is assumed to last long enough so the ship can roll to the other side completely. If the leeward roll angle is too large, or some openings may be flooded, the stability of the ship is considered insufficient (SDC 3/WP.5/Annex 6 [2016](#)).

Historical Development

Dead ship condition is the first mode of stability failure addressed with physics-based severe wind-and-roll criterion, also known as the “weather criterion,” which is adopted by IMO in 1985 (Res. A.562(14)) and is now embodied in section 2.3 of the 2008 IS Code, Part A. (Belenky et al. [2011](#)).

The severe wind and rolling criterion (weather criterion) is one of general provisions of the 2008 IS Code. This criterion was originally developed to guarantee the safety against capsizing for a ship losing all propulsive and steering power in severe wind and waves, which is known as a dead ship. Because of the forward velocity of ships is zero, this assumes an irregular beam wind and wave condition. Thus, operational aspects of stability are excluded in this criterion (IMO, MSC.1/Circ.1281 [2008a](#)).

The weather criterion firstly appeared in the IMO instruments as Attachment No.3 to the Final Act of Torremolinos International Convention for the Safety of Fishing Vessels, 1977. During the discussion for developing the Torremolinos Convention, the limitation of the GZ curve criterion based on resolution A.168 was remarked; it is based on experiences of fishing vessels only in limited water areas and it has no way for extending its applicability to other ship types and other weather conditions. Thus, other than the GZ curve criterion, the Torremolinos Convention adopted the severe wind and rolling criterion including a guideline of calculation.



Dead Ship Condition, Fig. 1 Scenario of stability failure in dead ship conditions (Belenky et al. 2011; SDC 3/WP.5/Annex 5 2016)

Because of the limitation of statistical data source of GZ curve criterion for passenger and cargo ships in resolution A.167, the discussion was raised at IMCO. As a result, a weather criterion was adopted also for passenger and cargo ships as well as fishing vessels of 45 m in length or over, as given in resolution A.562(14) in 1985. This new criterion keeps the framework of the Japanese stability standard for passenger ships but includes USSR's calculation formula for roll angle. For smaller fishing vessels, resolution A.685(17) was passed in 1991. Here the reduction of wind velocity near sea surface is introduced reflecting USSR's standard. When the IS Code

was established as resolution A.749(18) in 1993, all the above provisions were superseded.

In 2001 (IMO 2001a), following a submission from Italian delegation (IMO 2001b) arguing the methodology adopted to calculate several parameters of weather criterion, the IMO SLF Sub-Committee was tasked to start the revision of the Intact Stability Code as contained in Res. A.749. The working group operating in the frame of the SLF Sub-Committee was concentrated on polishing and restructuring Res. A.749 to make Part A of the Code mandatory, under SOLAS and ILLC Conventions, as requested by German delegation. This part was completed in 2007 with

adoption of the new International Intact Stability Code 2008 (IMO 2008b). This transformation provides alternative ways to comply with weather criterion for ship typologies which previously could be managed at national level (Francescutto 2019).

The basic principle of the weather criteria in IS Code 2008 is energy balance between the beam wind heeling and righting moments with a roll motion taken into account, as shown in Fig. 2, the energy required for restoring is larger than that required for the wind heeling moment. Since no roll motion is taken into account, a ship is assumed to suddenly suffer a wind heeling moment at its upright condition.

As the statement “Criteria included in the Code are based on the best state-of-the-art concepts, available at the time they were developed, taking into account sound design and engineering principles and experience gained from operating ships,” the new intact stability criteria include five stability failure modes developed, except for dead ship condition, which include excessive acceleration, pure loss of stability, parametric rolling, and surf-riding/broaching. The main purpose of these criteria is to enable the use of the latest numerical simulation techniques for evaluating the safety level of a ship from an intact stability viewpoint (IMO, Msc.1/Circ.1627 2020); the criteria will be still with reasonably changes.

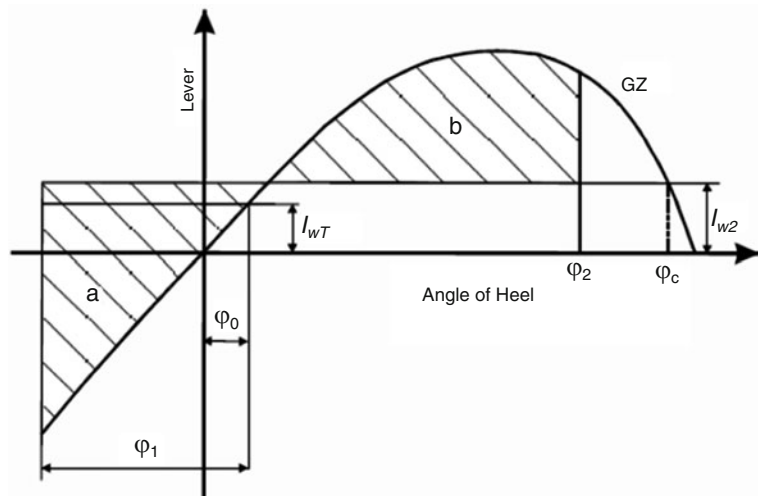
Experimental Method for Dead Ship Condition

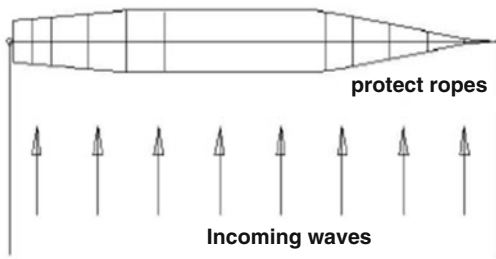
For the experiment under dead ship condition, three mooring-fixed model methods are used widely in beam wave. Firstly, the model is drifting freely in beam seas with two protected ropes which could also avoid the ship escaping from the worst scenario due to yaw moment as shown in Fig. 3a. Secondly, the ship model is orthogonal to the wave direction during the test through a wire system in which four wires connected to the ship model at bow and the stern, respectively, where the height is set be equal to water surface as shown in Fig. 3b. In the second method the drift and yaw could be softly restrained by the counterweight and heave, pitch, roll, and sway could be free. Thirdly, the ship model is also kept to be orthogonal to the wave direction by a wire system, however, in which four wires with four short springs connected to the ship model at bow and the stern, respectively, where the height is set be equal to water surface as shown in Fig. 3c. In the third method, the other ends of the four wires are fixed steadily, so the drift could be restrained and sway, heave, pitch, and yaw could be softly restrained (Gu et al. 2015).

The above experimental methods just consider the ship model in beam wave. However, the wind is also a key factor, which needs the laboratory equipped with wind blower. For the experiment under dead ship condition in beam wave and

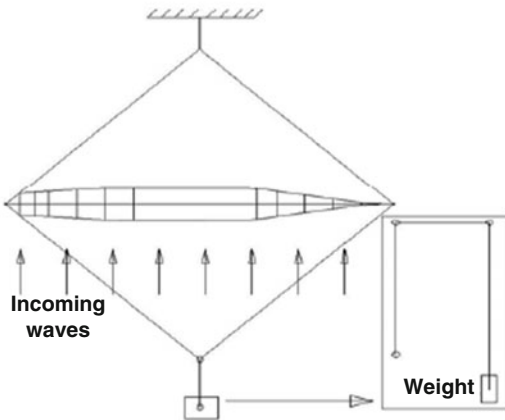
Dead Ship Condition,

Fig. 2 Energy balance method in weather criterion of IS Code 2008

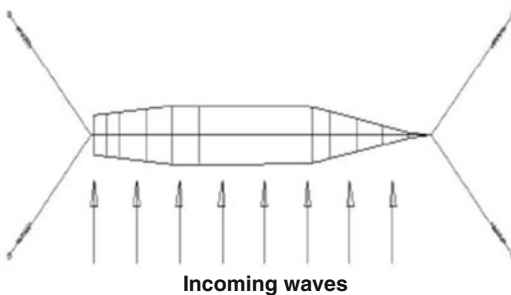




(a) Drifting freely



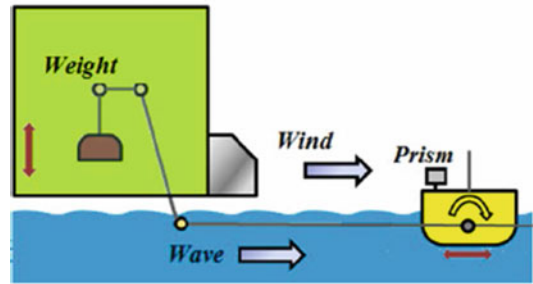
(b) Wire system with weight



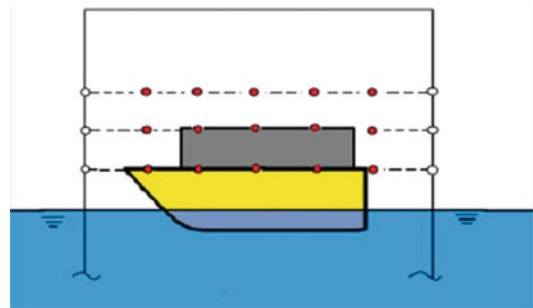
(c) Wire system with springs

Dead Ship Condition, Fig. 3 The experimental setup in beam wave

wind, the model is usually kept to be orthogonal to the wind and wave direction by a wire system, which softly restrains drift and yaw, as shown in Fig. 4. The wire system is connected to the ship model at bow and stern where the height is set to be equal to calm water surface based on measured hydrodynamic reaction force and moment in a captive model test of the subject ship. The mean



Dead Ship Condition, Fig. 4 Lateral views of experimental setup



Dead Ship Condition, Fig. 5 Measurement points for wind velocity

of fluid dynamic force in the sway direction is cancelled out by a counterweight. Waves can be generated by plunger-type wave makers. Fluctuating wind is generated by a wind blower in the wave direction. The wind velocity is directly measured with a hot wire anemometer, as shown in Fig. 5 (Umeda et al. 2014).

Numerical Method for Dead Ship Condition

For the motion of ship under dead ship condition, the roll is the most important part. When the ship moves into beam seas and wind with zero speed, the ship may encounter large amplitude roll motion.

Mathematical Model of Ship Motion Under Dead Ship Condition

1DOF Motion Equation

Considering dead ship condition, an uncoupled 1DOF roll mathematical model is widely used

for predicting ship motion. The mathematical model with stochastic wave excitation and wind moment taken into account is used as (1).

$$\ddot{\phi} + 2\mu\dot{\phi} + \beta\dot{\phi}|\dot{\phi}| + \frac{W}{I_{xx} + J_{xx}} GZ(\phi) = \frac{M_{wind}(t) + M_{wave}(t)}{I_{xx} + J_{xx}} \quad (1)$$

where ϕ : roll angle, μ : linear roll damping coefficient, β : quadratic roll damping coefficient, W : ship weight, I_{xx} : moment of inertia in roll, J_{xx} : added moment of inertia in roll, GZ : righting arm, $M_{wind}(t)$: wind induced moment consisting of the steady and fluctuating wind moment, and $M_{wave}(t)$: wave exciting moment of Froude-Krylov component. The dot denotes the differentiation with time (Gu et al. 2013).

The wind induced excitation moment can be calculated by following equation:

$$M_{wind}(t) = 0.5\rho_{air}C_m(U_w + U(t))^2A_LH_C \quad (2)$$

where ρ_{air} : air density, C_m : the aerodynamic drag coefficient, U_w : mean wind velocity, and $U(t)$: fluctuating wind velocity calculated by the Davenport spectrum. Furthermore, A_L : lateral windage area and H_C : the height of the center of the wind force from the center of the hydrodynamic reaction force.

The fluctuating wind velocity is calculated by following equation and Davenport spectrum is used.

$$U(t) = \sum_{i=1}^{N_w} b_i \sin(\omega_i t + \varphi_i) \quad (3)$$

$$b_i = \sqrt{2S_{wind}(\omega_i)\delta\omega}$$

$$S_v(\omega) = 4 \cdot K \cdot \frac{U_w^2}{\omega} X_D^2 (1 + X_D^2)^{-\frac{4}{3}}$$

where

$$K = 0.003, X_D = 600 \cdot \omega \cdot (\pi \cdot U_w)^{-1} \quad (4)$$

The wave exciting moment of Froude-Krylov component can be calculated by the following equation:

$$W_{wave}(t) = W \cdot GM \cdot \gamma \cdot \Theta(t) \quad (5)$$

where γ is the effective wave slope coefficient, and $\Theta(t)$ is a wave slope calculated by ITTC wave spectrum as follows:

$$\Theta(t) = \sum_{i=1}^{N_w} \frac{\omega_i^2}{g} a_i \sin(\omega_i t + \varphi_i) \quad (6)$$

$$a_i = \sqrt{2S_{wave}(\omega_i)\delta\omega}$$

$$S_{wave}(\omega_i) = \frac{A}{\omega_i^5} \exp\left[\frac{-B}{\omega_i^4}\right] \quad (7)$$

$$A = 172.75 \frac{H_{1/3}^2}{T_{01}^4}, B = \frac{691}{T_{01}^4}$$

where $H_{1/3}$ is the significant wave height and T_{01} is the mean wave period.

4DOF Motion Equation

1DOF model can obtain satisfied results in some wave condition, but other more accurate models are also used to simulate the motion under dead ship condition. If surge and yaw motion is constrained for realizing a beam sea condition in model experiments, sway, heave, roll, and pitch motions can be modeled as follows (Kubo et al. 2012):

$$m\ddot{x}_2 = F_2(x_2, x_3, x_4, x_5, \dot{x}_2, \dot{x}_3, \dot{x}_4, \dot{x}_5, \ddot{x}_2, \ddot{x}_3, \ddot{x}_4, \ddot{x}_5, t)$$

$$m\ddot{x}_3 = F_3(x_2, x_3, x_4, x_5, \dot{x}_2, \dot{x}_3, \dot{x}_4, \dot{x}_5, \ddot{x}_2, \ddot{x}_3, \ddot{x}_4, \ddot{x}_5, t)$$

$$I_{44}\ddot{x}_4 = F_4(x_2, x_3, x_4, x_5, \dot{x}_2, \dot{x}_3, \dot{x}_4, \dot{x}_5, \ddot{x}_2, \ddot{x}_3, \ddot{x}_4, \ddot{x}_5, t)$$

$$I_{55}\ddot{x}_5 = F_5(x_2, x_3, x_4, x_5, \dot{x}_2, \dot{x}_3, \dot{x}_4, \dot{x}_5, \ddot{x}_2, \ddot{x}_3, \ddot{x}_4, \ddot{x}_5, t) \quad (8)$$

where the suffices 2, 3, 4, and 5 denote sway, heave, roll, and pitch motions. The ship mass is indicated by m and the moment of inertia due to i th ($i = 1, \dots, 6$) direction is I_{ii} . The external forces or moments in the i th direction, F_i , are assumed to consist of buoyancy (B), the Froude-Krylov (FK), radiation (R), diffraction (D), gravity (G), wind (WD), and hydrodynamic reaction (HR) components as follows:

$$F_i = F_i^B + F_i^{FK} + F_i^R + F_i^D + F_i^G + F_i^{WD} + F_i^{HR} \quad (9)$$

Additional Requirements

For the dead ship condition failure mode, IMO Interim guidelines on the second-generation intact stability criteria takes additional requirements for adequately predicting ship motions (IMO, Msc.1/Circ.1627 2020):

1. Ship motion simulations should include at least the following four degrees of freedom: sway, heave, roll, and pitch.
2. Three-component aerodynamic forces and moments generated on topside surfaces may be evaluated using model test results. CFD results may be admitted upon demonstration of sufficient agreement with a model experiment in terms of values of aerodynamic force and moments. Empirical data or formulae could be applied within their applicability range.

Technical Challenges

Rolling Damping

A ship may occur extreme roll motion in severe waves and winds condition, thus leading to large-amplitude roll motion. Large roll angles and accelerations can cause significant crew inefficiency and reduce the stability of the hull, or leading to capsizing. A sufficient roll damping moment prevents the dynamic loss of stability, and roll damping is also fundamental in case of dead ship failure mode addressed by second-generation intact stability criteria, thus the accurate prediction of roll damping of ships is particularly important.

At present, the estimation of roll damping is both challenging as well as an extensively studied problem. Usually, there are three methods to evaluate the roll damping: semiempirical approach, model test, and numerical simulation. The most popular semiempirical method is component-based method, which considers the physical processes of roll motion damping with various means of energy dissipation, including friction, lift, wave-making, and vortex

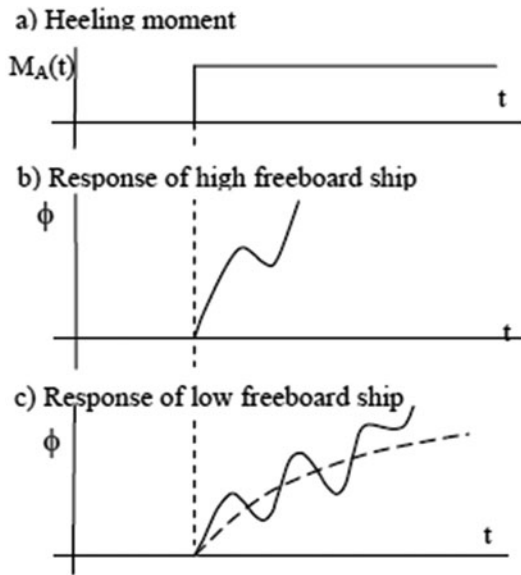
generation from the hull, as well as the influence of appendages (Ikeda et al. 1978; Himeno 1981). Unfortunately, the range of application is limited in most extreme roll motion. Different techniques of model test exist to estimate roll damping, which are roll decay motion, harmonic excited roll motion, and harmonic forced roll motion. Roll decay and harmonic excited roll based on roll angle measurements make the comparison of the roll damping coefficient between experiment and simulation difficult. Roll decay is fully nonstationary harmonic process, the initial state and higher Froude number have a strong influence on the test results. Considering the rudder is used to hold the model on course in harmonic excited roll motion, the propeller-rudder interaction adds the complexities for model test. The fixed roll axis of harmonic forced roll motion and the direct determination of the moment make it easier to measure roll period and amplitude.

Influence of the Deck Entering the Water

If a ship has lower freeboard for a substantial part of its length, the forces related to the entrance of the deck into the water may play a significant role in the dynamics of a ship. If the leeward side bulwark becomes submerged and a ship has significant lateral motion toward the immersed side, then a significant hydrodynamic reaction and corresponding additional heeling moment appear (Belenky and Sevastianov 2007).

There are two kinds of situation: water on deck and deck in water. Water trapped on deck (without an interface of the green water with the outside fluid domain) acts like a moving mass when the ship rolls, while the deck-in-water situation (where the water on deck and outside the hull can be considered as one fluid domain) leads to the development of hydrodynamic forces on the deck surfaces, which dominates the dynamics. These effects were studied since the late 1960s, and have been discussed previously at IMO (IMCO STAB/INF.27 1966) and in succeeding sessions.

The influence of the deck entering water leads to drastically different dynamics between high and low freeboard ships, under the action of a similar sudden gust of wind, see Fig. 6.



Dead Ship Condition, Fig. 6 Difference in response to a sudden wind gust

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Decommissioning of Fixed Platform

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Sustainable Arctic Marine and Coastal Technology (SAMCoT), Centre for Research-based Innovation (CRI), Norwegian University of Science and Technology, Trondheim, Norway

Synonyms

[Removal of fixed platform](#)

Definition

The decommission of a fixed platform is the final phase of its life cycle. The process starts by unplugging and abandoning the well; removing the conductors, subsea systems, and topsides followed by the installation process in reverse sequence; doing remediation work; and clearing debris from the project site. Modules can be removed and placed on barges and then towed to different locations depending on the planning for subsequent use. The usual methods of decommissioning are complete removal, partial

removal, and remote reefing (Ibrahim Khan and Islam 2007).

Scientific Fundamentals

Decommissioning Process

The decommissioning process of fixed platform is very complex, which can be divided into the following steps.

Planning and Engineering

The initial phase of decommissioning is to gather all the available information and prepare an Authorization for Expenditure (AFE) for owner approval. The structure is inspected, and an underwater survey may be performed. A lift analysis is developed by a structural engineer and reviewed by the project manager. Each well, pipeline, and platform requires permits for abandonment and removal which are prepared and submitted to the Bureau of Safety and Environmental Enforcement (BSEE) for approval and issuance before the work activities are performed. The project management group prepares a list of qualified contractors for each phase of the job, and the bid books are prepared and submitted to the owner for review and approval. Once approved, the contractors are sent a Request for Quotation (RFQ). Once received, the RFQs are reviewed by the project management team and recommendation for award sent to the platform owner for review and award of the work.

Well Plugging and Abandonment

After completing the preparation for decommissioning, it is time to start the on-site operation. Production wells that can no longer be used must be plugged to prevent the oil and gas reservoir fluids from migrating up hole over time and possibly contaminating other formations, fresh water aquifers, or the marine environment. A well is plugged by setting mechanical or cement plugs in the wellbore at specific lengths and intervals to prevent fluid flow. The main rig-less plugging process usually requires a spread containing the following: pump, cement blender, a combination wireline line unit, mud tank, and commonly

used materials such as cement, bridge plugs, and cutting equipment to conduct the operation. The plugging process can take from 7 to 10 days, depending on the number of wells on the structure, any issues in wellbore, and the service provider.

Pipelines Decommissioning

All pipelines connected to the platform are decommissioned. The decommissioning crew flushes the line by pumping a cleaning plug through the line with seawater. Divers or remotely operated vehicles (ROVs) will then expose the ends of the pipeline and cut the line above the riser bend and approximately 10 ft out from the base of the jacket. The riser bend is removed, and the cut end of the pipeline is plugged and buried 3 ft below the mudline. For the pipelines that run to shore, the line is cut just beyond the surf zone and plugged. The portion of the line that extends into the surf zone is removed to the beach line. The remaining line is plugged and buried.

Platform Preparation and Cleaning

Platform preparation includes the procedures associated with shutting down and preparing the facility for removal. Topside inspections and underwater inspections are generally conducted to determine the condition of the structure and to identify any problems for removal. Divers will perform the underwater inspection in water depths ranging from 0 to 200 ft, and ROVs will perform the underwater inspections from water depths ranging from 200 to 1200 ft. After the inspections have been completed, a crew paid on a day rate or fixed price bid prepares the structure for decommissioning after the wells have been permanently plugged and abandoned and the pipelines have been abandoned.

On the surface (topside of the platforms), the work includes the flushing/cleaning and degassing/purging of tanks, processing equipment and piping, disposal of residual hydrocarbons, removal of platform equipment, cutting of piping and cables between deck modules, separation of modules into individual units, installation of pad eyes for deck module lifting, removal of obstructions to lifting, and structural reinforcement.

Topside preparation also includes severing connections between the deck and jacket and verifying that all pipelines have had a section of pipe removed between the deck and jacket. Where permitted, marine growth can be removed from the structure at sea; otherwise it produces very foul odor during jacket disposal onshore.

Conductor Severing and Removal

The conductors can be removed prior to the HLV (heavy lift vessel) arrival or removed during platform decommissioning operations with the HLV. All conductors are severed and completely removed at least 15 ft below the mudline. The conductors can be severed by many methods, such as explosives, abrasive cutting, mechanical cutting, and diamond wire cutting.

Then casing jacks, platform crane, or the crane on an HLV can be used to pull the conductors. Regardless of the lifting device, the removal sequence is the same. The conductor is pulled upward until a 40-foot section is exposed. The conductor is cut using external mechanical cutters, or possibly the threaded connections are unscrewed. The cut section is then removed by the drilling rig or platform crane and placed on a workboat, barge or on the deck away from the work area. This procedure (pull, cut, and remove) is repeated until the entire conductor is removed. The jacks onboard may not be able to pull the combined weight if the conductor is grouted. In this case, a rig or crane is used until the jacks can pull the weight of the conductor by themselves.

Removing Topsides

Topsides consist of the deck and various modules. Its removal follows the installation process in reverse sequence. Each module is removed and placed on a barge. The module is secured by welding pieces of steel pipe (or plate) from the module to the deck of the barge. The deck section or support frames (cap trusses) are then removed by cutting the welded connections between the piles and the deck legs with oxygen-acetylene torches. All decks are removed in the same configuration as they were installed. Slings are attached to the deck/cap truss lifting eyes and to the HLV crane. The HLV's crane lifts the deck

sections from the jacket. The deck is then seated in the load spreaders and secured by welding steel pipe from the deck legs to the deck of the barge.

Substructure Removal

The substructure of fixed platforms can be roughly classified into jacket and concrete substructure.

Jacket Removal There are a few removal options including complete removal, partial removal, and remote reefing (Osmundsen and Tveterås 2003).

1. Complete removal

Complete removal of jacket begins with separating the structure from its foundation. After the legs and piles are cut and the diagonal braces removed, the jacket section is rigged, lifted, and sea-fastened on a barge. At times, the jacket is severed at multiple elevations because of its dimensions. The jacket is de-ballasted, picked up, towed to shallower water, set, cut in two (vertically), and removed in sections. This procedure is repeated until the jacket is completely removed and placed on barges. The structure is transported as a whole or in pieces to a scrap yard to be cut up and recycled.

2. Partial removal

This method suggests to remove only the top section of the jacket. The jacket's legs are severed as in complete removal, except that the cuts are made at the top portion of the jacket and the remaining part will be left in place. The remaining structure must not become a marine obstruction. The removed section of the jacket will either be transported to shore for disposal or reefed in place or at another approved reef location.

3. Remote reefing

Reefing may involve complete or partial removal of the jackets. The jackets are severed below the mudline (for complete removal) or at some partial depth (for partial removal) as described above. Structures may be reefed in place or at another approved location. If the jacket requires to be transported to an alternate

reef site, it will be de-ballasted as described in the complete removal method (Bull and Love 2019).

Concrete Substructure Removal There are three methods of concrete substructure removal:

1. Partial removal of the legs

After the deck and modules are removed, the upper part of the legs of the gravity base structures would be cut at the point of 55 m below sea level, in line with the International Maritime Organization (IMO) guidance. The rest of the structures would be left in situ on the seabed and documented in marine databases.

2. Leave in place

This method suggests the concrete substructures to be left wholly in place after the topsides being removed. As with partial removal, the location of the remaining structure will be marked and documented in marine databases and protected by a safety zone.

3. Complete removal

Complete removal of concrete substructure should follow a systematic procedure to avoid contamination and accidents. First, the integrity of the overall structure must be assessed. The drill cuttings accumulation on storage cells and the description of cell content should also be inspected. This operation can be carried out by an ROV (remotely operated vehicle) or AUV (autonomous underwater vehicle). Second, based on collected inspection results, remove the cuttings and store them in a barge. Third, remove the storage cell content. The oil storage cells are expected to be comprised of the following substances: attic oil, interphase material, water and sediment, etc. The last procedure is excavation of the concrete substructure foundation. The operation procedures of excavation is highlighted below:

Removing the conductors: The gravity platform structure is expected to be heavier than the original float-out, resulting from additions of conductors and sediments. Therefore, the removal of these conductors is necessary to reduce the weight. This will be done by a diver or ROV.

Sealing all penetration: A diver or ROV will seal all penetration areas to prevent the water from flooding into the structure during the refloating operation.

Installation of new control system: A new system would be required to monitor pressure in the cells, monitor and control the break out of the foundation, and monitor and control its draft and orientation during the tow to the shore. These systems are fitted onto various parts of the gravity platform using an ROV.

Ballast control system: All the water should be removed from the cells and legs using the hose fitted to an ROV to achieve the required buoyancy.

Equipment to break the under-base suction: The space between the foundation and the underlying seabed should be pressurized to move the foundation from the seabed.

Refloat and tow the foundation: As the water in the cells and leg is displaced by high-pressure air and sealed, a combination of buoyancy forces and under-base water pressure would lift the foundation from the seabed, and it will rise to the seabed. Once the structure is stable on the surface, it is towed near the shore site for the full dismantling process.

Disposal

The barge transports the whole topsides, including modules and deck, to a scrap yard. The modules may be lifted by cranes or skidded off the barge to the yard. Once the modules have been removed from the barge, the deck is skidded off the barge to the yard. All the structural components and modules are cut into small pieces and disposed of as scrap unless the deck will be reused. The production equipment is salvaged for reuse whenever possible.

Site Clearance and Verification

Following removal, the platform site and any location where the jacket is cut will be cleared of debris and verified that it has been cleared. Any trash that is retrieved from the sea floor is transported to shore for proper disposal.

Environmental Issues in Decommissioning

The main source of pollution in the decommissioning of platform is drill cuttings (the collective name for drilling mud, specialty chemicals, and fragments of reservoir rock) being deposited on the seafloor. It is clear that drill cuttings are a complex mixture which has an adverse impact on the environment. Each pile of drill cuttings represents a unique combination of sediment characteristics, contaminants, and benthic community, and each is affected by the local hydrodynamic regime. However, the constituents found in drill cuttings piles include heavy metals, specialty chemicals, organic contaminants, radioactive materials, and so on.

Available information on drill cuttings accumulations mainly focuses on three chemical groups, hydrocarbons, heavy metals, and to a lesser degree, radionuclides. Although there is uncertainty as to the extent to which oil concentrations reduce over time, it appears that most oil is dispersed during initial deposition. Thereafter, further degradation of oil in the pile will be very slow. In the case of drill cuttings piles, which contain potentially hazardous chemicals, synergistic effects from multiple contaminants should be considered. Dichloromethane extracted from sediments close to the platform are very toxic to Microtox, even after removal of sulfur. For the purposes of decommissioning, total amounts of wastes generation are required to be estimated. It helps to plan and develop appropriate measures during the decommissioning phase.

Key Applications

Decommissioning Regulations

1958 Geneva Convention on the Continental Shelf
This convention called for the complete removal of any installation that is abandoned or disused. It also outlines the responsibilities and duties of the states conducting oil and gas activities with regard to the continental shelf. Specifically, these activities are to be done in a way that does not interfere with the rights of other states. However, this convention does not contain specific regulations for

the methods of disposing of abandoned or disused structures.

1982 United Nations Convention on Law of the Sea (UNCLOS)

This international treaty on ocean governance came into force in 1994. Article 60(3) of this convention states:

Any installations or structures which are abandoned or disused shall be removed to ensure safety of navigation, taking into account any generally accepted international standards established in this regard by the competent international organization. Such removal shall also have due regard to fishing, the protection of the marine environment, and the rights and duties of other States. Appropriate publicity shall be given to the depth, position, and dimensions of any installations or structures not entirely removed.

This is to say that partial removal of marine structures may be permitted so long as the remaining structure does not have adverse effects on the natural environment, rights and duties of states, and navigational safety. This convention was the first to address the issue of sustainability with regard to disposal of offshore structures.

1989 Guidelines and Standards for the Removal of Offshore Structure (IMO)

These guidelines and standards specify that the decision to allow an offshore installation, structure, or parts thereof, to remain on the seabed should be based on a case-by-case evaluation. Matters to evaluate include, but are not limited to, potential effects on the marine environment, on navigation, or on other users of the sea and potential future effects relating to deterioration of materials.

The IMO guidelines and standards require monitoring of the accumulation and deterioration of material left on the seabed, to ensure there is no subsequent adverse impact on navigation, other uses of the sea, or the marine environment. This requirement is reflected in the EA by the requirement for a monitoring program at the site. Also among the standards in the IMO document is the requirement for the entire removal of abandoned or disused installations or structures standing in

less than 75 m of water and weighing less than 4000 t.

1992 Convention of the Protection of the Marine Environment of the North-East Atlantic (OSPAR) OSPAR is a regional treaty, only applicable in Europe, specifically to activities in the North Sea. The regulations for the decommissioning of offshore structures set by this convention are more stringent than the IMO guidelines (Hamzah 2003) and currently assume the complete removal of the structure and transport to shore for decommissioning, with limited allowances for alternatives (Ekins et al. 2006). Indeed, the OSPAR Decision 98/3 calls for:

- (i) The complete removal and onshore reuse, recycling, or disposal of all topsides and substructures or jackets weighing less than 10,000 t
- (ii) The complete removal of substructures weighing over 10,000 t, with the possibility to leave footings in place
- (iii) A consideration to derogate concrete gravity-based structures, floating concrete installations, and any concrete anchor base (ibid.)

Decommissioning Case

Brent Delta was one of the first platforms to be built in the very early days of Britain's oil and gas industry. It ceased production in 2011 after nearly 40 years of production. The topsides of Brent Delta was supported by a concrete GBS with three legs (two for drilling and one for utilities) that extend upward from the cluster of 19 concrete cells.

The Brent Delta topsides were removed and transported to the Able Seaton Port (ASP) facility at Teesside in 2017 after detailed technical and engineering studies, while its legs were left at sea. The SLV method was selected by Shell after analyzing several studies to determine the best method of removal the topsides (Fig. 1).

Plug and Abandonment of Wells

Before the topsides were removed, all of Brent Delta's 40 wells were plugged and made safe.

This program of work was performed in accordance with the "Oil and Gas UK Guidelines for the Suspension and Abandonment of Wells."

Removal of Conductors and Pipework

The topsides were linked to the GBS by pipework in the utility leg and by conductors in the two drilling legs. Most connections were severed at approximately 16 m above the height of the lowest astronomical tide (LAT), that is, slightly below the 19.8 m LAT height at which the concrete legs were cut. Other connections were cut at a lower level, but all were cut above the seabed.

Preparation for Lift

In order to ensure that the lifting beam of SLV contacted the topsides structure firmly and that topsides were strong enough to be lifted, eight lifting points were specially designed.

Cleaning of Topsides Process Facilities before Removal

The topside process systems were drained, purged, and vented, as appropriate, to ensure no pockets of hydrocarbon liquid or gas were present before the topsides were removed. Additional vents were created at selected locations in the topsides process system to ensure that they were not recharged from any trapped inventories, and drained systems were left open to the atmosphere to allow free-venting to occur so that gases did not build up.

Cutting the Legs

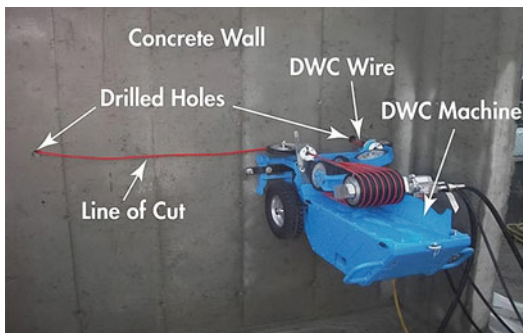
The topsides were fixed to the top of the legs by 5.5-m-long steel transition pieces located at +19.8 m LAT, just above the ring beam at the top of the concrete legs. In order to separate the topsides, a DWC machine was deployed inside the legs to cut through the grout and the bolts that secured the transition pieces to the ring beams (Fig. 2).

Removal of Attic Oil

When Brent Delta was in production, the 16 GBS cells were used to store oil prior to export. In preparation for CoP and subsequent decommissioning, the final amounts of crude oil



Decommissioning of Fixed Platform, Fig. 1 Brent Delta in 2013 (Shell UK 2018)



Decommissioning of Fixed Platform, Fig. 2 Using a diamond wire cutting machine on concrete legs (Shell UK 2018)

were exported from the cells and replaced by seawater. Because of the shape of the cell domes, however, and the configuration of the oil export line inside the domes of the Brent Delta cells, a small volume of crude oil and emulsion

remained trapped in the upper domes of the cells and could not be removed by the platform export system. This material, which amounts to about 50 m³ in each cell, is referred to as “attic oil.” It was removed to shore for treatment, recycling, or disposal as appropriate.

Lifting the Topsides

The Brent Delta topsides were lifted from the GBS as a single piece by the SLV Pioneering Spirit. Before finally committing to the operation, a detailed weather forecast was obtained by the Metocean engineer based on the vessel to determine whether the weather conditions would be within the design limits for the lift and a safe voyage to a predetermined safe harbor location afterward.

After completing pre-operational checks on the dynamic positioning (DP), ballast, and lift systems, the vessel was granted formal approval



Decommissioning of Fixed Platform, Fig. 3 Lift of the Brent Delta topsides (Brent Delta 2017)

from Shell and the Marine Warranty Surveyor to enter the 500 m safety zone and begin operations.

The vessel was accurately positioned so that the GBS legs fitted into the vessel's slot; the clearance between the vessel's hull and the platform substructure was sufficient to allow for some movement of the vessel. When all of the beams and yokes were in place, the SLV was progressively de-ballasted. Once the vessel was clear of the GBS legs, the hull connection beam was closed, and the vessel was ballasted to its transit draft, and all the X, Y, and Z drives of the lifting arms were placed into sea-fastening mode ready for transit (Fig. 3).

Cross-References

- [Complete and Partial Dismantling](#)
- [Ship Recycling](#)

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Decommissioning of Floating Platforms

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Definition

When the floating platforms reach the end of their lives or the oil field is exhausted, the platforms will be decommissioned. For floating platforms, there are two main decommissioning options: direct dismantling and reuse. The dismantling process of floating platform includes the following steps: (1) platform removal preparation, (2) well plugging and abandonment, (3) pipeline and flowline decommissioning, (4) structure removal, and (5) site clearance and verification (Kaiser and Liu 2015).

Scientific Fundamentals

Basic Decommissioning Method

Different types of floating platform will lead to different disassembly schemes:

1. Decommissioning of semisubmersible platform

For semisubmersible platforms, the piles anchoring the structure to the seabed will remain in place, but the chains and cables will be removed. The upper chain sections will be removed with the HLV during hull removal. Cables will be removed with anchor handling vessels with twin wench spools. The risers can be removed with the HLV or pipelay barge. According to the hull's metric ton, two or more tow tugs are mobilized and rig to the hull. The platform is towed to an onshore facility capable of handling refurbishment or scrapping. The hull will be dismantled or refurbished onshore.

2. Decommissioning of floating production storage and offloading (FPSO) systems

The ultimate disposition of the FPSO will depend upon its condition at the end of the productive life and available options for further use. If the FPSO is decided to be dismantled, it will be towed from the site to a shipbreaking yard in accordance with international conventions. In abandonment, the production system will be isolated from the subsea wells. The production system will be flushed from the FPSO end using seawater to displace any residual oil and production fluids. The flushing water will be returned to the FPSO for treatment. Any residual hazardous waste will be taken to shore and treated at waste treatment facilities. Once the production system has been flushed and confirmed clean, the FPSO will be released from the mooring system for removal. The lines and chains from mooring system will be recovered. The steel suction piles will be abandoned in place. The topsides equipment will be removed and recovered offshore. The hull of FPSO will be towed to the dock.

3. Decommissioning of tension leg platform (TLP)

For tension leg platform, its mooring system consists of tendons and risers. The TLP is ballasted to slacken the tension leg tendons. These tendons will be dropped to the ocean floor and cut loose from anchorage with ROV. The tendons will be retrieved the tendons using an ROV and a dynamically positioned derrick barge. The de-ballasted buoyant hull can then be floated and towed to near shore for reuse or scrapping. For mini tension leg platforms, its hull can be floated or placed on a cargo barge and transported to near shore for reuse or scrapping.

4. Decommissioning of spar platform

There are a few decommissioning options including in situ disposal, dismantling in horizontal or vertical position, and artificial reefing.

(1) In situ disposal

The in situ disposal method suggests to sink the spar at the original working site. This is by far the simplest and cheapest option and also the safest since it would involve minimal exposure of the

workforce to hazardous operations. However, it leads to the releasing of the spar's contents into the marine environment and leaving a large obstruction on the seabed.

(2) *Dismantling in vertical position*

For this option, the platform needs to be towed to a sheltered deepwater area, partially decontaminated and cut into horizontal slices in vertical position. The slices will be transported ashore for final disposal. However, cutting in vertical position is difficult and dangerous to perform, exposing the workers to hazardous operations. If the hull loses control during cutting operation, the structure may sink and break up. Besides, the suitable deepwater area is hard to find.

(3) *Dismantling in horizontal position*

This method includes the steps of (1) removal of the topsides section, (2) repair of the damaged storage tanks, (3) rotating the hull to the horizontal position, (4) transferring the hull to a cargo barge, and (5) transporting the hull ashore for final disposal. The most critical step is to turn the spar from vertical to horizontal position, which is expected to take between 5 and 10 days to complete. This process begins with the installation of the valves, pumps, and pipelines of the reverse upending control system. Then the spar would be restricted by a pre-installed mooring system. The contaminated water in the storage tanks will be pumped into the tanker and replaced by an inert gas mixture of nitrogen and carbon dioxide. By ballasting and de-ballasting the tanks, the spar would be gradually tipped to the horizontal position. This operation should be done carefully to avoid exposing the tank walls to excessive stress which may lead to structure failure. The repaired tanks are still vulnerable and have to be orientated out of the water as much as possible.

(4) *Artificial reefing*

Artificial reefing is assessed as the simplest decommissioning option from the technical perspective. In the typical

procedure, topsides will be cleaned and removed. Explosive charges will be emplaced. And then tow the spar to the disposal site. The reefing process begins with releasing the tow lines and moving away the tugs. Finally the explosives are detonated by remote control.

Reuse Option

Compared to dismantling directly, redeploying the platforms to another field is economically much more attractive.

1. Reuse of TLP

In order to reuse the TLP, the tendon length should be adjusted, and the mooring system requires a reinstallation.

2. Reuse of semisubmersible platform

The reuse of the semisubmersible platform is similar to that of TLP. The platform needs to be towed to new waters, and the mooring system will be adjusted.

3. Reuse of FPSO

FPSO is easy to migrate from one domain to another. Generally, a FPSO will serve in 4–5 oil fields in its life. The anchors extended mooring or turret mooring systems can be picked up and reused in new oil field.

4. Reuse of spar platform

The major obstacle to the reuse of spar platform is that the long slender underwater column is hard to be towed around in vertical position. Therefore, the hull must be de-ballasted and rotated into horizontal position after the drilling and production deck being removed. Then the hull can be towed or transported by a giant cargo barge or semisubmersible. The anchor lines and sea floor anchoring system can be modified and reinstalled based on the requirements of the new location (MMS 2009).

Key Applications

Around 200 floating structures have been identified as scrapped globally since 2015 (NGO Shipbreaking platform 2019). In the following

section, a typical decommissioning case is presented.

Typical Decommissioning Cases

Red Hawk spar is the world's first cell spar platform with a 560-ft-deep hull consisting of seven giant cylinders. It ended life as the deepest floating production unit ever to be decommissioned in the Gulf of Mexico in 2014. After considering all available decommission options, it is decided to convert the hull into an artificial reef. This is the first spar ever to be reefed. The process is introduced below (Fig. 1).

1. Mooring disconnect

The mooring system of the spar consists of a windlass and an 800-ft-long work chain located on the topsides to service all six mooring legs. The work chain was cut in three pieces and paid out on three mooring legs, thus effectively lowering the tension in the entire mooring system. In addition, to remain

the storm safe during the topsides lift, the high capacity C-link connectors were used to connect the work chain segments to the top chains. Tugs were hooked up to provide station keeping while the mooring disconnect was completed.

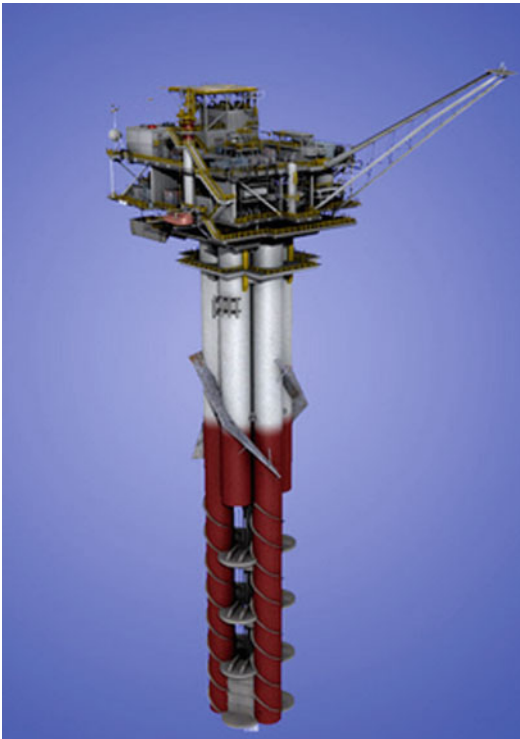
2. Ballasting and de-ballasting

During ballast and de-ballast, the access to the spar is limited. Therefore, it needs a remote ballast/de-ballast system for all operations. The spar hull was 560 ft. long, 64 ft. in diameter and assembled with six outer cells connected to the center cell. Each cell had void tank at the top and variable ballast tank at the bottom. In this phase, the hull was ballasted so that the heavy lift vessel can hook up and lift the topsides. Then the hull was de-ballasted to reach a tow draft of 400 ft in order to be able to enter the 430-ft-deep reefing site (Fig. 2).

3. Towing and reefing

The towing route was pre-surveyed several months ahead of the tow to verify information obtained from public databases as well as to account for any seabed objects or anomalies. The pipeline crossings and collisions with other platforms should be avoided. It is also necessary to maintain adequate bottom depth clearances during the approach to the reefing site. With some convenient seas assisting from the stern, the tow lasted less than 2 days.

After arriving at the reefing site, the air from variable ballast tanks was vented in order to let more water in to lower the spar, until the spar was sitting firmly on the seabed in a vertical position. After that, crew was transferred to the spar to prepare the hull for the reefing operations. Once complete, all air valves for variable ballast tanks were left open and the crew transferred back to the derrick barge. One tug disconnected from the hull, and the other two pulled in the planned reefing direction until the hull began to free flood. The hull gradually pivoted around the bottom edge of the two cells and laid on the bottom in a horizontal position in less than 10 min. During the sinking, the tugs powered off and paid out work rigging, which were then disconnected from tow rigging with ROV assistance (Fig. 3).



Decommissioning of Floating Platforms, Fig. 1 Red Hawk spar (Decom World 2015)



Decommissioning of Floating Platforms, Fig. 2 Topsides being lifted off towing (Decom World 2015)



Decommissioning of Floating Platforms, Fig. 3 Hull sinking (Decom World 2015)

Cross-References

- [Complete and Partial Dismantling](#)
- [Decommissioning of Fixed Platform](#)
- [Ship Recycling](#)

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Decommissioning of Offshore Oil and Gas Installations

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Synonyms

[Derrick barge \(DB\)](#); [Environmental, Social, and Health Impact Assessment \(ESHIA\)](#); [Floating Production Storage Offloading \(FPSO\)](#);

International Maritime Organization (IMO); International Standards Organization (ISO); Jack-up (JU); Oslo/Paris Convention (OSPAR); Pipeline End Manifold (PLEM); Semi-submersible Crane Vessel (SSCV); Sheerleg (SL); Tension Leg Platform (TLP); Umbilical Termination Assembly (UTA); UN Convention on the Law of the Sea (UNCLOS); United Nations (UN)

Definition

Decommissioning of offshore oil and gas installations is generally regarded as a major challenge for the industry with technical difficulties, significant associated risks, and high costs. As the industry is beginning to gain some experience with some major installations in the North Sea being successfully removed and with over 100 facilities already decommissioned, it is becoming clear that the industry is on its way towards facing and meeting the challenges of decommissioning. This chapter discusses the main guiding principles of decommissioning and international laws and regulations relating to decommissioning which form the main obligations for countries and oil and gas operating companies. These regulatory obligations form the starting point for developing plans and processes for decommissioning together with engineering design and execution aspects. In the final analysis success depends on sound engineering to address technical solutions and select the most appropriate ones for implementation. The most appropriate solutions would be those that score well on the environmental front, safety, and cost. Experience to date is also described, with specific examples, noting achievements in terms of recycling of materials, safety record, and recent advances in industry capability.

Decommissioning Principles

The main principles of decommissioning of offshore installations are:

- Decommissioning should aim to achieve a clear sea bed (acknowledging that this will not always be achievable, given the technical complexities and risks involved).
- A “polluter pays” principle is adopted which means that we expect those who have benefited from exploitation of hydrocarbons to bear the responsibility for and cost of decommissioning.

International Regulations Relating to Decommissioning

The prime international regulations and guidelines relating to decommissioning are summarized below. It should be clarified that these regulations do not carry force of law as such but individual nations or regional lawmakers may enact such laws with reference to the international regulations. Many countries have enacted such laws. The most relevant international laws and conventions are:

1. OSPAR Convention and OSPAR Decision 98/3 (OSPAR 1998)
2. UN Convention on the Law of the Sea, 1982 (UNCLOS 1982)
3. IMO Regulations (IMO A.672 (16) 1989)
4. ISO Requirements (ISO 19901-6 2006; ISO 19901-9 2018)

OSPAR Convention

Under the OSPAR Convention for the Protection of the Marine Environment of the North-East Atlantic (OSPAR 1998) it is prohibited to dump disused offshore installations or leave them wholly or partly in place within the area covered by the convention. The decision specifying this (OSPAR Decision 98/3) recognizes that certain types of structures may be difficult to remove and it is therefore possible to apply for derogations from the general rule in specific cases. The sea area covered by this convention includes the North Sea, West of Shetlands, and parts of the Arctic.

Under Decision 98/3 it is required that: (1) the topsides of all installations must be returned to the shore; (2) all steel installations with a jacket

weight less than 10,000 tonnes in air must be completely removed for reuse, recycling, or final disposal on land.

The decision recognizes that there may be difficulty in removing the footings of large steel jackets (weighing more than 10,000 tonnes) and in removing concrete installations. Derogations will only be granted if there are significant reasons why an alternative disposal option is preferable to reuse or recycling, or disposal on land.

Because of the significance of the North Sea in offshore oil and gas activity, having more than 800 offshore installations in this sea area, the impact of OSPAR Decision 98/3 extends beyond the region and has global implications.

UN Convention on the Law of the Sea (UNCLOS 1982): Provisions on

Decommissioning

Article 60.3 of UNCLOS states that any installations or structures that are abandoned or disused shall be removed as necessary to ensure safety of navigation, taking account of any generally accepted international standards established in this regard by the competent international organization (the International Maritime Organization – IMO and International Standards Organization TC/67 Covering Offshore Oil and Gas Standards). Such removal shall have due regard to fishing, the protection of the marine environment, and the rights and duties of other states. Appropriate publicity shall be given to the depth, position, and dimensions of any installations or structures not entirely removed.

Furthermore, through Article 210 of UNCLOS member states are legally obliged to adopt laws and regulations to prevent, reduce, and control pollution of the marine environment by dumping, which must be no less effective than the global rules and standards. Article 210 also provides that such laws, regulations, and measures must ensure that dumping is not carried out without the permission of the competent authorities of States.

IMO Regulations Relating to Decommissioning

In October 1989, the IMO adopted guidelines and standards for the removal of offshore installations

and structures on a nation's continental shelf or its exclusive economic zone (Resolution A.672, IMO A.672 (16) 1989), in consultation with IMO member nations. The IMO guidelines and standards provide that, in general, an abandoned or disused offshore installation or structure on a continental shelf or an exclusive economic zone should be removed as soon as reasonably practical once it is no longer serving the prime purpose for which it was originally designated. The following standards should be taken into account when a decision is made regarding the removal of an offshore installation or structure:

- All abandoned or disused installations or structures standing in less than 75 m of water and weighing less than 4000 MT in air, excluding the deck and superstructure, should be designed to be removed.
- All abandoned or disused installations or structures emplaced on the seabed on or after 1 Jan 1998, standing in less than 100 m of water and weighing less than 4000 MT in air, excluding the deck and superstructure, should be entirely removed.
- Removal should be performed in such a way as to cause no significant adverse effects upon navigation or the marine environment.

Alternatively, the guidelines also provide guidance for leaving offshore installations wholly or partially in place if complete removal:

- is not technically feasible
- would involve extreme cost
- would involve an unacceptable risk to personnel or the marine environment

but should ensure in such cases that an unobstructed water column sufficient to ensure safety of navigation (and, in any event, of not less than 55 m) is maintained above the partially removed installation.

The IMO guidelines and standards which were designed essentially to ensure the safety of navigation make clear that they are not intended to preclude a coastal state from imposing more stringent removal requirements for existing or future

installations or structures on its continental shelf or in its exclusive economic zone.

ISO Requirements Relating to Decommissioning

ISO requirements for decommissioning appear primarily in 19901-9 Structural Integrity Management (ISO 19901-9 2018) and 19901-6 Marine Operations (ISO 19901-6 2006). These requirements are discussed and referenced further below under decommissioning activities.

Costs of Decommissioning

With about 6000 offshore installations worldwide and thousands of km of pipelines the costs of decommissioning may appear colossal. As example the cost of decommissioning Norway's 120 offshore installations (fixed platforms plus floating) plus 350 subsea systems has been estimated to be about US\$20 billion (Decommissioning of Offshore Installations 2011), while the projected cost of decommissioning of UK's 550 offshore installations has been estimated by UK Oil and Gas Authority (in 2018) to be about US\$75 billion. On this basis the total bill for decommissioning 6000 installations (many of them being steel well-head jackets which are relatively easier to remove than their North Sea counterparts) is likely to be of the order of US\$400 billion. These are staggering sums which are sometimes misunderstood leading some to conclude that these costs are prohibitively high. These are very large sums of money but when they are viewed within the context of the size of the oil and gas industry, they are not prohibitively high. When it is borne in mind that the decommissioning activity will be spread over a period of about 40 years, then an annual cost of US\$10 billion is not dramatic. Furthermore, to put things further in perspective this decommissioning cost is equivalent to the revenue from recoverable oil reserves of 10 billion barrels, at an average price of US\$40/barrel. The total recoverable reserves of the offshore oil and gas industry (produced and underway with the infrastructure already in place) are about 300 billion barrels of oil equivalent. On this basis the cost of decommissioning is equivalent to about

10 out of 300 or 3.3% of the sale value of oil and gas extracted. If further we take into consideration that the cost of decommissioning is incurred at the tail end of field life and use a discount rate of 5%/annum for the time value of money, then the cost of decommissioning is reduced further from 3.3% to about 2.5%. It should be recognized however, that another 50% or so of the sale value is used up in capital and operating expenditures. Hence if we consider a large project which leads to a sale revenue of US\$100 billion, then approx. US\$50 billion is used up as capital and operating expenditure, US\$3.3 billion in decommissioning, and the remaining US\$47 billion is "profit" which is shared between the government and the oil companies undertaking the investment, with the government share being about 60%. Thus provided that the companies involved and the governments make appropriate provisions, the cost of decommissioning is a relatively modest component of the total cost and is affordable.

Decommissioning Process and Main Activities

An overview of the decommissioning process is given below. A decommissioning plan needs to be developed during the design of the installations so as to ensure that a credible method of decommissioning has been considered and design provisions have been made where necessary. For instance, if the substructure is a concrete gravity base structure, which is very heavy and cannot be removed by lifting (e.g., Sakhalin Gravity Base Structures), then a method of re-floatation (e.g., by pumping air into certain compartments) needs to be developed and provisions made during construction.

The decommissioning process in general may consist of the following activities:

- Planning and engineering phase of decommissioning
- Well decommissioning and conductor removal
- Topside facilities decommissioning and removal
- Substructure decommissioning and removal
- Pipeline and subsea decommissioning.

Each of these activities is discussed further below.

Planning and Engineering Phase of Decommissioning

At the start of the planning phase field surveys are carried out to establish the status and condition of the installation. The data collected during the survey includes weights and center of gravity for platform modules in the topsides and for the substructure. Lifting points are identified and their integrity is verified. Locations where structural cuts may be needed for lifting are identified. Topsides equipment and piping is surveyed and assessed for cleaning and decontamination requirements to avoid pollution. Information is gathered on wells, drill cuttings, pipelines, risers, and all subsea equipment.

Engineering and design work is carried out to the level needed to allow selection of the preferred execution plan to verify that decommissioning and removal can be executed safely satisfying engineering and environmental standards. Removal should include site clearance procedures that comply with international and national regulations. Typically these regulations require removal of all components above the sea bed and piling removal to at least 5 metres below sea bed.

Components left above the sea floor (e.g., bottom sections of jackets, provided appropriate derogation has been obtained, suction piles, pipeline exchange manifolds, and subsea wellheads have been plugged and abandoned) are likely to require placement of navigational aids for shipping. This should form part of the decommissioning plan which will need to be submitted to the regulator for approval.

Well Decommissioning and Conductor Removal

Well decommissioning involves the plugging and abandonment of the well bores. This is one of the essential and primary stages of decommissioning. It consists of isolating productive zones of the well with cement, removing the production tubing and setting a surface cement plug in the well with the top of the plug a set distance below the sea

floor. In many cases it can be advantageous to plug and abandon wells as they become unproductive towards the end of field life so as to reduce the potential environmental and life safety exposure.

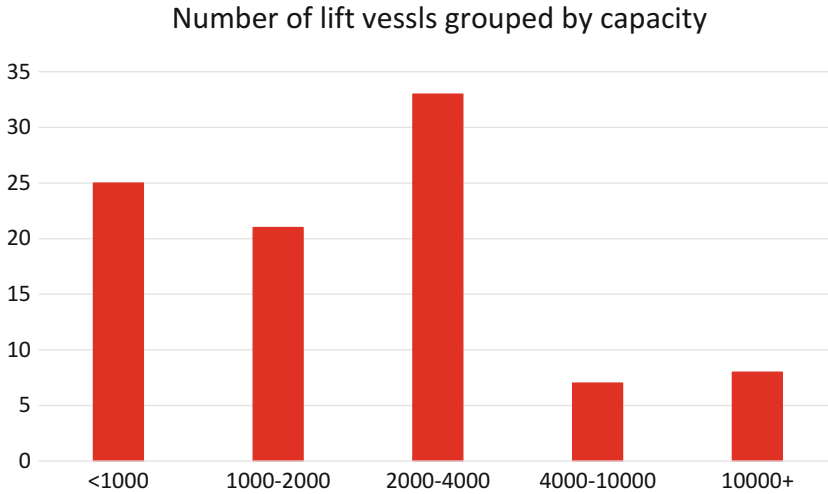
All components which span the water column, namely the conductor and parts of the casing strings, are cut at a suitable distance below mudline, and are removed prior to removal of the substructure.

Topside Facilities Decommissioning and Removal

Facilities decommissioning involves the flushing, cleaning, and removal of process equipment and facilities as well as the removal and environmentally sound disposal of waste streams.

The removal operations for topside facilities will in general utilize one of the following three methods:

- **Piece small dismantling:** The installation is dismantled offshore and cut into small sections which can be shipped onshore on barges or in containers.
- **Heavy lift:** Sub-assemblies like a complete module is lifted using a heavy-lift crane vessel following a sequence of reverse installation. The module is placed on a barge or on the crane vessel and transported to a decommissioning yard. Typical large offshore crane vessels are the **Thialf** and **Bolder** of Heerema or **Saipem 7000**. Owing to the heavy-lift vessel being the conventional method of installation and decommissioning of offshore platforms, a number of conventional lift vessels are available across the globe. This has also fostered a market with a healthy competition and competitive pricing. In naming convention, the following terms are commonly used to describe different lift vessels:
 - DB (Derrick barge or monohull crane vessel)
 - SSCV (Semi-submersible crane vessel)
 - SL (Sheerleg)
 - JU (Jack-up)



Decommissioning of Offshore Oil and Gas Installations, Fig. 1 Number of heavy-lift vessels grouped by lift capacity in tonnes

It is common to rate the capability of crane vessels by their maximum lifting capacity as illustrated in Fig. 1. Crane reach over the stern is also important, and also the station-keeping capability.

The jack-up lift barge is mainly designed for the offshore wind industry but could also be used for decommissioning projects.

- **Single Lift:** The entire topsides is removed in a single lift and transported onshore. If the topside weight is less than about 10,000 tonnes the above heavy-lift vessels may be used to carry out a single lift. A relatively new vessel constructed with decommissioning operations in mind is the **Pioneering Spirit** of AllSeas. This is a twin-hull vessel with a U-shaped slot at the bow which is 122 m long and 59 m wide. The width of the slot enables the vessel to embrace the platform within the slot and thereby lift the entire topsides up to a max weight of 48,000 tonnes, in a single lift using eight sets of specially designed horizontal lifting beams.

In 2017 the *Pioneering Spirit* executed a record-breaking operation with the 24,000 tonnes single-lift removal of Shell's iconic Brent D platform topsides in the North Sea as illustrated in Figs. 2, 3, and 4.

Substructure Decommissioning and Removal

Steel Substructures

Decommissioning of a **steel substructure** normally involves cutting of the foundation piles at a certain distance below sea bed. Local (national) regulations may have specific requirements regarding the minimum depth of removal below sea floor. Methods of cutting the steel piles below the sea floor (at a depth of typically 1 m below sea floor) are already available. The substructure may be then removed by lifting as a single piece, if it has a weight of less than 10,000 tonnes (in air). This is within the lifting capability of a large offshore crane vessel like the *Thialf* or *Saipem 7000*. If the weight exceeds this amount the substructure may need to be cut into two or more pieces and lifted using separate lifts. This picture is changing after 2016 due to the availability of the *Pioneering Spirit* which has capability to lift a steel jacket up to 20,000 tonnes in a single lift.

The complete list of options for steel substructures is similar to that of topsides, namely:

- Piece small dismantling
- Single or multiple lift using a heavy-lift vessel
- Single lift using *Pioneering Spirit*



D

Decommissioning of Offshore Oil and Gas Installations, Fig. 2 Pioneer Spirit approaches Brent D platform, April 2017



Decommissioning of Offshore Oil and Gas Installations, Fig. 3 Beams connected to Brent D Deck, ready to start lifting by de-ballasting, April 2017



Decommissioning of Offshore Oil and Gas Installations, Fig. 4 Lifting of Brent D deck onto Pioneering Spirit completed, April 2017

Subject to national and regional regulations the substructure may be toppled in place to form an artificial reef or transported and placed at a designated reef site. Also the lower part of a steel jacket may be left in place as a reef if it is demonstrated that there is no practicable way to remove it and appropriate derogation is obtained, again subject to national or regional regulations. It is advisable to integrate structural decommissioning activities with the late-life structural integrity management plan for the platform, for instance inspections should be targeted to verify data (e.g., weights, CoG) and integrity information (e.g., for lifting points which will be used during removal) required for decommissioning activities.

Removal of large N Sea steel jackets: North-west Hutton was the first of the N Sea steel giants to be decommissioned. BP as operator examined the option of complete removal of the jacket by cutting into two pieces and concluded that this was not practicable. The company obtained derogation from UK government to leave the bottom third of the jacket including the footings and the drill cuttings in place. The top part of the jacket was removed by Heerema in 2009 using the Hermod HCV.

The **Murchison** steel substructure also in the Northern North Sea, operated by CNR International, was the second of the steel giants to be removed. The entire steel jacket was removed successfully in 2017 by Heerema using the Thialf heavy-lift vessel. The steel jacket with a total weight of 14,000 tonnes was cut horizontally below water and removed in two separate lifts.

Shell's Brent: A steel jacket with a total weight of 25,800 tonnes (excluding conductors) is among the heaviest in the North Sea and lies in a water depth of 140 m. The platform is supported by eight large footings (skirt piles) extending about 40 m above the sea bed. These are very difficult to remove. The company has carried out detailed assessments of removal options and has concluded that the top part of the jacket down to a depth of 84.5 m below sea level will be cut and removed, while the bottom part including the footings will be left in place (see Brent 2017), with a derogation similar to that for NW Hutton.

Concrete Substructures

Concrete substructures: Removal of heavy concrete gravity base structures is recognized to be extremely difficult and if possible it certainly carries significant risks. There are 18 such

installations in the North Sea including three in the Brent Field (Brent B, C, and D), Cormorant A, Dunlin, and 12 in Norway. The weight in air of each of these is in excess of 100,000 tonnes and hence is beyond the lifting capability of any vessel. Re-floating may be an option provided that the integrity of the compartments to be re-floated can be verified. During installation the base of the structure was often grouted to improve stability. The weight of this grout is large but it is unknown and cannot be removed. Recognizing the above challenges OSPAR regulations clearly allow for concrete substructures to be left in situ or be partially removed. If the part above sea water is removed it is required for the removal to be extended down to 55 m below sea level so as to ensure that the remaining part of the structure is not a shipping hazard.

Floating Substructures

The decommissioning of a floating system can be simply described as a reverse installation. Semi-submersibles, FPSOs, and TLPs can be removed by disconnecting the mooring lines and risers and towing the floating hull (with topsides) back to an onshore location for disposal. The topsides may be removed offshore by a crane vessel if that is considered preferable. For a spar the topsides need to be removed first (because they were installed last). The spar hull may be de-ballasted and placed horizontally onto a transport vessel (e.g., mighty servant) for transport to an onshore location.

An alternative to decommissioning is reuse in another field. Hence selling of the asset to an interested party for reuse is an option which has already happened on several occasions, especially with FPSOs (Fig. 5).

Pipeline and Subsea Decommissioning

A typical subsea infrastructure (see Fig. 6) may consist of several subsea wells connected via flowlines to a manifold and from the manifold connected to a floating or fixed platform via one or more flowlines. Export may be through a subsea pipeline. Additional subsea structures may include Pipeline End Manifold (PLEM), Umbilical Termination Assembly (UTA), Subsea Control Module, etc.

Given the variety of elements the decommissioning plan will involve many different activities. Here we generically address the decommissioning of (i) a subsea module, e.g., a well manifold and (ii) flowlines, pipelines.

Decommissioning of a Subsea Module, E.G., Well Manifold

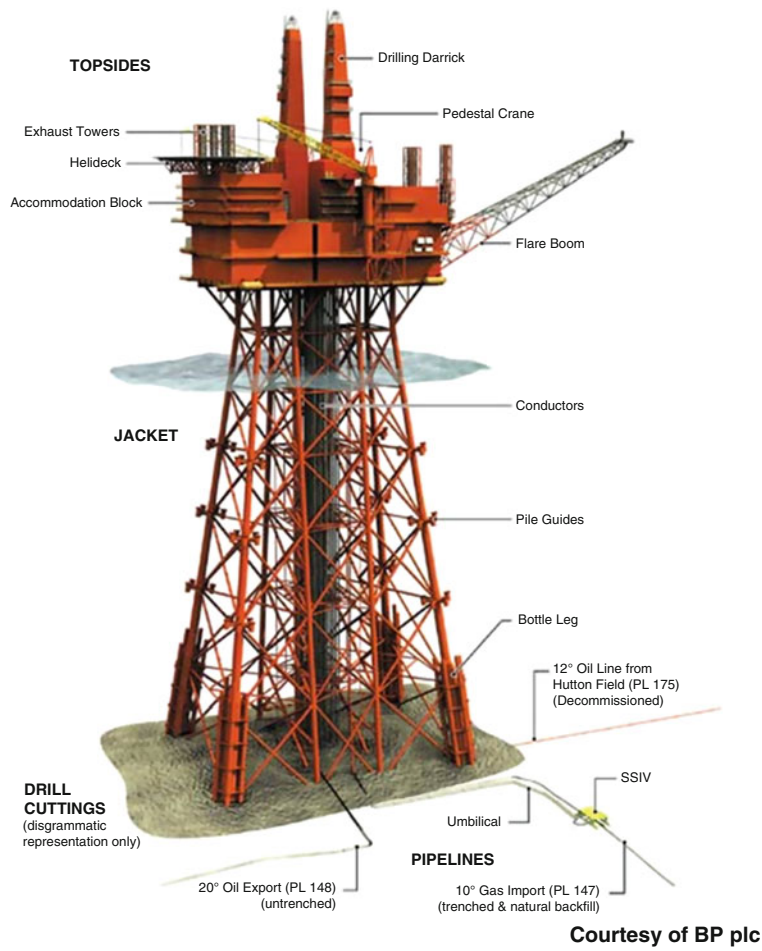
Subsea structures need to be flushed clean so as to remove any remaining hydrocarbons before they are left in place or removed. In shallow water (e.g., less than 200 m) or areas subject to trawl fishing subsea structures are most likely to be removed so as to leave a clean sea bed. Structures in deeper water are candidates for decommissioning in situ. If the module is piled to the sea bed, the piles need to be cut first. Thereafter the module is lifted using a crane to a barge for transport to shore for further disposal.

Decommissioning of a Flowline or Pipeline

The flowline or pipeline decommissioning plan may depend on geographic location and/or national or regional regulations. Pipelines can be decommissioned and left in situ or completely removed. If the pipeline is already buried, the most likely decommissioning plan will involve disconnecting the pipeline from the platform, cleaning and plugging, and leaving it in place, ensuring that both ends are buried. The pipeline is normally filled with water before plugging. In the latest edition of UK Guidance Notes on Decommissioning it is stated that when pipelines, mattresses, and associated items are decommissioned in situ, it should be aimed to achieve a burial depth of 0.6 m to limit the risk that sea bed currents may expose the pipeline in the future. In some jurisdictions flexible pipe and umbilicals have to be removed from the seabed due to their plastic content.

Environmental Assessment

Consideration to environmental data collection for regulatory compliance and decommissioning option assessment should be given during the planning phase. Proper risk assessment should be carried out to screen all decommissioning

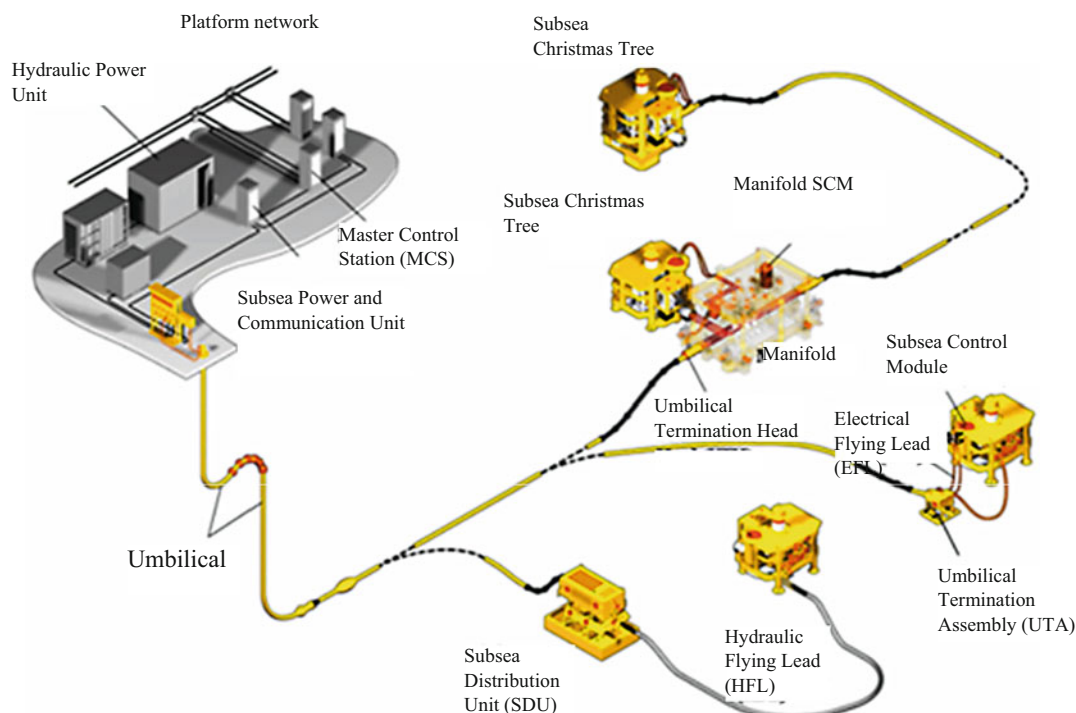


Decommissioning of Offshore Oil and Gas Installations, Fig. 5 Northwest Hutton Steel Jacket showing footings (skirt piles) top in place

options (options) to ensure the environmental implications of each are understood.

The environmental assessment method can vary, but in essence aims to identify the

environmental impacts of different options and enable a comparative assessment to be made to identify the most practically achievable option with the least environmental impact or



Decommissioning of Offshore Oil and Gas Installations, Fig. 6 Typical subsea infrastructure layout

maximize environmental benefit. Typically, once a specific option is agreed or as part of the assessment of options, a specific Environmental, Social, and Health Impact Assessment (ESHIA) will be required to support regulatory and company compliance. Relevant issues to consider typically include, but are not limited to:

- Physical impacts on the environment from removal of infrastructure
- Physical impacts or benefits on the environment from leaving infrastructure in place
- Potential emissions from platform operations and vessels during decommissioning
- Impacts from anchoring of vessels during decommissioning
- Potential disturbance to biodiversity or sensitive species (e.g., noise)
- Potential environmental impacts from residue or contamination of hazardous materials in installations
- Waste management and disposal, flush water handling, and potential for spills

- Biosecurity, accidentally transporting invasive species when moving between marine ecosystems
- Information, experiences, and base line data and from the operate phase of the asset to help scope and frame the ESHIA

More specific environmental risk management plans or procedures, such as waste management plans, biodiversity action plans, etc. may be required prior to implementation of decommissioning activities depending on the nature of the D&R project and the relevant regulatory regime; this will vary by project.

Social Performance

Some local stakeholders have lived with the local oil and gas operation from its construction and it is reasonable for them to expect to be engaged to understand the impact of decommissioning and get assurance that their needs should be considered. Other stakeholders, e.g., nongovernment

organizations with a stake in the process or the outcome need to be engaged so that their views can be heard. The social performance considerations include, but are not limited to, impacts of:

- Right of way or exclusion zones on livelihoods and fishery
- Residual impacts and safety risks from offshore installations decommissioned in place, e.g., impact on sea life and environment
- Opportunities from alternative use or reuse, such as local reefing, aquaculture, tourism, coastal protection, and local industrial area (onshore)
- Sustaining job opportunities, social investment, and local business development

An example where the need for engagement with a key stakeholder was not recognized in time and this had a decisive influence on the course of decommissioning is the Brent Spar. In 1995, Shell was embroiled in a public dispute over the decommissioning and disposal of the Brent Spar, a redundant oil storage installation in the North Sea. Brent Spar was a floating oil storage facility constructed in 1976 and moored approximately 2 km from the Brent Alpha platform. After evaluating a number of options for disposing of the spar, approval was sought by Shell/Exxon for deep sea disposal from the UK Government. Greenpeace became aware of the plan to sink the Brent Spar; the organization had been actively campaigning against ocean dumping in the North Sea since the early 1980s and lobbying for a comprehensive ban on ocean dumping through the OSPAR convention. Objections culminated in Greenpeace activists occupying the Brent Spar, an energetic international media campaign that ultimately influenced political and public opinion against Shell's preferred option. Brent Spar was internationally damaging to Shell's reputation: Despite the support of independent scientists for Shell's proposals, Shell did not win public acceptance. Use of emotional language against sea dumping and use of false information (by Greenpeace) regarding potential damage to the environment turned public opinion against Shell's decision for deep sea disposal of the Brent spar, towards a solution of onshore disposal

that is more damaging to the environment and much higher cost.

Industry Experience in Decommissioning

Industry experience is that the large concrete installations constructed over the period 1970–1985 are extremely difficult to be removed. They cannot be refloated safely and they cannot realistically be cut into pieces that can be liftable. This is recognized in OSPAR Decision 98/3. In Norway the government has granted exemptions from Decision 98/3 for two installations to be left in place. These are the Ekofisk tank and its protective concrete barrier and the concrete substructure of the Frigg TPC2 module.

On the other hand, experience with steel installations is that both substructure and topsides can be removed, brought onshore, and dismantled. The only exception is that the bottom part of very large steel jackets (notably NW Hutton and Brent Alpha in Northern North Sea), which includes large-diameter legs with grouted skirt piles, is very difficult to remove and some derogations have been granted to leave these in place. Recycling of structural steel rather than scrapping is the norm. In Norway alone the volume of steel recycling from decommissioning is about 80,000 tonnes of steel per year. Experience to date is that about 97% of recovered steel is recycled.

Overall the response of the industry to the challenges of decommissioning has been very positive with some significant improvements in capability (e.g., Pioneering Spirit, improvements in cutting technology). The experience accumulated over the last decade has been very valuable and would suggest that the offshore oil and gas industry will be successful in meeting the main objective of decommissioning which is to leave behind a clean seabed and clean environment.

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Decommissioning of Subsea Facilities

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Definition

Decommissioning is a general term for a formal process to remove something from an active status. Decommissioning of subsea facilities refers to the safe abandonment of subsea production facilities and associated infrastructure (such as well-heads, pipelines, umbilicals, and so on) that are no longer operational.

Scientific Fundamental

Decommissioning Importance of Subsea Facilities

Decommissioning is a normal part of the offshore oil and gas lifecycle. Decommissioning of subsea

facilities is an action taken to fully consider the safety and environmental protection of workers after the expiration of the use of subsea facilities or for other reasons.

Subsea facilities are playing a very important role in the creation of cost-efficient field developments. As a means to tackle both new and existing exploration and production challenges, oil and gas projects commonly involve a subsea concept. Christmas tree, subsea oil gathering manifold, and underwater manifold center are the main facilities of the subsea production system. A range of technical decommissioning and disposal options exists for offshore facilities, each of which must be evaluated on a case-by-case basis. The scope of consideration should include a broad range of aspects and not only focus on the vicinity of the structure (Ibrahim Khan and Islam 2007; Kaiser 2017).

Decommissioning Options

Decommissioning options can include complete or partial removal, alternate use as well as leaving the infrastructure in situ (in place). The selected decommissioning option will consider all safety, social, environmental, technical, and economic aspects to provide the best balance of these factors. Decommissioning options will vary from facility to facility. Each facility and its associated infrastructure are made up of many components that require different methods to be considered.

- Complete removal, disposal onshore, or offshore
- Partial removal, disposal onshore, or offshore
- Abandonment in situ
- Alternate use, either within the oil and gas industry or other industries/purposes

Basic Work Before Decommissioning of Subsea Facilities

During decommissioning, some, if not all, of the subsea system will be removed and carried to shore for disposal. It is likely that the majority of the pipeline is recycled. There may be some residual pipeline contents that could not be removed through the flushing and cleaning procedures, including wax, scale, oily sludge, or NORMS (naturally occurring radioactive materials).

During project design, engineers should establish the methodology for both the bulk removal of hydrocarbons and cleaning the pipelines to the proposed cleanliness standards. The bulk removal procedure, possibly completed in parallel with the precommissioning procedure, would require calculated pigging and draining routes to transport the bulk hydrocarbons to shore for processing, whether that is through a trunk line or tankers. During the design phase, cleanliness verification procedures must be assessed, for example, sampling pipeline contents at topside until a specified cleanliness is reached or by removal of a section of the subsea pipeline for inner wall inspection.

Key Applications

Wellheads

Production wells that can no longer be used must be plugged to prevent the oil and gas reservoir fluids from migrating up hole over time and possibly contaminating other formations, fresh water aquifers, or the marine environment. A well is plugged by setting mechanical or cement plugs in the wellbore at specific lengths and intervals to prevent fluid flow. The main rigless plugging process usually requires a spread containing the following: pump, cement blender, a combination wireline/electric line unit, mud tank, and commonly used materials such as cement, bridge plugs, and cutting equipment to conduct the operation. The plugging process can take from 7 to 10 days, depending on the number of wells on the structure, any issues in wellbore, and the service provider (Vrålstad et al. 2019). The well plugging and abandonment techniques are listed in Table 1.

Rigid Pipelines

Rigid pipelines, if not buried or trenched, may have to be removed after Cessation of Production. Options for flushing and cleaning, making the pipeline safe and abandoning in situ, removal of structures and mattresses, etc. are established on a case-by-case decision and are usually based on engineering justifications and approval by the regulator. Generally, rigid pipelines are difficult to remove, especially the larger diameter

pipelines. Once a pipeline is buried beyond the mobile sediment layer, a minimum depth of approximately 0.6 m, it will likely remain buried after decommissioning. A recommendation is to bury the pipeline during installation to avoid costly removal in the future (Rouse et al. 2018). Pipeline abandonment techniques are listed in Table 2.

Umbilicals

Umbilicals are required to be removed unless they are buried and are proven that they will remain buried post-decommissioning. Decommissioning umbilicals creates problems that are similar to flexible flowlines, including flushing and cleaning, removal, and integrity checks. Hydraulic fluid and chemicals inside umbilicals must be cleaned out, with more stringent cleaning standards than trace hydrocarbons in pipelines. There may be issues with umbilical recovery, and an integrity assessment is recommended to determine the umbilical integrity at the end of life. The removal method and whether potential reuse is possible will depend on the integrity. Umbilical decommissioning techniques are listed in Table 3.

Pipeline Bundles

Experience on removal of pipeline bundles is limited. Similar issues need to be considered as for rigid pipelines – cleaning of each pipeline, removal procedure, can justification be made to leave in place, reuse potential, etc.

If a bundle consists of an umbilical and a pipeline, the bundle is generally considered one unit. However, if the pipeline is unburied and is to remain, a further assessment is recommended on the stability of the straps. If the straps are expected to corrode, the straps may have to be cut, and the umbilical removed to prevent any snagging hazards.

Flexible Flowlines

Flexible flowlines, due to their size and length, are typically recovered by re-reeling after the life of the field. Flushing and cleaning flexibles are challenging especially if they are not designed with a smooth bore. Smooth bore flexibles are easier for hydrocarbon removal during decommissioning

Decommissioning of Subsea Facilities, Table 1 Well plugging and abandonment techniques (ICF 2016)

Facility type	Characteristics	Abandonment technique
Fixed platform wells	Problem free wells on platform that can support P&A spread	Use a workboat and rigless abandonment
		If no platform crane or insufficient capacity, include a portable crane or casing jacks. Drilling rig, if present on the platform, is used in place of jacks
		Rig-up, perform diagnostics, and P&A the wells
Fixed platform wells	Problem free wells on platform that cannot support P&A spread	Use a liftboat and rigless abandonment
		If no platform crane or insufficient capacity, include a portable crane or casing jacks
		Rig-up, perform diagnostics and P&A the wells
Fixed platform wells	Problem wells on platform that can support P&A spread	Use a workboat and rigless abandonment
		Depending on well problem, the following spread(s) would be included
		Snubbing unit
		Coil tubing unit
		Hydraulic workover unit
		Drilling rig
		Milling tools
		If no platform crane or insufficient capacity, include a portable crane or casing jacks. Drilling rig, if present on the platform, is used in place of jacks
Fixed platform wells	Problem wells on platform that cannot support P&A spread	Rig-up, perform diagnostics, and P&A the wells
		Use a liftboat and rigless abandonment
		Depending on well problem, the following spread(s) would be included
		Snubbing unit
		Coil tubing unit
		Hydraulic workover unit
		Drilling rig
		Milling tools
Floating platform dry tree wells	Problem free wells	If no platform crane or insufficient capacity, include a portable crane or casing jacks. Drilling rig, if present on the platform, is used in place of jacks
		Rig-up, perform diagnostics, and P&A the wells
		Use a workboat, rigless abandonment, and casing jacks. Drilling rig, if present on the platform, is used in place of jacks
Floating platform dry tree wells	Problem wells	Depending on well problem, the following spread(s) would be included
		Snubbing unit
		Coil tubing unit
		Hydraulic workover unit
		Drilling rig
		Milling tools
		Rig-up, perform diagnostics, and P&A the wells
Subsea wells	Water depth <800 m?	Selection of work and vessel spreads depends on condition of well and water depth
	Non problem well or problem well?	Rig-up, perform diagnostics, and P&A the wells

(continued)

Decommissioning of Subsea Facilities, Table 1 (continued)

Facility type	Characteristics	Abandonment technique
Subsea wells	Divers or ROV?	Complete removal 15' bellow mud line
	Can well be plugged using rigless methods?	Remove tree and wellhead, unless waiver to leave wellhead is obtained
	Water depth >800 m	Selection of work and vessel spreads depends on condition of well and water depth
	Non problem well or problem well	Rig-up, perform diagnostics, and P&A the wells
	Divers and or ROV	Remove tree, obtain waiver to leave wellhead
	Can well be plugged using rigless methods?	

Decommissioning of Subsea Facilities, Table 2 Pipeline abandonment techniques (ICF 2016)

Facility type	Characteristics	Abandonment technique
Riser to riser	Diameter, length, product	Selection of work and vessel spreads depends on type of pipeline, whether fluids are flushed, depth of cover over the pipeline, and water depth Flushed clean, cut ends, remove a section from each end, plug, and bury Flush clean and remove pipeline in sections Flush clean and remove pipeline by reverse reeling
Riser to SSTI	Flushed fluids can or cannot be pumped down hole, through the pipeline or filtering equipment is or is not present on the receiving end	
SSTI to SSTI		

Decommissioning of Subsea Facilities, Table 3 Umbilical decommissioning techniques (ICF 2016)

Facility type	Characteristics	Abandonment technique
ALL	Diameter, length	Selection of work and vessel spreads depends on length, size, and purpose of umbilical and water depth
	Purpose – electrical or other	Flush clean non-electrical umbilicals
		Abandonment in place – the ends cut, plugged, and buried. Umbilicals are to be removed unless waiver is obtained to abandon in place
		Abandonment by removal – the ends cut, and umbilical is removed

than rough bore as hydrocarbons become trapped in the flowlines' rough inner wall and are unable to be cleaned. Besides, a rough bore, upon reeling, may spill trace hydrocarbons into the ocean and cause for a difficult onshore disposal for the trace hydrocarbons that remain trapped in the flexible flowlines' inner walls.

Stability Units/Concrete Mattresses

Mattresses are used for stability and protection on pipelines, valves, and subsea structures. They

withstand severe storms, and they are required to be removed during decommissioning if the structural integrity allows. Generally, any stability units that are placed on the seabed are to be removed during decommissioning. An additional complexity for underwater lifts is exacerbated by the uncertainty in weight and center of gravity of the unit/mattress. During design, heavier and stronger materials may be included to withstand more severe storms to ensure the mattresses remain intact for removal for decommissioning.

Subsea Valves

Valves and tees are useful pipeline components during decommissioning. Strategically placed valves could be used as an isolation point to remove a subsea structure while leaving the pipeline in place. For example, a subsea manifold with an elevation of 5 m on the seabed must be removed, while the buried flowlines connecting it to the riser base remained in situ. A subsea isolation valve was located 2 m away from the subsea manifold. Therefore, the logical cut point was directly upstream of the isolation valve and provided an end cap for the pipeline remaining in situ. After many years of use, the seabed valves may not provide a seal for decommissioning. Therefore, timely inspections and maintenance are important throughout the life cycle so the valve can maintain a seal at the end of the design life. The valve may require a design life for beyond the life of the field to ensure adequate performance after the Cessation of Production during decommissioning.

During decommissioning, valves should be ROV operated for ease. In many cases, operators choose diverless decommissioning methods to minimize the personnel safety risk. It is recommended that all valves are designed to be ROV operated to allow for a diverless decommissioning option, if applicable. The determining factor with diver assisted vs. ROV tasks is the access around valves and subsea structures. Debris removal by jetting, dredging, or mass flow excavation creates extremely low visibility and hinders the ROV's workability.

Manifolds and Other Subsea Structures

Subsea structures, including manifolds, jumper, and subsea tree, will require removal during decommissioning. During removal, actual weight of the structure will be unknown. Corrosion will decrease the unit weight by consumption of the sacrificial anodes, while spillage of grout, marine growth, and particle settling on the structure will increase it. The original pad eyes will remain on the structure, and ideally, they will be suitable to lift the structure with a crane. Break out loads can double the underwater weight of the manifolds. For ease in lifting, bigger and stronger pad eyes on

Decommissioning of Subsea Facilities, Table 4 Subsea structure decommissioning techniques (ICF 2016)

Facility type	Characteristics required	Abandonment technique
Manifolds	Lift weights and dimensions	Complete removal
	Water depth	Abandon in-place
PLET	Lift weights and dimensions	Complete removal
	Water depth	Abandon in-place
Jumper	Lift weights and dimensions	Complete removal
	Water depth	Abandon in-place
Subsea tree	Lift weights and dimensions	Complete removal
	Water depth	Abandon in-place
SUTA	Lift weights and dimensions	Complete removal
	Water depth	Abandon in-place

the template should be used to allow for extra weight uncertainty during removal (Table 4).

Cross-References

- [Complete and Partial Dismantling](#)
- [Decommissioning of Fixed Platform](#)
- [Decommissioning of Floating Platforms](#)

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Deep Sea Mining

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Synonyms

Deep seabed mining; Ocean mining

Definition

Deep sea mining is defined as the process of seabed mineral excavation, transmission, lifting from the deep to the surface, and extracting valuable metal elements or products from the minerals. The definition of deep seabed is usually distinct in different disciplines. Generally, the depth of deep seabed mining is referred to much deeper than 1000 m. Most of the seabed area deeper than 1000 m is located in the international seabed area which is under the regime of the International Seabed Authority (hereinafter referred to as ISA). In accordance with the regulations of ISA, any state or entity can submit and apply an exploration contract sponsored by the state or states (China Ocean Mineral Resources R&D Association 2015).

Scientific Fundamentals

Deep Seabed Mineral Resources

The amounts of deep seabed mineral resources are huge and usually include mineral resources such as polymetallic nodules, cobalt-rich crusts, polymetallic sulfides, deep seabed rare earths, and so on.

Deep seabed polymetallic nodules are usually found on the deep sea basins from 4,000 to 6,000 m. Due to the high manganese content, it is also commonly referred to as manganese nodules. In addition, polymetallic nodules are also rich in other elements such as copper, cobalt, nickel, and molybdenum. The rich-identified

polymetallic nodules in the international seabed (hereinafter referred to as the AREA) are located in the Clarion-Clipperton Fracture Zone in the Pacific Ocean. The estimated global resource for polymetallic nodules in the seabed is about 70 billion tons.

Cobalt-rich crusts means cobalt-rich iron/manganese (ferromanganese) hydroxide/oxide deposits formed from direct precipitation of minerals from seawater onto hard substrates containing minor but significant concentrations of cobalt, titanium, nickel, platinum, molybdenum, tellurium, and other metallic and rare earth elements. The cobalt-rich crust exploration contract areas are mainly located on the seamounts in the northwest Pacific Ocean. The estimated global resource for cobalt-rich crusts in the seabed is about 21 billion tons.

Polymetallic sulfides means hydrothermally formed deposits of sulfides and accompanying mineral resources in the seabed which contain concentrations of metals including, inter alia, copper, lead, zinc, gold, and silver. The contract areas of polymetallic sulfides in the AREA are located along the ridges of oceans, just like the Atlantic Ocean and India Ocean. The estimated global resource for polymetallic sulfides in the seabed is about 0.4 billion tons.

Deep Seabed Mining Technology (Hoagland et al. 2010; Wang 2015)

Broadly speaking, deep seabed mining technology includes submarine mineral excavation (cutting and/or collection), transmission and lifting, monitoring and control, surface support, mineral transportation, and subsequent mineral processing and metallurgy. In the narrow sense, deep seabed mining only involves the process of excavating seabed minerals to the surface ship.

With the development of deep sea technology and the gradual improvement of the understanding through deep sea mineral resource exploration, there have been successively proposed three prototypes of deep seabed mining, just as continuous linking buckets, pipeline lifting, and shuttle submarine type. Considering the factors such as mining scale and technical economy, it is

generally believed that pipeline lifting is the most promising type.

The whole deep seabed mining system is composed of the submarine (cutting) collection sub-system, the pipeline transmission and lifting sub-system, the monitoring and control sub-system, and the surface support sub-system.

The submarine (cutting) collection sub-system mainly removes and collects the seabed minerals into the mining vehicle by means of cutting, water jet, etc., and then crushes the collected minerals to adapt to subsequent pipeline transmission and lifting.

The pipeline transmission and lifting sub-system mainly uses the pumps to transport the minerals collected by the mining vehicle to the surface support vessel.

The monitoring and control sub-system mainly realizes the collection and comprehensive processing of positioning, navigation, and various operation information of the entire mining process, from underwater to surface, and realizes the smooth control of mining operation.

The surface support sub-system, also known as the deep seabed mining vessel, is the support platform for the whole mining operation and is the living platform for mining operators. The main functions include the launching and recovering of the underwater mining equipment, deep seabed mining operations, mineral onboard storage and transshipment, living facilities providing, etc.

Deep Seabed Mining Stages

Deep seabed mining activities usually include three stages: exploration, mining, and closure. The environment is an important factor in the mining process and is considered in parallel with all phases of mining.

Each stage of mining activities has different tasks. The exploration stage mainly includes prospecting, general exploration, detailed exploration, mining and metallurgical preparation, as well as environmental baseline survey and environmental impact assessment, and finally determining the mining areas. The focus is to get the inferred, indicated, measured resources and the probable, proved reserves according with the different exploration stages.

The mining phase mainly includes feasibility studies, mining system construction, mining operation, and environmental monitoring and assessment. The results of the feasibility study determine whether commercial mining is possible. If feasible, the mining system construction and specific mining work could be started. Feasibility studies include not only commercial and technical but also the feasibility of environmental monitoring and assessment. Mining may be terminated due to environmental issues.

If all goes well, it will enter the closure stage after the end of the mining. The closure stage is mainly the ecological rehabilitation, restoration, and evaluation.

Deep sea mining activities will be subject to the ISA Mining Code, which include regulations on prospecting and exploration for different resources and exploitation regulation in the process of formulation in accordance with the United Nations Convention on the Law of the Sea. Mining activities including the exploration and exploitation within the exclusive economic zone of a coastal State are subject to the domestic mining regulations of the coastal State. Mining activities in the international seabed area are not only subject to the exploitation regulations of the ISA but also to the relevant other international conventions and laws of the country in which the mining companies are registered.

Deep Seabed Mining Progress (Li 2017)

Up to 2018, the deep seabed mining activities generally are still in the exploration stage in the world. The ISA has formulated three regulations for the prospecting and exploration of polymetallic nodules, polymetallic sulfides, and cobalt-rich crusts in the AREA. The exploitation regulation of deep seabed mineral resources is still under drafting. Some coastal countries have issued the prospecting, exploration, and exploitation regulations for deep seabed mineral resources in the exclusive economic zone. At present, the ISA has approved 29 international seabed exploration contracts, including 17 exploration contracts for polymetallic nodules, 7 contracts for exploration of polymetallic sulfides, and 5 contracts for exploration of cobalt-rich crusts.

Although the first 15-year exploration contracts have finished for polymetallic nodules in 2016, there are still no contractors to start commercial exploitation. But many contractors have carried out different scales of deep seabed mining test including the USA, European Union, China, Japan, Korea, and some international companies. The mining activities in the exclusive economic zone go faster in comparison with the international seabed.

Commercial deep seabed mining has not yet become a reality mainly from the following challenges. First, the deep seabed mining must have a more competitive advantage before the similar ground resources become scarce and not only must be profitable but also have more comparatively advantages in cost, environmental impact, and so on. Second, it must be technically feasible, especially for commercial-scale mining technologies. Current existing mining technologies still require long-term testing during the mining operations. Third, the awareness from the environment is still insufficient. Too strict environmental regulations will make deep seabed mining work impossible. Certainly, if there is a necessary mineral only existing in the deep seabed, commercial deep seabed mining would soon be achieved, but this mineral has not yet been discovered.

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Deep Seabed Mining

► Deep Sea Mining

Deep Submergence Rescue Vehicle (DSRV)

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Synonyms

Submarine rescue vehicle (SRV)

Definition

A deep submergence rescue vehicle (DSRV) is a type of deep-submergence vehicle used for rescue of downed submarines and clandestine missions. While DSRV is the term most often used by the United States Navy, other nations prefer to call it submarine rescue vehicle (SRV). SRV is one type of submarine rescue submersible, which can carry submariners from a disable submarine to the surface. The function of submarine rescue vehicle is the same as submarine rescue bell. The typical difference between them is that the rescue bell is a bell shape submarine rescue system, the submarine rescue vehicle is a free move manned submersible. As a special manned occupied submersible, it consists of four major components: manned pressure hull, propellers system, observational system, and rescue system.

Scientific Fundamentals

Historical Development

Before 1970s, the McCann Rescue Chamber System remains in service in several contemporary navies, including the USN and the Turkish Navy. Submarine rescue philosophies evolved further in the 1960s following the loss of two American nuclear-powered submarines, US Ships Thresher, and Scorpion, despite both boats being lost in waters that precluded escape or rescue. After considering a variety of options, including

submarines with in-built escape pods and submarines with front ends that could be blown to the surface, the USN developed the deep submergence rescue vehicle (DSRV). Entering service during the 1970s the DSRV, a manned mini-sub that mates with a DISSUB's hatch and could carry 24 people at a time, offered great flexibility. With two built, one is maintained in an operational state so it can be flown in a C-5 cargo plane to a port nearest the DISSUB. It can then be placed onboard either a modified US or allied submarine. Operating from a submarine means that rough surface conditions or ice is less likely to adversely affect rescue operations (Fig. 1).

The deep submergence rescue vehicle (DSRV) was replaced with the submarine rescue and diving recompression system (SRDRS). The submarine rescue diving and recompression system (SRDRS) will be the Navy's twenty-first century submarine rescue system.

SRDRS will provide a new capability of pressurized transportation of rescuees from a stricken submarine directly to the decompression system eliminating the requirement for deep submergence rescue vehicles, mother submarines, and submarine rescue chambers. SRDRS is to include an air transportable rapid assessment/underwater work system, a decompression chamber system, and a pressurized rescue module. The SRDRS

will provide a global rapid response capability to support submarine rescue missions with an increase in capability at a fraction of the cost of the currently available systems.

The submarine rescue diving and recompression system (SRDRS) consists of four distinct elements, which include the assessment/underwater work system (AUWS); submarine decompression system (SDS); and the pressurized rescue module system (PRMS); and PRMS mission support equipment, including the launch and recovery system (LARS) a deck mounted A-frame crane used to launch and recover the PRM. The SDS, PRM, LARS, and associated generators and auxiliaries all compose the submarine rescue system (SRS). The submarine rescue diving and recompression system (SRDRS) consists of the submarine rescue system (SRS) in conjunction with the assessment/underwater work system (AUWS). These systems are designed to rapidly deploy to any location in the world via air and ground transportation and will be installed on a military or commercial vessel of opportunity (VOO). The SRS, approximately 183 tons, is installed aboard the VOO (Fig. 2).

As the world's leading submarine rescue service provider, James Fisher Defence and Divex (JFD) offers a world-class submarine rescue system capability, vital to the safety of submariners in the global defense industry.



Deep Submergence Rescue Vehicle (DSRV), Fig. 1 The US DSRV

**Deep Submergence
Rescue Vehicle (DSRV),
Fig. 2** The US PRMS



Deep Submergence Rescue Vehicle (DSRV), Fig. 3 LR5

The Royal Navy's LR5 submarine rescue vehicle (SRV) is similar to the DSRV in most aspects but instead of using a modified vessel the LR5 uses a ship of opportunity as the Mother Ship. The LR5 was replaced by the NATO submarine rescue service (NSRS), a system developed jointly by Britain, France, and Norway, while the USN is developing the submarine rescue diving and recompression system (SRDRS). Both systems are similar and will carry out rescue operations in three phases; reconnaissance, rescue, and crew decompression.

The Korean and Singapore navy also have LR5 SRV, manufactured by JFD (Fig. 3).

Russian deep submergence rescue vehicle AS-28 is a Priz-class deep-submergence rescue vehicle of the Russian Navy, which entered service in 1986. It was designed for submarine rescue operations by the Lazurit Design Bureau

in Nizhny Novgorod. It is 13.5 m (44 ft) long, 5.7 m (19 ft) high, and can operate up to a depth of 1,000 m (3,300 ft) (Fig. 4).

China began to research deep submergence rescue vehicle from 1971, named 7103 SRV. At that time, it was jointly developed by Harbin Engineering University, Shanghai Jiao Tong University, Huazhong University of Science and Technology, and Wuchang Shipyard. The boat was started construction at the Wuchang Shipyard in 1976 and was successfully tested in 1986. The boat is 14.88 m long, 2.6 m wide, 4 m high, with a displacement of about 30 tons and a speed of 4 knots. The working depth of 7103 SRV is about 300 m, and 22 crew members can be rescued at once time (Fig. 5).

In 2009, the Chinese Navy purchased the LR7 SRV from Perry Slinsby Systems Co. LR7 is an upgraded product from LR5 SRV. The LR7 SRV

Deep Submergence Rescue Vehicle (DSRV),

Fig. 4 Russian deep submergence rescue vehicle



Deep Submergence Rescue Vehicle (DSRV), Fig. 5 China 7103 SRV

has a total length of 25 ft (about 7.6 m) and a maximum working depth of 500 m. There are two pressurized manned cabin. The front cabin is the operation cabin; the rear cabin is rescue cabin. The LR7 SRV can carry out 18 rescuer at one time.

Compared with the US PRMS, the LR7 docking skirt cannot be rotated. LR7 SRV has a large angle adjustment ability by its angle adjustment system. LR7 SRV has a large spherical crown observation window, which gives the pilot a large visual range.

LR7 SRV was powered by zebra batteries. China has replaced the zebra batteries by lithium-ion batteries (Fig. 6).

Key Technology in the Development of a Deep Submergence Rescue Vehicle

The application of deep submergence rescue vehicle is a high-tech system engineering. In order to ensure the timely and effective completion of rescue operations, it is necessary to set up a strict operating procedure. How to travel to the disable submarine point and reach the crashed sea area in short time is not easy.

The main difficulties include: (1) The efficiency alarm system. The golden rescue time is only 72 h. (2) The internal environmental conditions of the disable submarine. The SRV must reach the disable submarine before the emergency



Deep Submergence Rescue Vehicle (DSRV), Fig. 6 China LR7

life support supplies are exhausted, or the toxic gas leaks in the submarine. (3) Efficient operating procedures. (4) The navigation ability of mother ship and SRV. It mainly includes the ability to search for the specific location of the disabled submarine and the position of the lifesaving platform. (5) Docking capabilities.

Technically, several aspects need to be considered. (1) The disabled submarine emergency life support system must be reliable and lasting. (2) The notification system should be established. (3) Improve operational efficiency and minimize the time for carrying and searching targets. (4) The docking process should be simple and reliable.

In the rescue operation, how to solve docking process is a big problem. How to adjust the docking angle of the SRV according to the disabled submarine's lifesaving platform position. How to observe the lifesaving platform when the visibility is zero and the current is very urgent. The movement of people will change the center of gravity of the entire SRV system and how to maintain the center of gravity system, how to evaluate the strength, and stability of a rotary skirt (if it is used). The interior space of the docking skirt is limited. It is not only used for

accommodating the wrecked submarine crew, but also used for installing the winch, pump, air sampling device, etc. The structure of the transfer skirt may be more complicated. How to achieve optimized design results in finite space to obtain the maximum available space. The maneuverability, terminal navigation ability, and docking skirt structure form determine the success of docking and rescuing. The two roles complement each other. The early deep submarine lifeboats required high handling performance due to the use of fixed docking skirts.

Development trend: The main components use the market products as much as possible. It is not only more reliable, but also can reduce the cost. The functions of SRV should be diversified and not limited to military use. SRV devices have special characteristics; they can also draw on the advantages of other submersible vehicles. With the technological advances in microelectronics, high speed digital computers, artificial intelligence, guidance and control hardware and software, automatic docking technology will gradually improve. The overall performance of SRV is expected to be greatly improved, and it can be developed in the direction of self-propelled manned submersible with cables. It is possible to

obtain reliable evaluation results through digital simulation and optimize the operating procedures through simulation to reduce the cost.

Cross-References

- [Human Occupied Vehicle \(HOV\)](#)
- [Submersible](#)

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Deep Tow System

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Synonyms

Acoustic deep tow system; Acoustic towed system; Marine Controlled-Source Electromagnetic Method (MCSEM); Tow system; Tow-body; Towed vehicle; Tow-fish

Definition

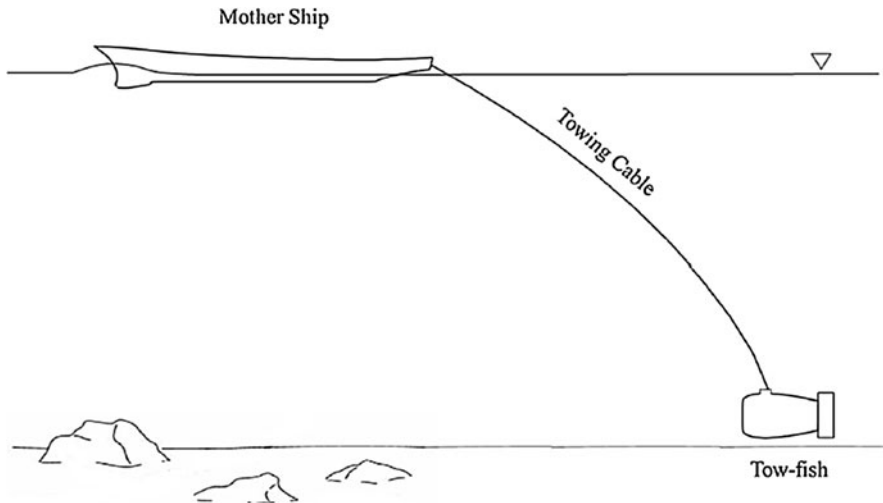
Deep tow system An underwater instrument system for comprehensive observation of deep-sea geology and geophysics, which is towed behind the mothership. Its main detection equipment is an underwater detection towed vehicle with depth and attitude adjustment ability. The equipment installed on the underwater towed vehicle includes a navigation and positioning system, side sonar, echo sounder, formation profiler, magnetometer, stereo camera, continuous temperature measurement device, sediment or biological sampler, and other detection instruments.

Scientific Fundamentals

The deep tow system is an undersea survey apparatus that is extensively used for ocean observation and ocean research. The system can perform high-resolution real-time online detection of seabed topography, shallow strata profiles, and physical and chemical parameters of seawater near the seabed. The deep tow system usually consists of an onboard control system, a towing cable, and a towed vehicle. Depending on different exploration purposes, different kinds of undersea detective sensing instruments could be installed in the towed vehicle. The operation of the deep tow system generally requires that the towed vehicle should have the ability to adjust its trajectory rapidly and stably in order to accomplish various kinds of high-speed mobility survey tasks. It can work continuously for a long time without energy limitation during operation. It is particularly suitable for long measuring line operation. However, the application of the deep tow system is a high-risk job, and the problems and challenges have been accompanied by the development of the acoustic deep tow system.

General Composition of Deep Tow System

The diagram of the underwater deep tow system is shown in Fig.1. The mothership is the loading



Deep Tow System, Fig. 1 Diagram of general deep tow system (Go and Ahn 2019)

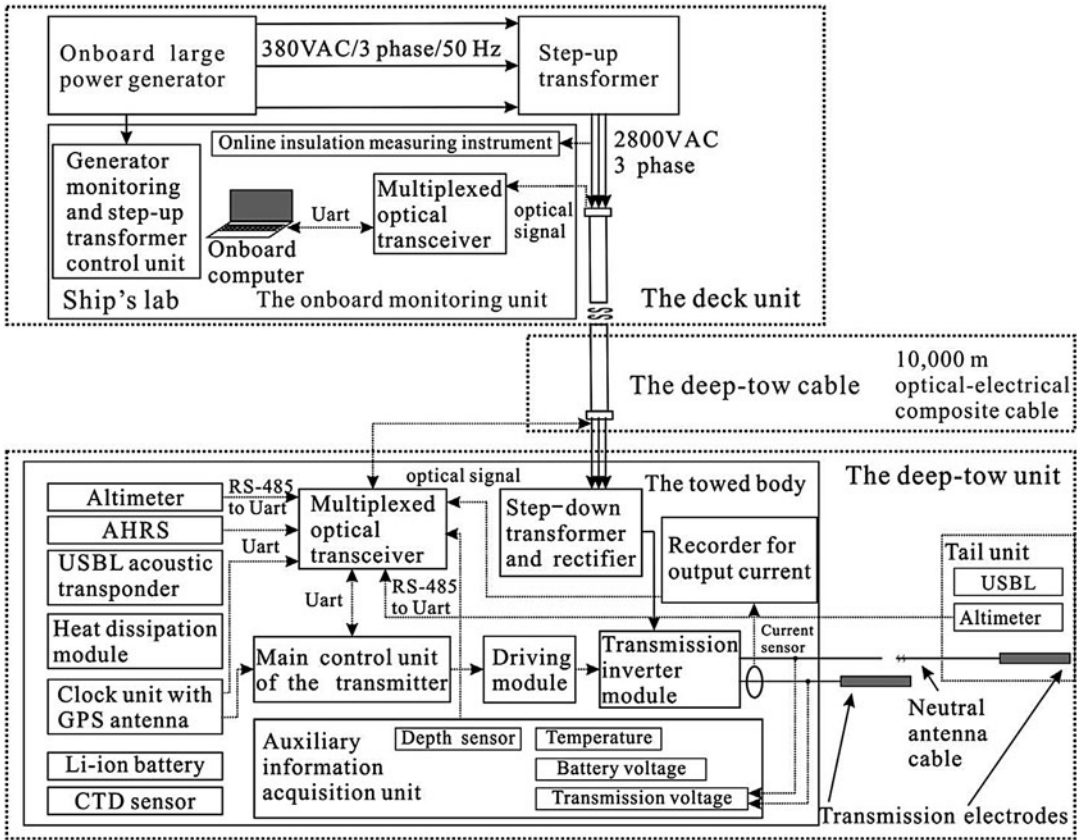
platform of the deep tow system. The tow system is carried by the mothership to the operating site. After the planned task is completed, the deep tow system will be retrieved onto the mothership and be carried to the next operating site. At the same time, the control subsystem of the deep tow system is installed on the deck or some cabins of the mothership. The control subsystem provides the required electric energy for the towed vehicle and records and analyzes the detection data sent back from the towed vehicle. The towing cable connects the mothership and the towed vehicle. Since the deep tow system itself does not have the active movement ability, it must be towed by the mothership to realize underwater movement. In addition, the function of the towing cable also includes power transmission and communication links. According to different working depths, the length of the towing cable can be adjusted. In shallow water, it is about tens of meters. However, in the deep sea, the length will exceed 10,000 m. The towed vehicle is the core part of the deep tow system. The underwater detection function must be realized by the sensors installed in the towed vehicle. In order to provide a stable underwater attitude, the towed vehicle should have a certain adjustment ability.

Wang et al. (2017) present the Marine Controlled-Source Electromagnetic tow system,

as shown in Fig. 2. This tow system is divided into three parts, namely, the deck unit, the deep-tow cable, and deep-tow unit. Furthermore, the detailed design for each subsystem is given.

Tow Vehicle Control Method

The depth or trajectory of a towed vehicle can generally be controlled by changing its control surface deflection or by altering the length of the towing cable. Since the former control pattern has the ability of precise positioning and attitude control, it has widely been applied as a major control manner for underwater towed vehicles. Several typical types of this kind of underwater towed vehicle have been proposed. Among these, the towed vehicles named Flying Fish and Delta developed by the research group of Kuyshu University in Japan are quite representative (Koterayama et al. 1995; Nakamura et al. 2001). The trajectories of these towed vehicles are maneuvered by changing the deflection of control surfaces with certain control techniques, and underwater observation or sensing instruments are installed in a cylindrical fuselage. The depth of Flying Fish is mainly controlled by altering the deflection of the vehicle's principal horizontal control surface; while in the towed vehicle Delta



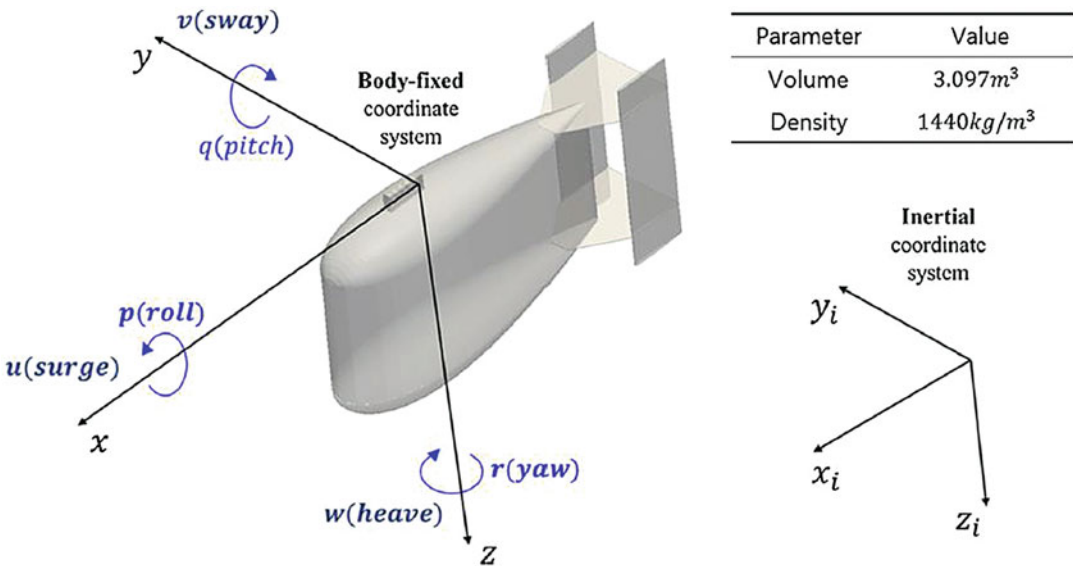
Deep Tow System, Fig. 2 Diagram of the Marine Controlled-Source Electromagnetic tow system (Wang et al. 2017)

the control surface is rigidly mounted, the depth of the towed vehicle is maneuvered by shifting a trim weight inside the vehicle to adjust the attack angle of its control surface. Furthermore, two thrusters are utilized at the stern of the Delta to generate a trim moment for a rapid depth change. Several other towed vehicles with a similar control way have also been proposed such as the towed vehicles Aurora, Towed torpedo, and Batfish as described in Buckham et al. (2003), Latorre and Mims (2000), and Dessureault (1976).

Analysis of Deep Tow System Motion

An accurate analysis of deep tow system motion is highly desirable for development and operational purposes. Reliable hydrodynamic force modeling is an essential part of an accurate prediction of tow

system motion. Kim and Choi (2007) and Zhang et al. (2010) start from a simplified version of hydrodynamic force model equations based on engineering intuition or empiricism. This kind of pre-simplification can accelerate the determination of hydrodynamic force modeling, but it could bring in an unexpected error in the expression of the nonlinear hydrodynamic model. Many studies have been aimed toward analyzing the cable and the tow-fish system (Chapman 1984; Grosenbaugh 2007). In these previous studies, the research effort was more focused on the global motion of the cable rather than on the detailed prediction of the tow-fish motion. The tow-fish was modeled either as a particle mass or as a simplified version of a rigid-body. Go and Ahn (2019) present a simulation-based methodology to determine hydrodynamic derivatives for the tow-fish, as shown in Fig. 3. Based on the actual



Deep Tow System, Fig. 3 Inertial and body-fixed coordinate systems of the tow-fish (Go and Ahn 2019)

operating conditions of the tow-fish, 32 simulation cases were set up and subsequently simulated using a CFD tool. A least-squares procedure was applied to determine the complete set of hydrodynamic damping coefficients. The forces and moments obtained from the CFD simulation were successfully reproduced by the second-order modulus expansion and the optimal hydrodynamic derivatives in the least-squares sense. The determined hydrodynamic derivatives were applied to the actual prediction of the tow-fish motion.

Design Method of Towed Vehicle

Towed vehicle design is the core part of the deep tow system development process. The key technologies of the towed vehicle design mainly include towing system design technology, acoustic detection technology, and system integration technology.

Tow System Design Technology. The tow system is a platform for various equipment installed on it. The design method of the tow system is also particularly critical in the deep-sea acoustic tow system. Moreove, the micro-

topography detection system also puts forward high requirements on the towed body. In summary, the technology contains the following points, towed vehicle hydrodynamic design technology, towed vehicle performance prediction technology, installation structure design technology, and deployment and recycling technology.

Micro-Topography Detection Technology. In the deep-sea acoustic tow system, micro-topography is the most important detection element of the system. The detection of micro-topography here mainly needs to obtain high-resolution topographic maps of the seabed, side-scan maps (seabed backscattered sound intensity maps), and shallow formation profiles. In summary, the technology includes the following points, micro-landform detection sonar design technology, sonar array design technology, and data fusion and post-processing technology

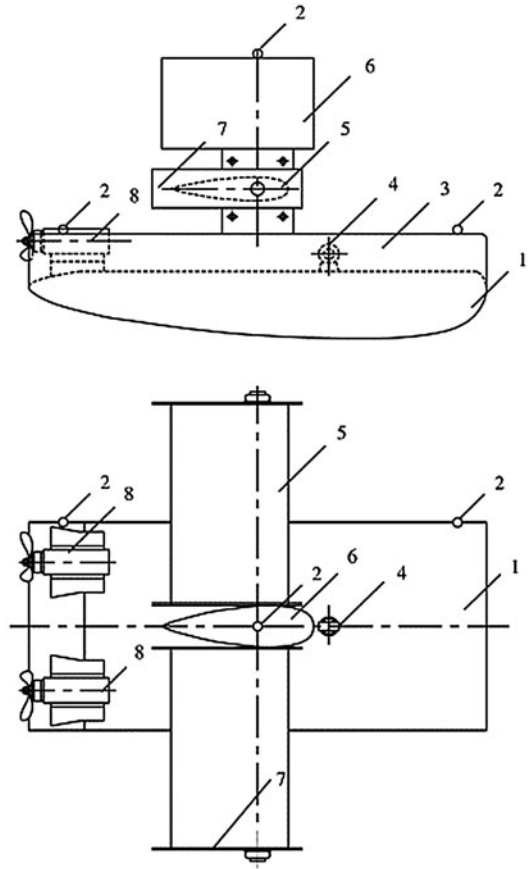
System Integration Technology. There is a variety of equipment installed on the towed vehicle. Optimizing the performance of the system, making the equipment work together, and reducing the mutual influence between the equipments is the main task of system integration. In summary, the technology includes the following points, technical index distribution technology,

power supply and communication technology, acoustic compatibility technology, towed body positioning technology, software system integration, and underwater safety assurance technology.

Key Applications

Wu et al. (2005) describe an alternative type of controllable underwater towed system. The towed vehicle in the system is composed of an adjustable depressing wing and a wing-shaped main body. The depressing wing serves as an actuator for the towed vehicle submerged depth control and the wing-shaped main body assists to produce an additional downward lift force to counteract the drag force on a towed cable and provides a large damping moment of pitching for the vehicle when it is towed at a certain velocity. It is expected that with such a design the objective of a rapid trajectory manipulation and a relatively stable attitude towing for the towed vehicle during a surveying operation can be obtained. The towed vehicle discussed in the research is a prototype intended to apply in field underwater survey towing (Fig. 4).

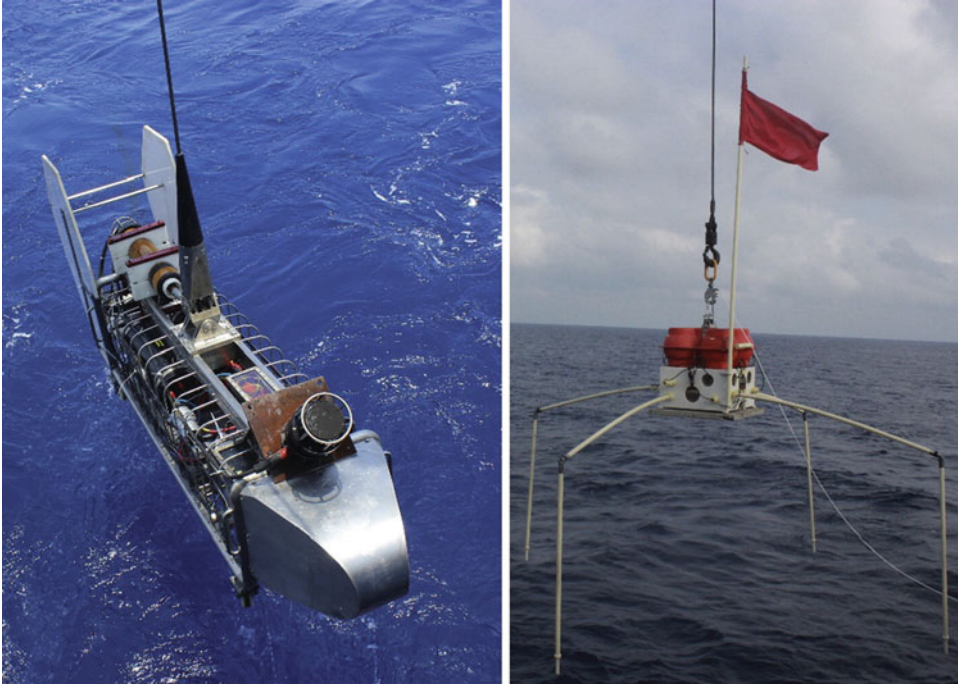
Considering the specifics of the deep-water environment, a deep tow system has been developed, named Marine Controlled-Source Electro-magnetic Method (MCSEM) transmitter system as shown in Fig. 5, which incorporates the high-voltage and long-distance power supply, high-speed bidirectional communication data transfer between the deck unit and the towed body, intelligent control, high-power current pulse inverter, and precise time synchronization between transmitter and receiver. They have described the “real-time control technology of hardware parallelism” of the MCSEM transmitter. A gas hydrate exploration test was carried out in 2014 to verify the performance of the transmitter system. It can be used for gas hydrate exploration as an effective source. The advantages of the tow system are mainly reflected in the stable reliability and real-time response. In the future development of the MCSEM transmitter system, the technical index of the system will be continuously



Deep Tow System, Fig. 4 Schematic drawing of the towed vehicle; 1 main body, 2 luminous diode, 3 end plate, 4 towing ring, 5 depressing wing, 6 buoyancy chamber, 7 wing end plate, 8 thruster. (Wu et al. 2005)

improved so that it can perform better in marine resource exploration.

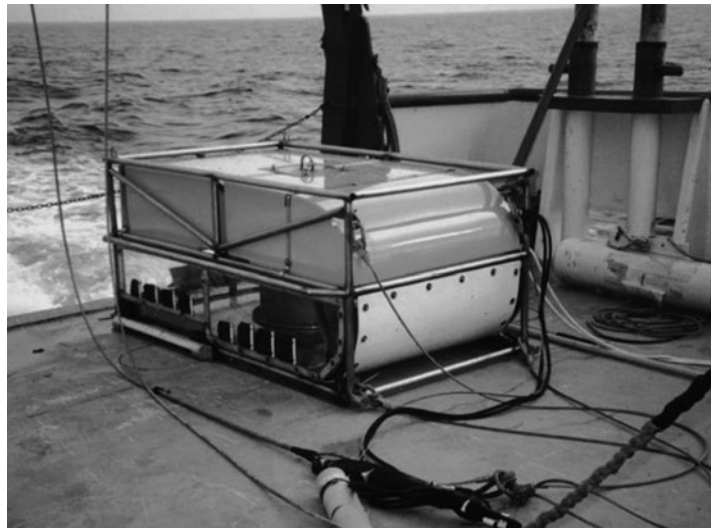
With the improvement of underwater transducers, there is a need for stable towing platforms for oceanographic measurements. The operational concerns result in a wide range of requirements. These performance requirements are often interconnected, which makes the design of a stable tow body a challenging work. Latorre and Mims (2000) describe the development of these performance requirements, along with the subsequent design, manufacture, and deployment of a 260 kg, $1.83 \times 1.06 \times 0.81$ m tow body. The project was completed in seven steps. With the completion of fabrication and flotation installation, the stable tow body was successfully towed



Deep Tow System, Fig. 5 Deployment of the deep-tow MCSEM transmitter (left) and the MCSEM receiver (right). The transmission current waveform and the

received signals were sampled at 150 Hz rate. The tail unit and transmission antenna were installed adjacent to the scheduled station. (Wang et al. 2017)

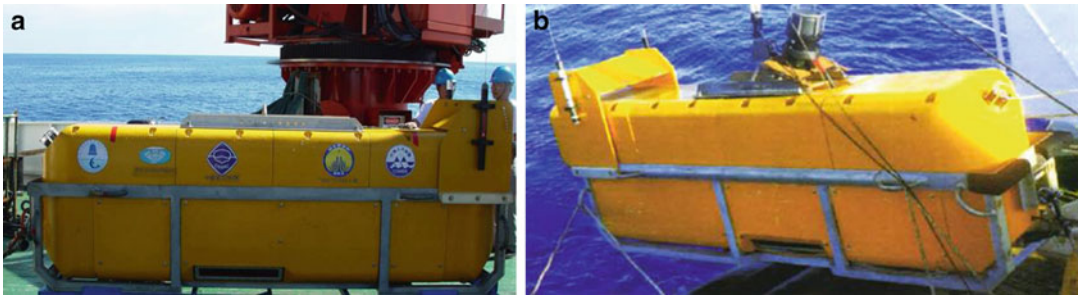
Deep Tow System, Fig. 6 Tow body photo of 5 September 1997 trials of tow body aboard M.V. Murno (Latorre and Mims 2000)



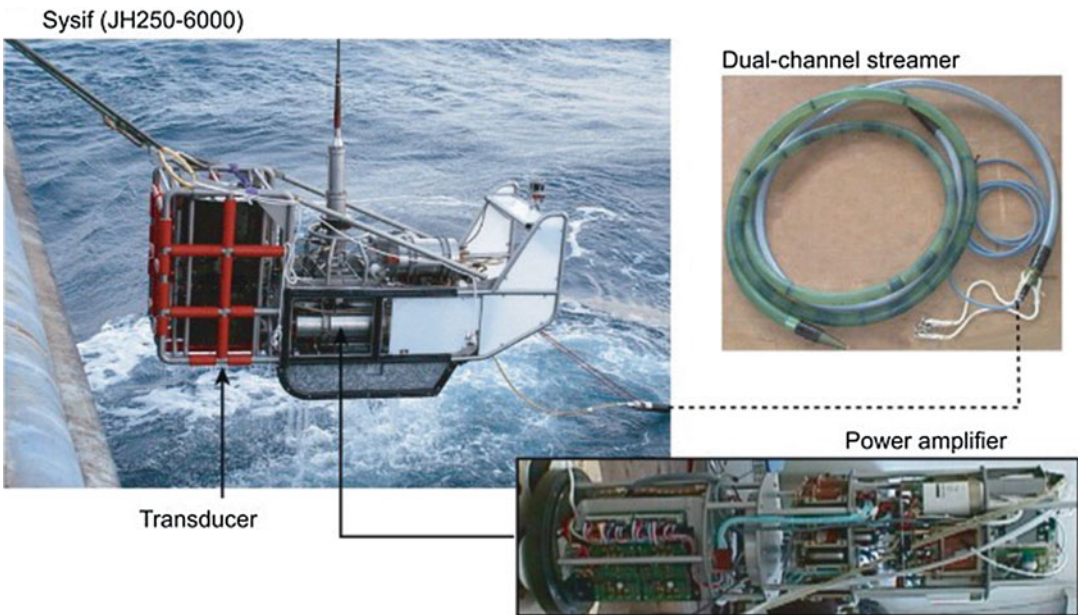
off Biloxi, Mississippi using the research vessel Tommy Murro (Fig. 6).

In 2008, the research team of the Institute of Acoustics of the Chinese Academy of Sciences, with the support of the China Oceanic

Association, started the upgrade of the DTA-4000 (as shown in Fig. 7, left) acoustic deep towing system. First, it upgraded the working depth to 6000 meters, and then gradually installed shallow profile sonar, DVL, and other equipments



Deep Tow System, Fig. 7 Deep tow system developed in the Institute of Acoustics, Chinese Academy of Sciences, DTA-4000 (left), DTA-6000 (right). (Liu et al. 2015)



Deep Tow System, Fig. 8 Ifremer near-bottom seismic system (Marsset et al. 2010)

to the DTA-6000 acoustic deep towing system (as shown in Fig. 7, right), which makes the deep tow system applicable to multiple ships and enhances the function and stability of the system software. During the upgrade and application, from 25 to 27 March 2011, with the scientific research ship Ocean I in the sulfide zone of the Mid-Atlantic Ridge, the DTA-6000 acoustic deep towing system completed three effective survey operations. The length of the dragging survey line is 30 km. In 2013, the DTA-6000 deep tow system successfully completed two 54.6-km detection missions in Haishan along with the scientific

research ship Ocean VI. For the first time, the micro-topography and shallow stratigraphic profile data were simultaneously obtained in Caiwei Haishan Mountain. In 2014, the deep tow system undertook the search task of Malaysia Airlines' missing passenger plane, with a search and rescue route of 34 km. (Liu et al. 2015).

The deep-towed seismic project is part of a more general project aiming at a deep-towed multisensor platform including a side-scan sonar, a magnetometer, and a sub-bottom profiler, as shown in Fig. 8. A neutrally buoyant towed fish behind a heavy depressor was chosen for the

future project to protect the imagery sensors from heave transmitted through the towing cable. (Marsset et al. 2010).

Cross-References

- [Acoustic Deep Tow System](#)
- [Acoustic Towed System](#)
- [Deep Tow System](#)
- [Tow System](#)
- [Tow-Body](#)
- [Towed Vehicle](#)
- [Tow-Fish](#)

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Deep-Ocean Assessment and Reporting of Tsunamis

- [Deep-Ocean Assessment and Reporting of Tsunamis \(DART\) BUOY](#)

Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY

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Synonyms

[DART; Deep-ocean Assessment and Reporting of Tsunamis](#)

Definition

The full name of DART buoy is Deep-ocean Assessment and Reporting of Tsunamis buoy, which is an enhanced tsunami warning system. The DART buoys are used to confirm the existence of tsunami waves generated by undersea earthquakes. These buoys observe and record changes in sea level out in the deep ocean. This enhances the capability for early detection and real-time reporting of tsunamis before they reach land.

To ensure early detection of tsunamis and to acquire data critical to real-time forecasts, the National Oceanic and Atmospheric Administration (NOAA) has placed Deep-ocean Assessment and Reporting of Tsunami (DART) stations at sites in regions with a history of generating destructive tsunamis. NOAA completed the original 6-buoy operational array (map of original six stations) in 2001 and expanded to a full network of 39 stations in March, 2008 (Zhao et al. 2013).

Originally developed by NOAA, as part of the U.S. National Tsunami Hazard Mitigation Program (NTHMP), the DART Project was an effort to maintain and improve the capability for the early detection and real-time reporting of tsunamis in the open ocean. DART presently constitutes a critical element of the NOAA Tsunami Program. The Tsunami Program is part of a cooperative effort to save lives and protect property through hazard assessment, warning guidance, mitigation, research capabilities, and international coordination. NOAA's National Weather Service (NWS) is responsible for the overall execution of the Tsunami Program. This includes operation of the U.S. Tsunami Warning Centers (TWC) as well as leadership of the National Tsunami Hazard Mitigation Program. It also includes the acquisition, operations, and maintenance of observation systems required in support of tsunami warning such as DART, local seismic networks, coastal, and coastal flooding detectors. NWS also supports observations and data management through the National Data Buoy Center (NDBC) (2018).

Scientific Fundamentals

Working Mechanism

DART systems consist of an anchored seafloor bottom pressure recorder (BPR) or so-called tsunameter and a companion moored surface buoy for real-time communications (Gonzalez et al. 1998). An acoustic link transmits data from the BPR on the seafloor to the surface buoy. The schematic diagram of DART system is shown in Fig. 1 (DART[®] 1998).

The BPR collects temperature and pressure at 15 s intervals. The pressure values are corrected

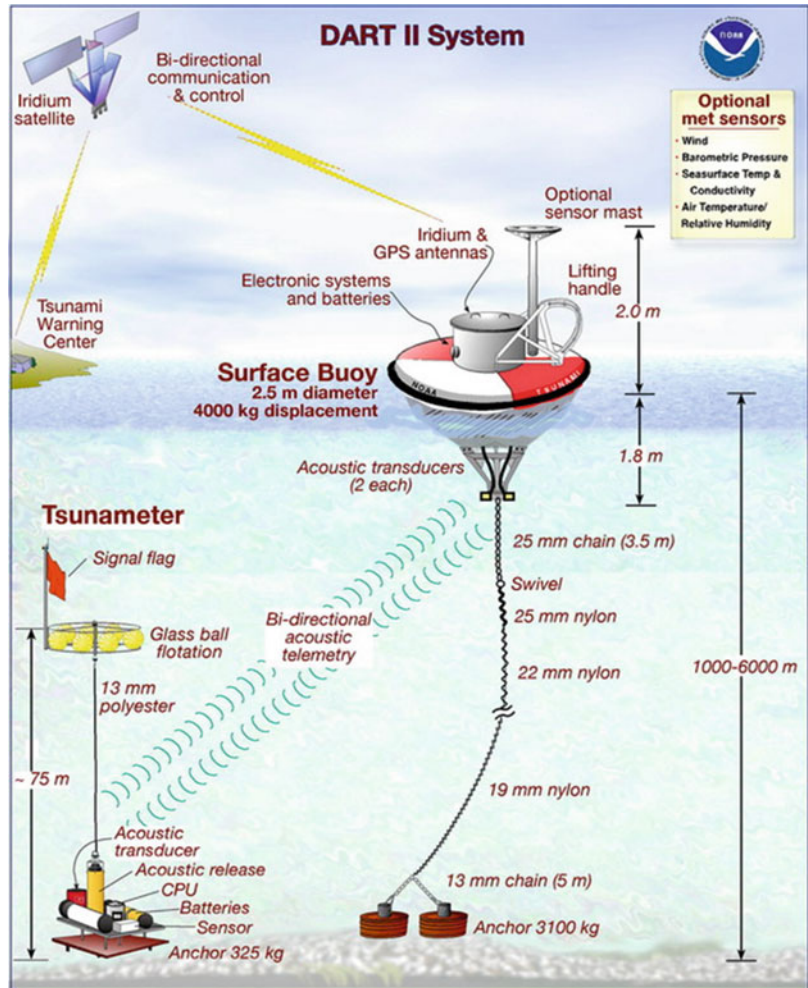
for temperature effects and the pressure converted to an estimated sea-surface height (height of the ocean surface above the seafloor) by using a constant 670 mm/psia. The system has two data reporting modes, standard and event. The system operates routinely in standard mode, in which four spot values (of the 15-s data) at 15-min intervals of the estimated sea surface height are reported at scheduled transmission times. When the internal detection software (Mofjeld) identifies an event, the system ceases standard mode reporting and begins event mode transmissions. In event mode, 15 s values are transmitted during the initial few minutes, followed by 1-min averages. Event mode messages also contain the time of the initial occurrence of the event. The system returns to standard transmission after 4 h of 1-min real-time transmissions if no further events are detected (DART[®] 1998).

Generations

The first-generation DART (DART I) systems had one-way communications from the BPR to the Tsunami Warning Centers (TWC) and NDBC via the western Geostationary Operational Environmental Satellite (GOES West) (Milburn et al. 1996). DART I became operational in 2003. NDBC replaced all DART I systems with the second-generation DART systems (DART II) in early 2008. DART I transmits standard mode data once an hour (four estimated sea-level height observations at 15-min intervals) (Milburn et al. 1996).

DART II became operational in 2005 (Green 2006). A significant capability of DART II is the two-way communications between the BPR and the TWCs/NDBC using the Iridium commercial satellite communications system (Meinig et al. 2005). The website for the DART data is supported by the National Data Buoy Center. A context diagram showing a DART II system and the related telecommunication nodes is shown in Fig. 2. It is clearly shown that the DART II system inside the dashed lines consists of two physical components: a tsunameter on the ocean floor and a surface buoy with satellite telecommunication capability. The two-way communications allow the TWCs to set stations in event mode in anticipation of possible tsunamis or

Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY, Fig. 1 The DART II tsunami buoy system (DART® 1998). (Courtesy of NOAA)

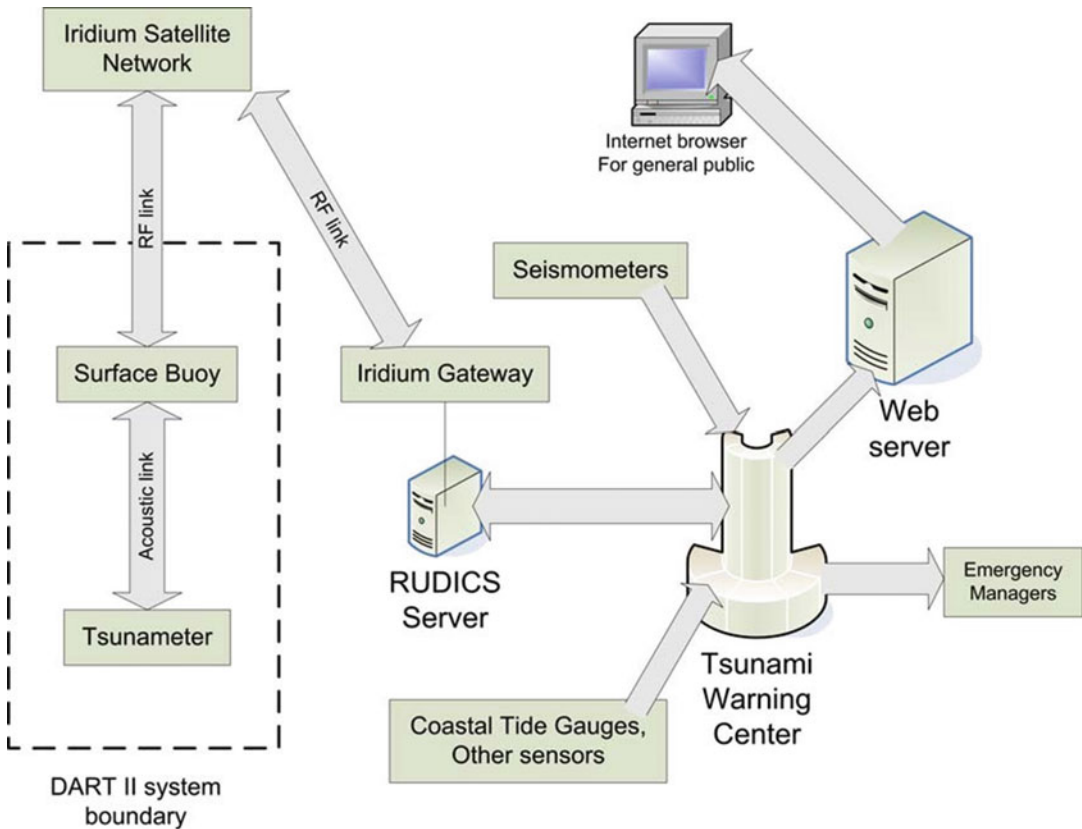


retrieve the high-resolution (15-s intervals) data in 1-h blocks for detailed analysis. DART II systems transmit standard mode data, containing 24 estimated sea-level height observations at 15-min intervals, once every 6 h. The two-way communications allow for real-time troubleshooting and diagnostics of the systems. The DART buoys have two independent and redundant communications systems. NDBC distributes the data from both transmitters under separate transmitter identifiers. NDBC receives the data from the DART II systems, formats the data into bulletins grouped by ocean basin, and then delivers them to the National Weather Service Telecommunications Gateway (NWSTG) that then distributes the data

in real-time to the TWCs via NWS communications and nationally and internationally via the Global Telecommunications System (NDBC 2018).

DART II Buoy System Components

From the above, we can see that the DART II buoy system, which incorporates the latest technologies and advances, has longer maintenance intervals, and feature two-way communication. The DART II Buoy System consists of two physical components: a tsunameter on the ocean floor and a surface buoy with satellite telecommunications capability. Here we will introduce the two components in DART II.



Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY, Fig. 2 Context diagram showing a DART II system and the related telecommunication nodes (Meinig et al. 2005)

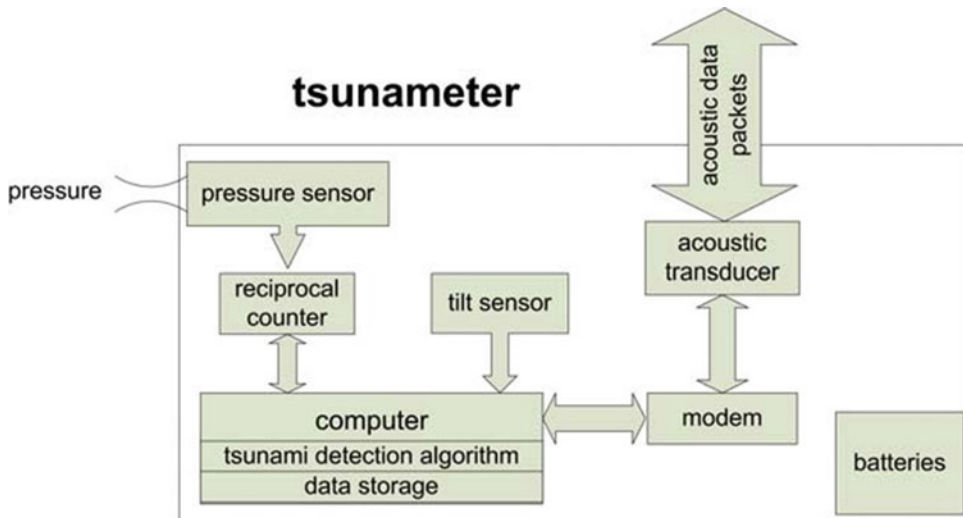
Tsunameter

The block diagram in Fig. 3 shows how the components of a tsunameter function together (Meinig et al. 2005). The computer reads pressure readings, runs a tsunami detection algorithm, and sends and receives commands and data to and from the buoy via an acoustic modem.

Pressure Sensor The DART II pressure sensor is a 0–10,000 psi model 410K Digiquartz unit manufactured by Paroscientific, Inc. The transducers use a very thin quartz crystal beam, electrically induced to vibrate at its lowest resonant mode. The oscillator is attached to a Bourdon tube that is open on one end to the ocean environment. The pressure sensor outputs two frequency-modulated square waves, proportional to the ambient pressure and temperature. The

temperature data is used to compensate for the thermal effects on the pressure-sensing element (Eble and Gonzalez 1991).

Reciprocal Counter The high resolution precision reciprocal counting circuit continuously measures the pressure and temperature signals simultaneously, integrating them over the entire sampling window, nominally set to 15 sec. There is no dead period between the sampling windows. The circuit has a sub-millimeter pressure and sub-millidegree temperature least-count resolution. The reference frequency for the reciprocal counter is derived from a low power, very stable, 2.097152 MHz, temperature-compensated crystal oscillator. A real-time calendar-clock in the computer also uses this reference for a time base. At the end of each sampling window, the computer



Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY, Fig. 3 Tsunameter block diagram showing how the components interact (Meinig et al. 2005)

reads the pressure and temperature data and stores the data in a flash memory card. A 15 s sampling period generates about 18 megabytes of data per year.

Computer The embedded computer system in both the buoy and the tsunameter was designed around the 32-bit, 3.3 volt Motorola 68332 micro-controller, and was programmed in C. It was built to be energy efficient for long-term battery powered deployment. The computer has 4 Mb of flash memory, a 12-bit A/D converter with 8 input channels, two RS232 channels, a hardware watchdog timer, a real-time clock, and 512 bytes of RAM. The embedded computer implements and regulates the primary functions of the surface and seafloor units: transmitting data communications, running the tsunami detection algorithm, storing and retrieving water column heights, generating checksums, and conducting automatic mode switching.

Acoustic Modem and Transducer A Benthos11 ATM-880 Telesonar acoustic modem with an AT-421LF directional transducer has a 40° conical beam which is used to transmit data between the tsunameter and the surface buoy. Modems transmit digital data via MFSK

modulated sound signals with options for redundancy and convolutional coding. Transducers are baffled to minimize ambient noise from entering the receiver.

Tilt Sensor Each tsunameter has a Geometrics 900-45 tilt sensor mounted in the base of one of the housings. This is used to determine the orientation of the acoustic transducer when the system has settled on the seafloor. If the tilt is greater than 10°, the tsunameter can be recovered and redeployed. The watch circle of the surface buoy could carry it out of the acoustic projection cone from the tsunameter if the angle from the vertical is too great.

Batteries The tsunameter computer and pressure measurement system uses an Alkaline D-Cell battery pack with a capacity of 1560 watt-hours. The acoustic modem in the tsunameter is powered by similar battery packs that can deliver over 2000 watt-hours of energy. These batteries are designed to last for 4 years on the seafloor; however, this is based on assumptions about the number of events that may occur and the volume of data request from the shore. Battery monitoring is required to maximize the life of the system.

Tsunami Detection Algorithm Each DART II tsunameter is designed to detect and report tsunamis autonomously. The Tsunami Detection Algorithm works by first estimating the amplitudes of the pressure fluctuations within the tsunami frequency band, and then testing these amplitudes against a threshold value. The amplitudes are computed by subtracting predicted pressures from the observations, in which the predictions closely match the tides and lower frequency fluctuations. If the amplitudes exceed the threshold, the tsunameter goes into Event Mode to provide detailed information about the tsunami.

Reporting Modes Tsunameters operate in one of two data reporting modes: a low power, scheduled transmission mode called “Standard Mode” and a triggered event mode simply called “Event Mode.”

Standard Mode reports once every 6 h. Information reported includes the average water column height, battery voltages, status indicator, and a time stamp. These continuous measurements

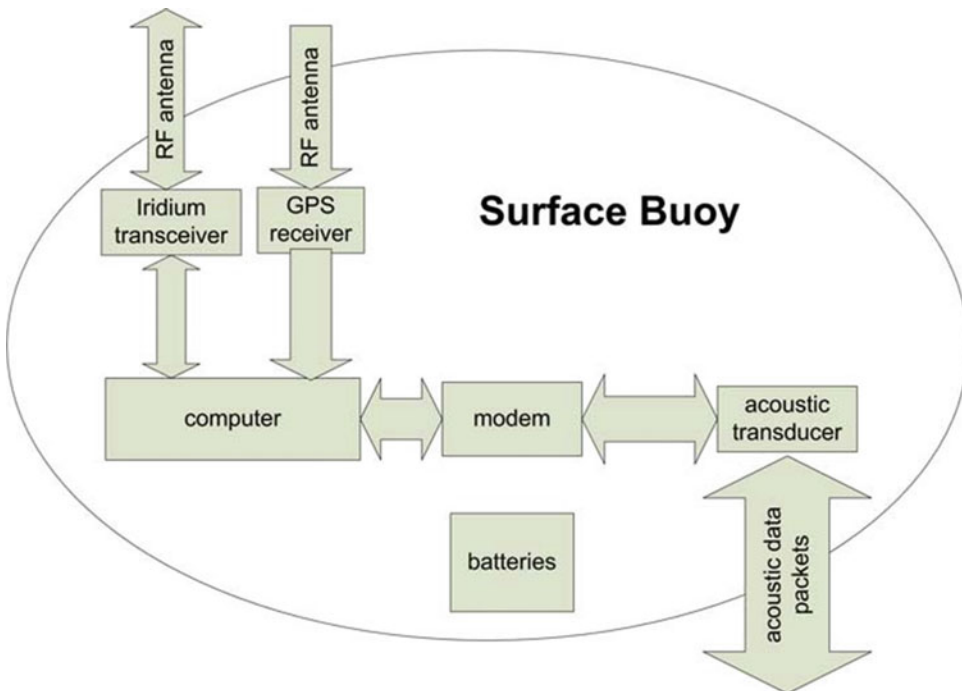
provide assurance that the system is working correctly.

Event Mode reports events such as earthquakes and/or tsunamis when a detection threshold is exceeded. The Tsunami Detection Algorithm triggers when measured and predicted values differ by more than the threshold value. Waveform data are transmitted immediately (less than a 3-min delay).

Tsunami waveform data continue to be transmitted every hour until the Tsunami Detection Algorithm is in a non-triggered status. At this point, the system returns to the Standard Mode.

Surface Buoy

The DART II surface buoy, shown in Fig. 4, relays information and commands from the tsunameter and the satellite network (Meinig et al. 2005). The buoy contains two identical electronic systems to provide redundancy in case one of the units fails. The Standard Mode transmissions are handled by both electronic systems on a preset schedule. The Event Mode transmissions, due to their



Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY, Fig. 4 Block diagram of DART II surface buoy (Meinig et al. 2005)

importance and urgency, are immediately transmitted by both systems simultaneously.

The surface mooring uses a 2.5 m diameter fiberglass over foam disk buoy with a displacement of 4000 kg. The mooring line is 19 millimeter eight-strand plaited nylon line with a rated breaking strength of 7100 kg, and is deployed to maintain a tight watch circle, keeping the buoy positioned within the cone of the acoustic transmission. In temperate areas where fish tend to aggregate and bite lines, wire rope is used on the upper few hundred meters of the mooring.

Two downward-looking transducers are mounted on the buoy bridle at a depth of 1.5 m below the sea surface. A multilayered baffle system of steel, lead, and syntactic foam shields the transducers from noise and cushions them with rubber pads for a soft mount.

Modem and Acoustic Transducer The Benthos Telesonar acoustic modems and transducers are the same as used in the tsunameter. To improve the reliability of data transmission, two identical systems are used on the buoy.

Computer The computer is the same type used in the tsunameter as described in section “[Tsunameter](#)”. It processes messages from both the satellite and the tsunameter.

Iridium Transceiver A Motorola 9522 L-Band Iridium transceiver from NAL Research provides data connectivity via the Iridium Satellite Network. The buoy computer connects to the transceiver using an RS232 serial port. Data is transferred at 2400 baud similarly to the familiar dial-up modem connections. A typical Standard Mode report takes approximately 30 s, including the time it takes to complete the connection, transmit the data, and disconnect.

GPS A Leadtek model 9546 GPS receiver is used to maintain the buoy’s computer clock’s accuracy to within ~1 s of GMT. Additionally, a GPS position is reported once per day to monitor buoy position.

Batteries The buoy’s fiberglass well houses the system electronics and power supply, which is

made up of packs of D-cell alkaline batteries. The computer and Iridium transceiver are powered by 2560 watt-hour batteries; the acoustic modem is powered by 1800 watt-hour batteries. These batteries will power the buoy for at least 2 years.

The buoy is designed to mitigate the potentially dangerous build up of hydrogen gas that is naturally vented from alkaline cells. Design features include: (1) hydrogen getters (such as those from HydroCap Corp); (2) pressure relief valves; and (3) spark-free components such as fiberglass or plastic.

Data Communications

There are messages that are sent and received to and from the DART II systems. The data communications include workstation (on land) to buoy, tsunameter to buoy, buoy to satellite, and satellite to ground stations. Details can be found in Meinig et al. (2005) about how the data is physically transported over the distance between the hardware components, what information is contained in the messages and how the message is formatted.

Key Applications

As described above, the DART buoys observe and record changes in sea level out in the deep ocean so as to enhance the capability for early detection and real-time reporting of tsunamis before they reach land. During the application of DART systems, the following should be noted.

Site Characteristics

To reliably send and receive the acoustic packets to and from the tsunameter, which might be submerged between 1000 and 6000 m below the buoy, the tsunameter must be located on a relatively flat portion of the ocean floor, and the buoy must be moored such that it stays within a 40° cone; a cone whose vertex is at the tsunameter, and whose base encompasses the buoy. Outside of this cone, the signal-to-noise ratio deteriorates rapidly, and data integrity will be compromised (Meinig et al. 2005).

Deep-Ocean Assessment and Reporting of Tsunamis (DART) BUOY,

Fig. 5 The deployment of a DART™ buoy in the Tasman Sea (Bureau of Meteorology 2018)



The mooring needs to be strong enough to withstand harsh ocean conditions of wind, waves, currents, fish bites, and vandalism (Meinig et al. 2005).

Location for Deployment

The best location for deployment of a DART buoy is determined by careful consideration of a number of factors. The DART buoy needs to be far enough away from any potential earthquake epicenter to ensure there is no interference between the earthquake signal at the buoy and the sea-level signal from the tsunami. On the other hand, the DART buoy needs to be close enough to the epicenter to enable timely detection of any tsunami and maximize the lead time of tsunami forecasts for coastal areas. In addition, DART buoys must ideally be placed in water deeper than 3000 m to ensure the observed signal is not contaminated by other types of waves that have shallower effects (e.g., surface wind-waves). International maritime boundaries must also be considered when deploying DART buoy systems. Figure 5 shows the deployment of a DART buoy in the Tasman Sea by the Australian government (Bureau of Meteorology 2018).

In-Water Life

The life cycle of a deployed DART buoy system is approximately 2–4 years. The maintenance regime involves the replacement of the surface buoy and the sea-floor pressure sensor every 1–2 years. The

devices retrieved during regular maintenance are refurbished and made ready for the next redeployment (Bureau of Meteorology 2018).

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Deep-Ocean Profiling Float (DOPF)

- [Profiling Float](#)

Deep-Sea Mining

- [Water Surface Support \(Layout and Recovery\) System](#)

Deepwater

- [Taut Mooring](#)

DEM, Discrete-Element Method

- [Numerical Simulation of Ice-Going Ships](#)

Density

- [AUV/ROV/HOV Hydrostatics](#)

Dependability

- [Reliability and Safety in Offshore Engineering](#)

Deployment and Recovery

- [Water Surface Support \(Layout and Recovery\) System](#)

Derrick Barge (DB)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Design and Installation

- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

Design of Mooring System

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Harbin Engineering University, Harbin, China

Synonyms

[Floating structure](#); [Mooring system](#); [Station-keeping system](#)

Related Specifications

The design of mooring system is a complicated integrated system, including design specifications, sea conditions, characterization of floating structures, hydrodynamic performance, mooring line data, numerical simulations, data analysis and comparison, etc.

An industry widely recognized positioning system analysis specification and standard is an indispensable important reference in the entire design process. The specification mainly introduces the function, form, design, and analysis method of the floating structure positioning system and gives the corresponding design criteria. At present, there are four main types of analysis specification for mooring system commonly used worldwide:

1. API RP 2SK – Specification for design and analysis of floating structure positioning system
2. API RP 2SM – Specification for the design, manufacture, installation, and maintenance of positioning systems for mooring lines in the form of synthetic materials
3. DNV Offshore Standard-E301 – Marine engineering mooring positioning specification
4. ABS Guide for Building and Classing Floating Production Installations – Floating production and installation specifications

The above specifications specify the common design standards for mooring systems.

Considerations and Design Process

Considerations

In the initial stage of design, it is necessary to consider comprehensively various possible situations, so that reasonable and accurate analysis can be carried out in the later stage. The following factors mainly need to be considered (Huang 2000):

1. Design standards for mooring systems, design loads, design life, operational and maintenance requirements, etc.
2. The pipeline between the riser-type seabed and the production of the drilling platform, so its existence is one of the main limiting factors for the mooring system. It limits the displacement range of the platform and, importantly, the mooring line and the riser. The interaction between the waves and the current on the riser will increase the movement of the platform, which is equivalent to increasing the environmental load of the mooring system, but at the same time, the riser system has a certain rigidity. The mooring system has certain auxiliary functions.
3. The layout of the mooring system must take into account the layout of the subsea facilities, such as the distribution of the mid-sea system, the riser foundation, and the submarine pipelines. For the installation, operation, and

maintenance phases of the mooring system, the mooring lines cannot combine with the submarine facilities. There is any collision; otherwise there will be great damage to each other.

In addition to the above basic factors, the designer must also understand the potential failure risks of the mooring system and avoid dangerous situations during the design process. In API RP 2SK, the following conditions are usually considered: intact condition, damaged condition, and transient condition (Table 1).

Design Procedure

The first step of the mooring design process is basic data input, which mainly includes the basic characteristic parameters of the floating structure, its operation requirements, and working environment conditions. The basic characteristic parameters mainly include the main dimension of the structure, the profile diagram of the ship structure, the position of the center of gravity, the moment of inertia, the displacement under different design conditions, the draft, the layout of the upper deck structure, the wind receiving area, etc. If there is a turret system, it is also necessary to provide its exact position and size, structure type, and characteristic parameters, and there will be a big gap between the mooring method and the mooring material selection. Therefore, it is necessary to determine accurately the relevant parameters of the floating structure under different working conditions to ensure the rationality of the design of the mooring system. Multipoint mooring and single-point mooring are two common

Design of Mooring System, Table 1 Recommended analysis methods and conditions

Type of mooring	Analysis method	Conditions to be analyzed
Permanent mooring system		
Strength analysis	Dynamic	Intact/damaged
Fatigue analysis	Dynamic	Intact
Mobile mooring system		
Strength analysis	Quasi-static or dynamic	Intact/damaged/transient
Fatigue analysis	Not required	Not required

mooring methods. Multipoint mooring is a natural extension of the traditional form of ship mooring, that is, a certain number of mooring lines are symmetrically distributed around the platform, which can control the linear motion and the rotational motion of the platform. Multipoint mooring is simple and economical and can install more risers and umbilical systems. It is suitable for sea areas with better sea conditions or a single sea area with wind and waves. However, for a boat-shaped floating body, if it is subject to a large trajectory, the mooring line of the wave will be subject to a large load, and there is a risk of breaking. Common SPAR platforms, semisubmersible platforms, and some FPSOs use multipoint mooring. Considering the drawbacks of multipoint mooring of the boat-shaped floating body, a single-point mooring has appeared, allowing the hull to rotate freely around a single point, so that the bow always faces the maximum environmental load, thus effectively reducing the force of wind and waves (Liu 2012). Therefore, most FPSO mooring systems are single-point moorings. It is easy to operate, safe, flexible, and environmentally adaptable. However, compared to multipoint mooring, single-point mooring has higher manufacturing costs and more complicated technology. With the increasing demand, there are many types of single-point mooring, such as turret mooring, tower mooring, catenary buoy mooring (CALM), and single-anchor leg mooring system (SALM).

The type and function of the floating structure determine the choice of mooring materials and mooring methods. Therefore, the design of the mooring system first needs to determine the type, main purpose, and service life requirements of the floating structure. For permanent mooring systems, the working time is generally required to be several decades. For mobile mooring systems, the working time in the same sea area is shorter, which can be as short as several months, depending on the specific situation. In addition, restrictions on the use of risers or other pipelines or other structures and facilities around the floating structure, the displacement of the floating structure in certain directions is usually limited, and the mooring system has higher requirements.

Wind, wave, and current are the main environmental loads directly acting on the floating structure and its mooring system, and it decides whether the mooring system works safely (Fujii et al. 1982). Therefore, it is necessary to fully collect the long-term statistical data of the sea area and grasp the accuracy of each main characteristic parameter. It is no doubt that the working water depth is an important factor in determining how to choose the mooring form and the mooring line material. At the same time, the composition characteristics, slope, and seabed structure layout of the seabed soil also play a major role in the selection of the seabed anchor and the layout of the mooring system. Finally, we must also understand the climate change, atmospheric icing, tides, and growth of marine organisms in the sea area. The type and growth rate of seawater microorganisms will affect the quality of floating structures and mooring lines, hydrodynamic diameters, towing force coefficient, etc. In short, the more sufficient the understanding of the working environment conditions, the better it is to design a safer and more reasonable mooring system.

All the basic information is ready, that is, the design and scheme of the mooring system, including the choice of mooring method, the composition of the mooring line material and the length and material characteristics of each part, the floating structure fairlead, the determination of the anchor position, and the choice of the anchor type. The anchoring points fixed to the seabed are the critical part of a mooring system for keeping a floating structure on location. Depending on soil conditions and the required performance, there are various types of anchors that can be selected. The feasibility of the final design requires comparing the calculated results with the actual requirements. The API RP 2SK specification proposes the following four basic design guidelines. There are two types of anchor: conventional drag embedment anchors and anchors designed to resist vertical loads, such as pile, suction, and vertically loaded plate anchors. Finally, the floating structure and its mooring system are analyzed and calculated, and a reasonable calculation method is determined. The overall

motion response of the floating structure and the stress of the mooring line and the anchor are obtained. Comprehensively evaluate the rationality of the design of the mooring system. If there are any defects, make appropriate adjustments until the requirements are met.

Computational Methods

About the computational methods, the following aspects should be considered:

1. Frequency domain analysis, time domain analysis, and combined time and frequency domain analysis.

In the frequency domain analysis, the general equations of motion describing the response of the vessel are decoupled and analyzed separately for mean, low, and wave frequency responses. Statistical peak values, such as significant or maximum responses, are then evaluated based on certain peak response distributions. Finally, the wave frequency and low-frequency responses are properly combined to yield the maximum combined response for specific storm duration.

In the time domain analysis, the general equations of motion describing the combined mean, low, and wave frequency response of the vessel are solved in the time domain. The forcing functions include the mean, low frequency, and wave frequency forces due to wave, wind, current, and thrusters. The dynamic equations describing the vessel, mooring lines, risers, and thruster forces are all included in a single time domain simulation. Time histories of all system parameters (vessel displacements, mooring line tensions, anchor loads, etc.) are available from the simulation, and the resulting time histories are then processed statistically to yield expected extreme values. The time domain simulation should be long enough to establish stable statistical peak values.

In the combined time and frequency domain analysis, time and frequency domain solutions for mean loads, wave, and low-frequency motions can be combined in different ways. In a typical approach, the mean and low-

frequency responses (vessel displacements, mooring line tensions, anchor loads, etc.) are simulated in time domain, while the wave frequency responses are solved separately in frequency domain. The frequency domain solution for wave frequency responses is processed to yield either statistical peak values or time histories, which are then superimposed on the mean and low-frequency responses (API RP 2SK 2005).

2. Quasi-static and the dynamic analysis for the force analysis of mooring lines.

In the quasi-static analysis, the dynamic wave loads are taken into account by statically offsetting the vessel by wave-induced motions. Vertical motions and dynamic effects associated with mass, damping, and fluid acceleration on the mooring line are neglected. Research in mooring line dynamics has shown that the accuracy of tension predictions based on this method can vary widely depending on the vessel type, water depth, and line configuration.

In the dynamic analysis, the time-varying effects due to mass, damping, and fluid acceleration are accounted for. In this approach, the time-varying fairlead motions are calculated from the vessel's surge, sway, heave, pitch, roll, and yaw motions. Dynamic models are used to predict mooring line responses to the fairlead motions. Two methods, frequency domain and time domain analyses, can be used for predicting dynamic mooring loads. In the time domain method, all nonlinear effects including line stretch, line geometry, fluid loading, and sea bottom effects can be modeled. The frequency domain method, on the other hand, is always linear as the linear principle of superposition is used. Methods to approximate nonlinear effects in the frequency domain and their limitations should be investigated to ensure acceptable solutions for the intended operation (API RP 2SK 2005).

3. Non-coupling, semi-coupling, and full-coupling analysis for the processing of the coupling relationship between the mooring system and the floating structure.

In the non-coupling analysis, the structure and the mooring responses are analyzed separately. The following mooring/riser effects can be included: mooring system stiffness, direct current loads, and estimated slow drift damping. The structure static offset, the slow drift, and wave frequency motions are solved, and, based on the derived FPSO responses, mooring line responses are then predicted.

In the semi-coupling analysis, the mooring line dynamics are fully modeled. Using the wind, current, wave drift coefficients, and structure hydrodynamic coefficients, the structure responses are solved in the time domain taking into account the mooring line responses. It is assumed that the structure wave frequency motions are not affected by the mooring lines. Even though the mooring lines are not coupled with the structure wave frequency motions, their contributions to such motions are often negligible since the structure's inertia properties are an order of magnitude higher than those of the mooring lines.

In the full-coupling analysis, the mooring line dynamic responses are coupled to the whole range of structure responses, including the wave frequency responses. In this method, the complete system dynamic equations are solved simultaneously in the time domain (Luo and Baudic 2003).

At last, it should be pointed out that it is difficult to accurately calculate the low-frequency motion, especially the low-frequency motion damping. The evaluation of viscous damping has accumulated a certain amount of experience (Huse 1991). It is also considered in the low-frequency motion damping of computational analysis, but the mooring system damping is more complicated and difficult to accurately estimate, which is usually ignored. However, the study found that these damping are important, even larger than viscous damping. If ignored, it may lead to overestimation of low-frequency motion. For large ship structures, low-frequency motion is an important design factor and should be accurately evaluated for low-frequency damping. The accurate evaluation of low-

frequency motion damping remains to be further studied.

Feasibility Verification

The feasibility of the final design requires comparing the calculated results with the actual requirements. The APL RP 2SK specification proposes the following four basic design guidelines: limitation of motion displacement of floating structures, length of mooring line, ultimate bearing capacity of anchor, and mooring line breaking force. The extreme value of the mooring line tension is generally determined by its ratio to the minimum breaking strength (MBS) of the mooring line, i.e., the safety factor, and compares the calculated safety factor with the safety factor required by the specification. If the former is larger than the latter, the mooring line is safe, and the design of the mooring system is reasonable; otherwise, it is not safe, and the mooring system needs to be redesigned. The specification determines the safety factors of different analysis conditions and different analysis methods, which can be used as a basic standard for mooring system analysis (Table 2).

All in all, the design of the mooring system requires comprehensive consideration of the structural requirements, the environmental conditions, and the cost of its own construction. Theoretical analysis, numerical simulation, and model testing are the main design and verification methods. Although there are still some difficulties in some aspects, such as the limitation of the dimensions of the deepwater mooring model test in offshore basins, great progresses have been made.

Design of Mooring System, Table 2 Mooring safety factor

Condition	Analysis method	Tension limit (percent of MBS)	Factor of safety
Intact condition	Quasi-static	50	2.00
Intact condition	Dynamic	60	1.67
Damaged condition	Quasi-static	70	1.43
Damaged condition	Dynamic	80	1.25

Cross-References

- [Dynamic Analysis Method](#)
- [Static Analysis Method](#)

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Design of Pipelines and Risers

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Definition

Deepwater riser is the main connecting member between the floating offshore platform and the seabed wellhead, and it is the key part of the whole submarine pipeline system. Generally, there is high-pressure oil or air flow passing through the riser, and the external part bears the action of wave and current load (PRCI 1987). There are essentially two kinds of risers, namely,

rigid risers and flexible risers. A hybrid riser is the combination of these two. At present, there is no unified classification of deepwater riser, but according to its structure and use, it can be roughly classified as follows:

- Top tensioned risers
- Steel catenary risers
- Flexible risers
- Hybrid risers
- Drilling risers

Scientific Fundamentals

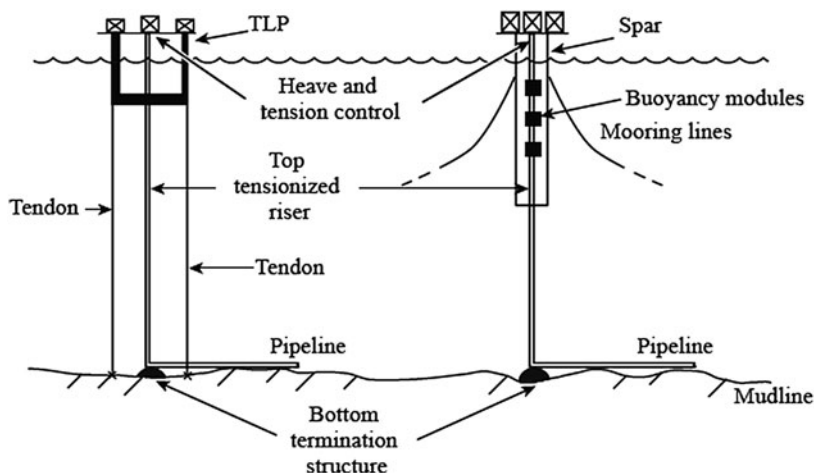
Top Tensioned Risers

Introduction

Top tensioned risers (TTRs) are used as the conduits between dynamic floating production units (FPU) and subsea systems on the sea floor, for dry tree production facilities such as spars and tension leg platforms (TLPs). Top tensioned riser (TTR) is one of the main configurations of riser systems, which has long been adopted to serve as the link between the floating production systems (FPSs) and the wellhead at the seabed for oil and gas transportation. For some TLP and spar applications, top tensioned risers provide a good solution. In this application, the flow line or pipeline terminates at a junction point beneath the structure (see Fig. 1). A straight riser is run down from the platform and terminates either in the same junction structure or in its own junction structure, in which case a short pipe section installed with the aid of an ROV connects the riser and flow line (API 1998).

In general, TLP risers use tensioners, whereas spar risers are tensioned by air cans, although one spar (Holstein) uses tensioners. Top tension ratios (ratio of top tension to riser apparent weight) vary greatly. On some TLPs, they have been designed to be as low as 1.2, while on others they have been in the range of 1.4–1.8. There is one significant difference between tensioners and air cans, resulting from the setdown of the riser top end as the platform is offset. In the case of tensioners, the top tension increases with platform offset because

Design of Pipelines and Risers, Fig. 1 Top tensioned riser



of tensioner stiffness. In the case of air cans, it is slightly reduced because of the compressibility of the air or gas in the cans.

General Design Considerations

The major design drivers of TTRs are, for instance:

- Allowable floater motions
- Allowable stroke of tensioning system
- Maximum riser top tension
- Size of stress joints and flex joints
- Keel joints
- Increasing length of riser joints
- Design criteria for the safety philosophy of liquid barriers, valve, and seals
- Current interface with array and vortex-induced vibration
- Impact between buoyancy cans and hull guides

Steel Catenary Risers

Introduction

Steel catenary risers (SCRs) are a preferred solution for deepwater wet tree production, water/gas injection, and oil/gas export (see Fig. 2). Statistics, illustrated in Fig. 3, demonstrates that the number of SCRs currently either under design or planned exceeds the current worldwide inventory of operating SCRs (Gelfgat et al. 2006).

Steel catenary risers are an elegant solution to the riser challenge of connecting to floating

production platforms in deepwater. These risers can be installed by laying a predetermined length of pipe, making the connections easier. They tolerate a certain amount of movement of the production platform, making them useful for TLPs, FPSs, FPSOs, and spars, as well as fixed platforms, compliant towers, and gravity structures. However, excessive motion can cause metal fatigue at or near the touchdown point and at or near the top support (Hatton and Willis 1998).

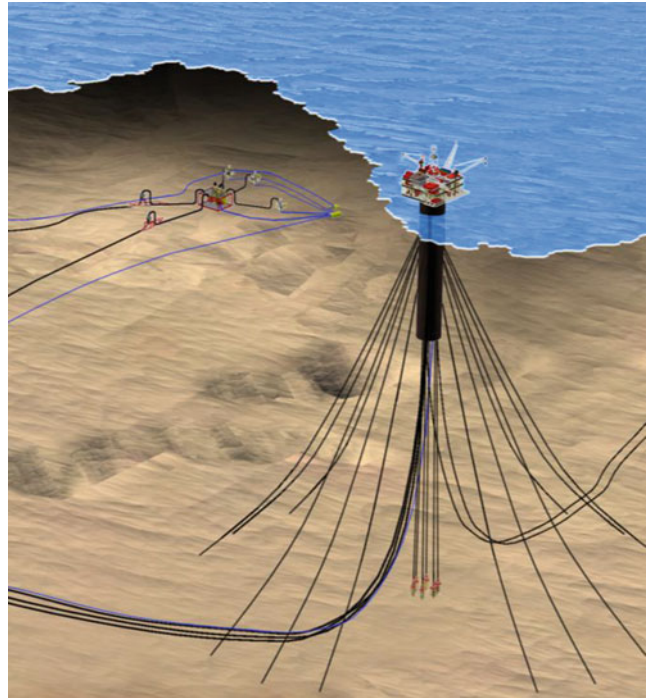
SCR Initial Design Analysis

In the pre-FEED phase, an initial design is carried out to define:

- Riser host layout (for interface with other disciplines)
- Riser hang-off system (flex joint, stress joint, pull-tube, etc.)
- Riser hang-off location, spacing, and azimuth angle (hull layout, subsea layout, total risers, and interference consideration)
- Riser hang-off angle for each riser
- Riser location elevation at hull (hull type, installation, and fatigue consideration)
- Global static configuration

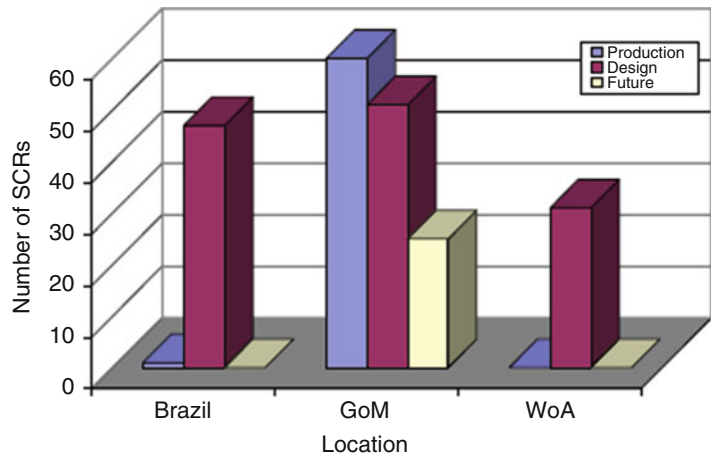
A static configuration may be determined based on catenary theory accounting for hang-off angle, water depth, and riser unit weight). The SCR design shall meet basic functional requirements such as SCR internal (and/or

Design of Pipelines and Risers, Fig. 2 SCR (steel catenary riser) (William et al. 2011)



D

Design of Pipelines and Risers, Fig. 3 Number of SCRs installed/proposed worldwide (215 SCRs) (Bai et al. 2004)



external) diameters, submerged tension on host vessel, design pressure/temperature, and fluid contents.

Due considerations shall be given to future tie-back porches to accommodate the variations for hang-off system, riser diameter, and azimuth angles and the required extreme response and fatigue characteristics (Song and Stanton 2007).

Flexible Risers

Introduction

Petrobras pioneered the use of flexible pipe as risers starting in 1978. Because of the bending characteristics, flexibles are most appropriate for floating systems that experience significant vertical and/or horizontal movement. Flexible pipe, because of its beneficial bending characteristic,

is often used to make the connections from the production equipment on floating systems to the production and export risers (Neto and Mauricio 2001). Because of these same bending characteristics, these flexibles are also finding more

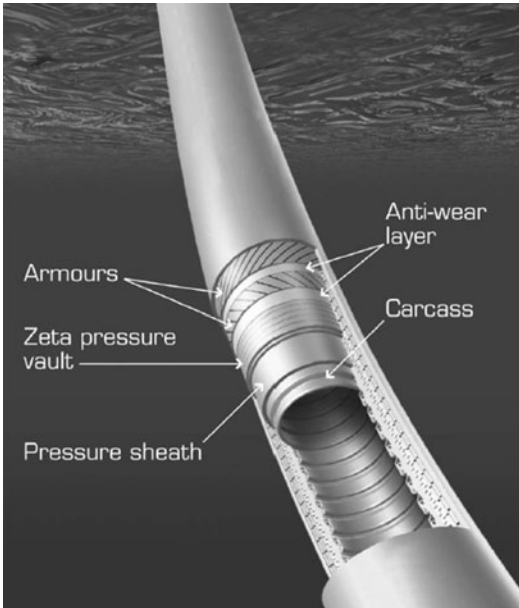
and more use as the primary risers. Today flexible pipes have been used as risers and flow lines in water depths down to 2,000 m (Fig. 4). Off-shore Brazil in the Campos Basin alone, approximately 4,000 km of flexible pipes have been deployed.

Model tests showed that flexible pipes could be used in the free-hanging mode but only in benign environments. In more severe environments, changes in lateral offset of the platform tended to induce unacceptably large variations in pipe tension and large changes in position of the touch-down point (TDP) at the seabed. To reduce those effects, a number of different architectures were developed for the section close to the seabed. Some of these are shown in Fig. 5.

Flexible Riser Design Analysis

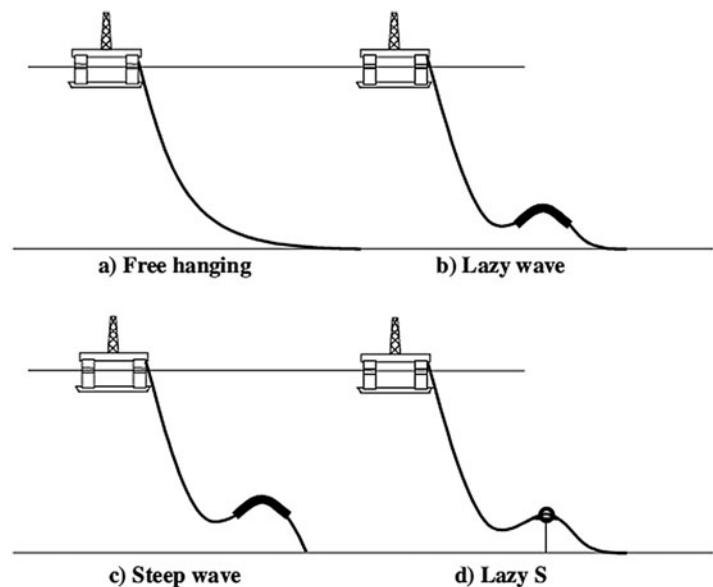
The essential tasks for design and analysis of flexible risers are similar to those described for other types of risers; see below (Novitsky and Serta 2002).

- **Design Basis Document:** The document should as minimum include the following:
 - Host layout and subsea layout
 - Wind, wave, and current data and vessel motion that are applicable for riser analysis



Design of Pipelines and Risers, Fig. 4 Flexible pipe wall structure. (Courtesy of Technip)

Design of Pipelines and Risers, Fig. 5 Example architectures for flexible risers



- Applicable design codes and company specifications
- Applicable design criteria
- Porch and I-tube design data
- Load case matrices for static strength, fatigue, and interference analysis
- Applicable analysis methodology
- **FE Modeling and Static Analysis:** a finite element model is built, and a nonlinear static analysis is carried out assuming the vessel is in NEAR, FAR, and CROSS positions.
- **Global Dynamic Analysis:** A global regular wave dynamic analysis is carried out assuming the vessel is in NEAR, FAR, and CROSS positions. A sensitivity study is performed on critical parameters such as wave periods, effect of marine growth, and hydrodynamic coefficients.
- **Interference Analysis:** A dynamic regular wave analysis is carried out to check the minimum clearance between the risers and with the vessel system along the water column, for various predefined load cases. The interference analysis shall confirm the selection of riser hang-off angles and departure angles, etc.
- **Cross-Sectional Model:** a detailed cross-sectional model is built to calculate key cross-sectional properties such as bending stiffness, axial stiffness, FAT pressure, etc.
- **Extreme and Fatigue Analysis:** The wire and tube stresses are calculated at design pressure. An extreme response analysis is carried out using regular wave theory to estimate tensions, cyclic angles, etc. The cross-sectional model is then used to perform to fatigue analysis.
- **Design Review:** This includes check of global configuration, bell mouth design, interference and fatigue design, etc. In some special situations, upheaval buckling and onbottom stability of flexible flow lines are also checked, following pipeline design practice (Walker and Davies 1983).

Typically, a detailed design of flexible pipe is carried out by the supplier for the flexible pipe materials. A third party, normally a riser engineering company, is engaged to carry out a verification of the design, as aforementioned.

Hybrid Risers

Introduction

Fisher and Berner (1988) presented the systems details, engineering design, testing, and installation of the first hybrid risers. The hybrid riser (shown in Fig. 6) uses a rigid pipe for the vertical free-standing portion and a flexible pipe for the near-surface dynamic motion region. Top tension riser is used to hold a vertical riser when the well is underneath the floating structure. A pre-tension is applied to the riser, so the riser pipe will not be in compression when the floating structure moves down. Figure 7 shows hydropneumatic tensioner of which the piston cylinder in each tank works like a shock absorber of automobile.



Design of Pipelines and Risers, Fig. 6 Hybrid riser (Brian and Chris 2000)



Design of Pipelines and Risers, Fig. 7 Riser top tensioner

Preliminary Analysis

A single string approach may be taken to model the hybrid riser system:

- Mass, bending stiffness, and axial stiffness are the sum of all members.
- Buoyancy diameter is taken as the outside diameter of the buoyancy modules.
- Air can is a model using pipe elements with appropriate buoyancy and drag properties.
- Flexible jumpers are modeled individually or in group.

The key objective of the preliminary analysis is to get indication of the requirements for (Wald 2004):

- Buoyancy and tension distribution along the riser
- VIV suppression devices

Low submerged weight and low tension can make the riser susceptible to VIV. However, it is recommended to avoid over-specification of strakes, which can increase drag and may have negative effect on buoyancy requirements and stresses at riser base.

Drilling Risers

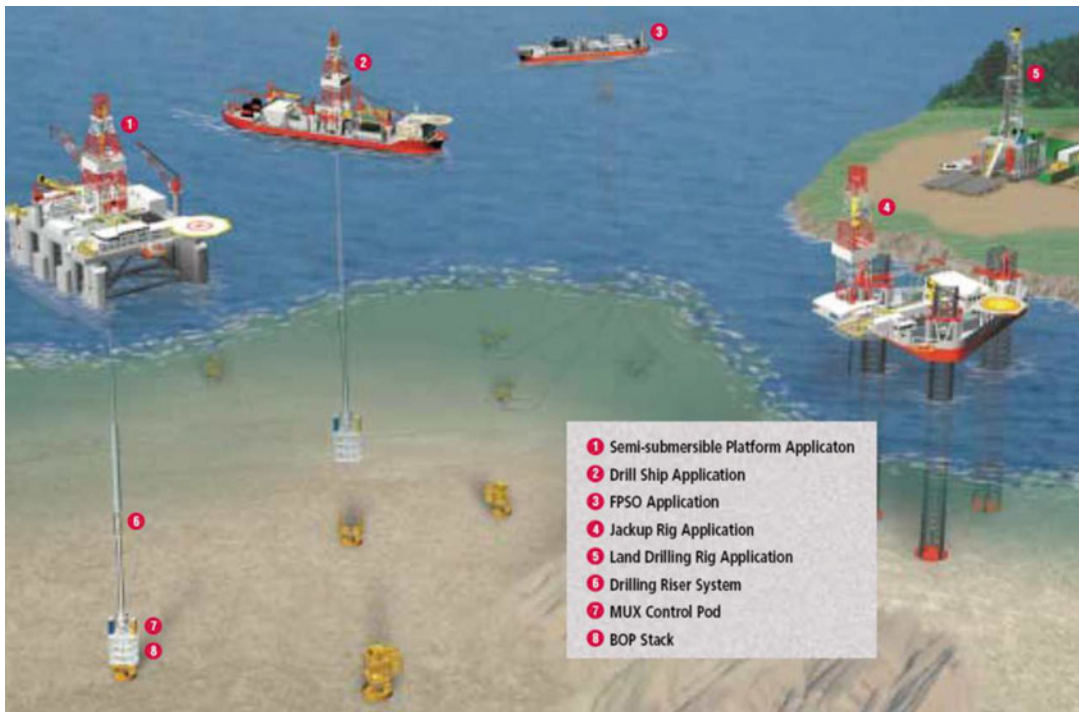
Introduction

Floating drilling risers are used on drilling semi-submersibles and drilling ships (see Fig. 8). As the water depth increases, integrity of drilling risers is a critical issue. The design and analysis of drilling risers are particularly important for dual operation, DP-based semi-submersible rigs. For the integrity assurance purpose, a series of dynamic analysis needs to be carried out. The objective of the dynamic analysis is to determine vessel excursion limits and limits for running/retrieval and deployment. In recent years, qualification tests are also required to demonstrate fitness for purpose for welded joints, riser coupling, and sealing systems. For risers installed in the Gulf of Mexico, vortex-induced vibrations are a critical issue. Some oil companies encourage use of monitoring systems to measure real-time vessel motions and riser fatigue damage. The monitoring results may also be used to verify the VIV analysis tools that are being applied in the design and analysis (Bai and Bai 2005).

Riser Design Criteria

Table 1 presents typical criteria to be used for determination of the operability limits for the drilling riser, defined mainly based on API 16Q.

For the drilling riser, the safety factor on fatigue life is 3 because the drilling joints can be inspected. The fatigue calculations are to account for all relevant load effects, including wave, VIV, and installation-induced fatigue. In some part (s) like the first joints nearest to the LFJ joint,



Design of Pipelines and Risers, Fig. 8 Drilling semi-submersibles, drilling ships, and drilling risers

Design of Pipelines and Risers, Table 1 Criteria for drilling riser operability limits

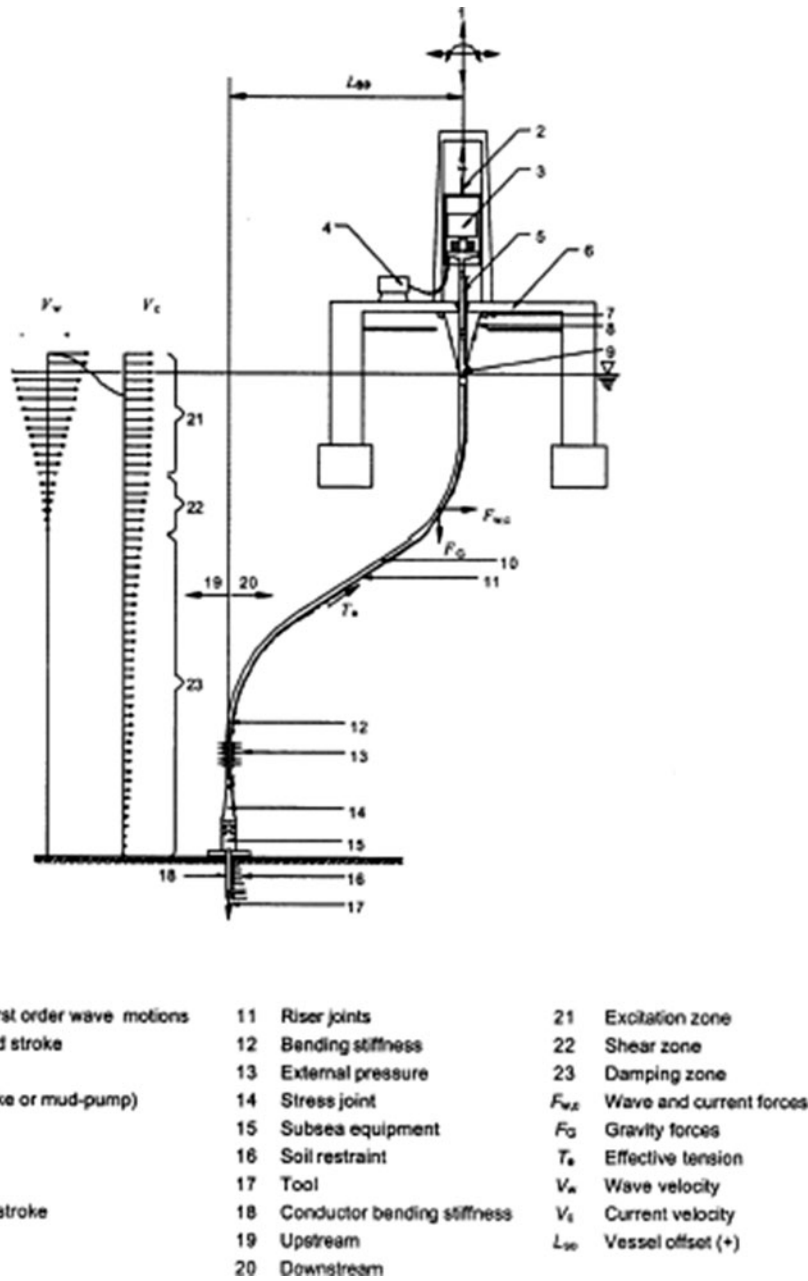
Design parameter	Definition	Drilling conditions	Non-drilling conditions
Low flex joint angle	Mean	1°	NA
	Max	4°	90% of capacity (9°)
Upper flex joint angle	Mean	2°	NA
	Max	4°	90% of capacity (9°)
Von Mises stress	Max	67% of σ_y	80% of σ_y
Casing bending moment	Max	80% of σ_y	80% of σ_y

the fatigue life could be less, in which case the fatigue life will determine the inspection interval.

Drilling Riser Analysis Methodology

Some keywords for drilling riser design and analysis are shown in Fig. 9 from structural analysis

point of view; a drilling riser is a vertical cable, under the action of currents. The upper boundary condition for the drilling riser cable is rig motions that are influenced by rig design, wave, and wind loads. One of the key technical challenges for deepwater drilling riser design is fatigue of VIV due to (surface) loop currents and bottom currents (Bai and Bai 2005).



Design of Pipelines and Risers, Fig. 9 Principal parameters involved in C/WO riser design and analysis

Cross-References

► Vortex Shedding and VIV Suppression

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Design of Renewable Energy Devices

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Synonyms

ALS – Accidental limit states; DLC – Design load cases; FLS – Fatigue limit states; LS – Limit states; PSF – Partial safety factors; SLS – Serviceability limit states; ULS – Ultimate limit states

Definition

Design is the creation and specification of a structure that fulfills the requirements of functionality, serviceability, and survivability during its lifetime.

Introduction

Renewable energy devices exist in various forms. Wind turbines, wave energy converters, tidal turbines, and photovoltaic systems are typical devices that convert energy, respectively, from wind, wave, tidal, and solar resources into electrical power.

Figure 1 (left) shows an offshore wind turbine which is composed of blades, nacelle, drivetrain, control systems, support structures, and so forth. During operation, the cyclic revolutions of the rotor are driven by the aerodynamic, gravitational, and inertial loads on the blades. The kinetic energy is further transferred to the electrical generator through the drivetrain. In a wind turbine, most components are designed separately by industrial partners with the aim to achieve similar level of probability of failure. Such a wind turbine system may be subject to complex external loading conditions including gust wind (Bierbooms

et al. 1999), wave impacts (Chella et al. 2012), biofouling (Degraer et al. 2010), and seabed scour (Prendergast et al. 2015) during the design life. Internal conditions may arise including mechanical system failures and faults. As operational and maintenance experiences are gained from the industry, the designs of commercial offshore wind turbines are increasingly improved. For instance, generally, upscaled wind turbines with greater rotor diameters are considered more cost-effective designs, and the direct-drive technologies are often applied to avoid gearbox failures.

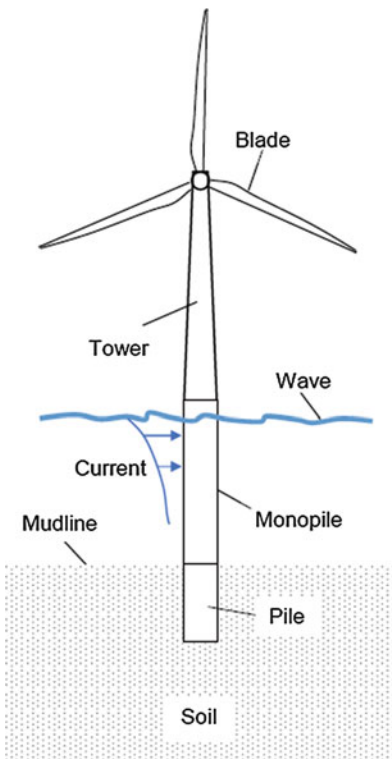
A concept of wave energy converter is also schematized in Fig. 1 (right). This concept was invented in the 1970s and was officially named “the Edinburgh duck” (Salter 1974). As illustrated, the rotor has a pear shape. The power takeoff rocks and turns over as waves pass by, transferring the energy of the impact to pumps. Such a device usually consists of components like rotor, hydraulic components, and generator.

Compared to wind turbines, wave energy converters are less mature commercially. Although a multitude of concepts have been proposed and tested through modeling and wave-tank tests, only a few designs progressed to sea testing. Design of such renewable energy devices is not standardized and faces challenges case by case. To account for the uncertainties in real-life scenarios, novel designs are usually conservative concerning the consumption of materials, and prototypes are needed to demonstrate the feasibility.

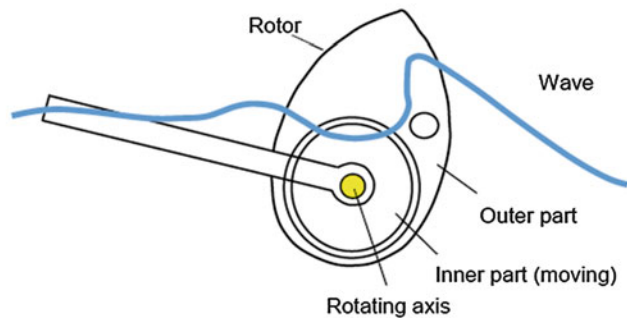
In the following, the focus will be on the design philosophy of structural components of offshore wind turbines which can be representative of other renewable energy devices that are industrialized.

Limit States (LS)

Any structural components in a renewable energy device can fail if the actual load effects exceed



Monopile-type offshore wind turbine



the Edinburgh duck wave energy converter

Design of Renewable Energy Devices, Fig. 1 Schematics of an offshore wind turbine and a wave energy converter

certain limits. Here, R is defined as a resistance and S is defined as a load effect. To ensure that the structural performance is in a desirable state, Eq. (1) must be satisfied with a high probability of occurrence. If such a relation is not satisfied, the structural performance is undesirable. The boundary between the desirable and undesirable domains is called a limit state; refer to Eq. (2).

$$R \geq S \quad (1)$$

$$R = S \quad (2)$$

Such relations are general and can be formulated for various failure modes. The ultimate limit states (ULS) correspond to the ultimate resistance for carrying loads. Some examples of ULS are (DNV GL 2014):

- Excessive yielding and buckling
- Loss of static equilibrium of the structure, e.g., capsizing
- Transformation of the structure into a mechanism

Among these examples, the loss of static equilibrium applies particularly to floating structures.

Fatigue refers to the cumulative damage due to fluctuating stresses and strains. Fatigue limit states (FLS) can be formulated based on the fracture mechanics or the S-N curve-based approaches; see Lassen and Recho (2013). Most of the design rules use the S-N curve-based approach.

Accidental limit states (ALS) are related to accidental loads which have relatively low occurrence rate but may have severe consequences. For

bottom-fixed offshore wind turbines, ship collision can be a typical scenario (Bela et al. 2017; Moulas et al. 2017). For floating renewable energy facilities with mooring systems, loss of mooring lines can also be a consideration.

Serviceability limit states (SLS) can be formulated based on specific requirements of engineering projects. For instance, large deformation of wind turbine blade tips may be treated as an SLS condition because of possible interference with the tower structure during operation. Because the settlements of foundations can cause tilting of the turbine structures which is not permitted, other SLS should be established accordingly.

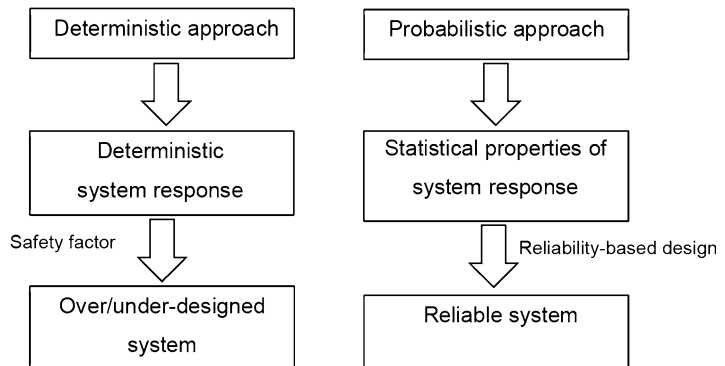
Categorization of the abovementioned LS is like that of offshore steel structures (DNV GL 2014).

Design Methods

Design of the structural components is an iterative process, and one needs to choose the design parameters and understand the loads and load effects. Two design approaches for load-bearing modern structures are presented in Fig. 2. In the deterministic approach, Eq. (3) gives an simplified expression for the design load (S_d). In Eq. (3), ψ is the safety factor, and S_k is the characteristic value of the load effect. ψ is chosen such that the possibility of unfavorable deviations of the characteristic values can be accounted for. Safety factors can be empirical and are not fully representative of the uncertainties the structure components are exposed to during the service life. Consequently, choice of

Design of Renewable Energy Devices,

Fig. 2 Design approaches for modern structures. (Adapted from Choi et al. 2006)



design parameters based on this approach often leads to over- or under-designed systems.

$$S_d = \psi \cdot S_k \quad (3)$$

As an improvement over the deterministic approach, the probabilistic (stochastic) approach addresses the uncertainties relating to loads, materials, and analysis methods and formulates the LS functions for critical failure modes explicitly. Then, the failure probability of structural members can be estimated using the structural reliability methods including Monte Carlo simulation, first-order reliability method, or second-order reliability method (Ditlevsen and Madsen 1996). In practice, the failure probabilities are often small, and the calculations of crude Monte Carlo simulations are inefficient. Improved reliability methods like the subset simulation technique (Au and Beck 2001) have been developed in recent years. The probabilistic method is applicable to design and analysis of modern civil engineering structures including offshore wind turbines (Jiang et al. 2017). Using this method, the aim is to design all components with a consistent level of reliability or safety and to achieve minimal cost with optimal design parameters (Sørensen and Toft 2010). Although such an approach typically yields robust and economical designs, the full probabilistic approach can be complicated to designers, as the design standards provide no procedures for performing structural reliability analysis. Particularly, for the design of new structures, various uncertainties related to physical model, numerical model, loads, and resistances need be assessed. Thus, the probabilistic approach is more often used in research projects and calibration of partial safety factors (PSF).

The method suggested by international standards (IEC 2007, 2009; GL 2010, 2012; DNV 2014) is a semi-probabilistic approach which is a medium between the deterministic and probabilistic approaches. This is also the most common approach for performing code checks of offshore wind turbines. Using the semi-probabilistic approach, PSF for loads or materials are applied in the design checks of the LS. These PSFs are calibrated by a code committee based either on

statistical evaluation of experimental data, field observation (CEN 2002), or preferably, structural reliability methods (Ditlevsen and Madsen 1996). For loads and materials, Eqs. (4) and (5) can be used to calculate the characteristic values (IEC 2007).

$$S_d = \gamma_f \cdot S_k \quad (4)$$

$$R_d = \frac{1}{\gamma_m} \cdot R_k \quad (5)$$

where S_k is the characteristic value of the load effect, S_d is the design value of the load effect from given design load cases (DLC), R_k is the characteristic value of resistance, R_d is the design value of resistance, and γ_f and γ_m are the load and material factors, respectively. In practice, a set of PSF are applied to account for the uncertainties in different load effects and materials. Refer to Ronold et al. (1999), Ronold and Larsen (2000), Tarp-Johansen et al. (2002), Márquez-Domínguez and Sørensen (2012), and Sørensen and Toft (2014) for calibration of PSF of wind turbine components. In a wind turbine system, some parts are nonredundant and more critical than others. Based on the consequence of failure and component classes, additional PSF should be multiplied with the load effect. Note that the PSF should be applied in conjunction with the DLC. The design standards divide the type of design situations into three main categories. For normal scenarios such as power production and transport scenarios, relatively high PSF are specified. For abnormal scenarios including control or protection system faults, low partial factors are recommended. Three representative PSF from IEC 61400-3 are listed in Table 1. The classification society DNV GL recommends PSF for design of support structures and foundations for wind turbines according to a target annual probability of failure of 10^{-4} (DNV 2007).

Design of Renewable Energy Devices, Table 1 Partial safety factors for unfavorable loads (IEC 2009)

Type of design situation		
Normal	Abnormal	Transport and erection
1.35	1.1	1.5

Analytical and numerical design methods are useful. Model testing and full-scale tests, albeit at high expenses, can be important supplements to validate numerical codes and check unknown effects (Nielsen et al. 2006; Myhr et al. 2011).

Design Load Cases

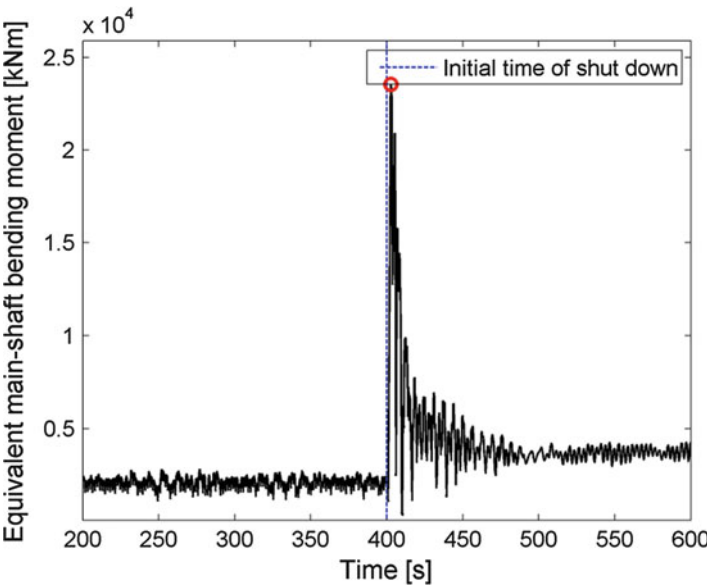
A set of DLC are specified by the design standards covering the most significant load conditions that a renewable energy device may experience during its lifetime. Table 2 summarizes the eight design situations recommended by the international

Design of Renewable Energy Devices, Table 2 Design load cases from IEC 61400-3 (IEC 2009)

Design situation	Type of analysis
1 Power production	Ultimate, fatigue
2 Power production plus occurrence of fault	Ultimate, fatigue
3 Startup	Ultimate, fatigue
4 Normal shutdown	Ultimate, fatigue
5 Emergency shutdown	Ultimate
6 Parked (standing still or idling)	Ultimate, fatigue
7 Parked and fault conditions	Ultimate, fatigue
8 Transport, assembly, maintenance, and repair	Ultimate, fatigue

electronical commission for offshore bottom-fixed wind turbines. Each situation is composed of a few cases of wave and wind conditions recommended for analysis. As shown, ultimate and fatigue analysis are required for most design situations including transport and maintenance. Note that some DLC are more clearly specified than others. The DLC related to faults (DLC 2 and DLC 7) are unclear as to what kind of faults should be considered and which structural components are concerned. Such cases, if ignored, may lead to nonconservative design of wind turbines. According to Jiang et al. (2013a), Jiang et al. (2015), and Etemaddar et al. (2016), DLC 2 can be a critical design situation. The transient responses due to pitch mechanism fault and shut-down can impose significant loadings on the main shaft. A time series of the main-shaft loading from numerical simulation is presented in Fig. 3. In comparison with the cases with power production plus fault (DLC 2), the parked conditions with pitch mechanism faults do not necessarily give greater structural responses than the parked cases (DLC 6) (Jiang et al. 2013b). The transport and assembly cases of wind turbine need be considered at the design stage, too. Still, there is limited research work on this aspect (Wang et al. 2014; Jiang et al. 2018). IEC 61400-3 specifies additional DLC covering ice loads if the wind turbine

Design of Renewable Energy Devices, Fig. 3 Time series of the main-shaft equivalent bending moment (Adapted from Jiang et al. 2013c)

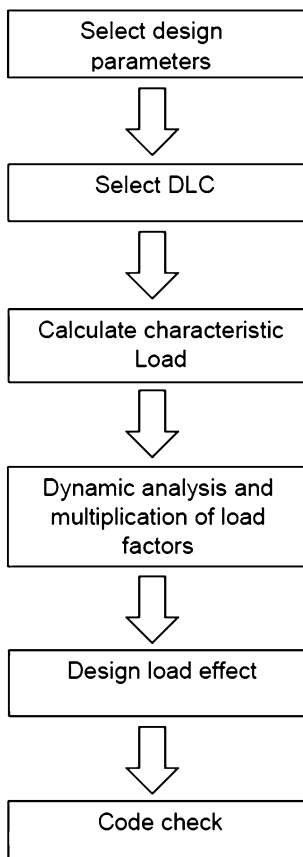


shall be installed at an offshore site affected by sea ice. As shown by Shi et al. (2016), ice loads can induce significant tower vibrations of monopile-type wind turbines.

There are special design considerations for floating renewable energy devices: intact and damaged stability, station-keeping capability, analysis duration of short-term simulations, and installation challenges, to name a few. Although relevant design standards have been proposed (ABS 2013; DNV 2013), the DLC for floating wind turbines have not been addressed explicitly.

Design Procedure

Figure 4 presents the design procedure of the semi-probabilistic method recommended by



Design of Renewable Energy Devices, Fig. 4 Flow-chart for semi-probabilistic design procedure. (Adapted from IEC 2009)

international standards. First, the design parameters are chosen possibly based on baseline models. Then, the recommended DLC will be used in the assessment of design load effects. Numerical modeling is often involved. Integrated dynamic analysis considering the simultaneous actions of wind loads, wave loads, and control actions (if any) is carried out for support structures including the wind turbine tower to determine the design load effect if the dynamic response is of primary concern. Finally, the design load will be compared against the structural resistance in the code check. If the general condition of Eq. (1) is not satisfied, more iterations are involved to modify the design parameters. The design problem can often be formulated mathematically as an optimization problem, and different numerical tools can be integrated to automatize the workflow. Design optimizations of renewable energy devices can be found in Hu et al. (2012, 2016), Kurniawan and Moan (2013), and Yang et al. (2015).

Cross-References

- [Analysis of Renewable Energy Devices](#)
- [Design Rules and Standards](#)
- [Offshore Wind Turbines](#)
- [Wave Energy Converters](#)

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Design of Submersibles

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Synonyms

Concept design; Design spiral; Detailed design; Empirical design; Hydrodynamic design; Multi-disciplinary design optimization (MDO); Optimal design; Preliminary design; Reliability-based design (RBD); Structural design; Technical design

Definition

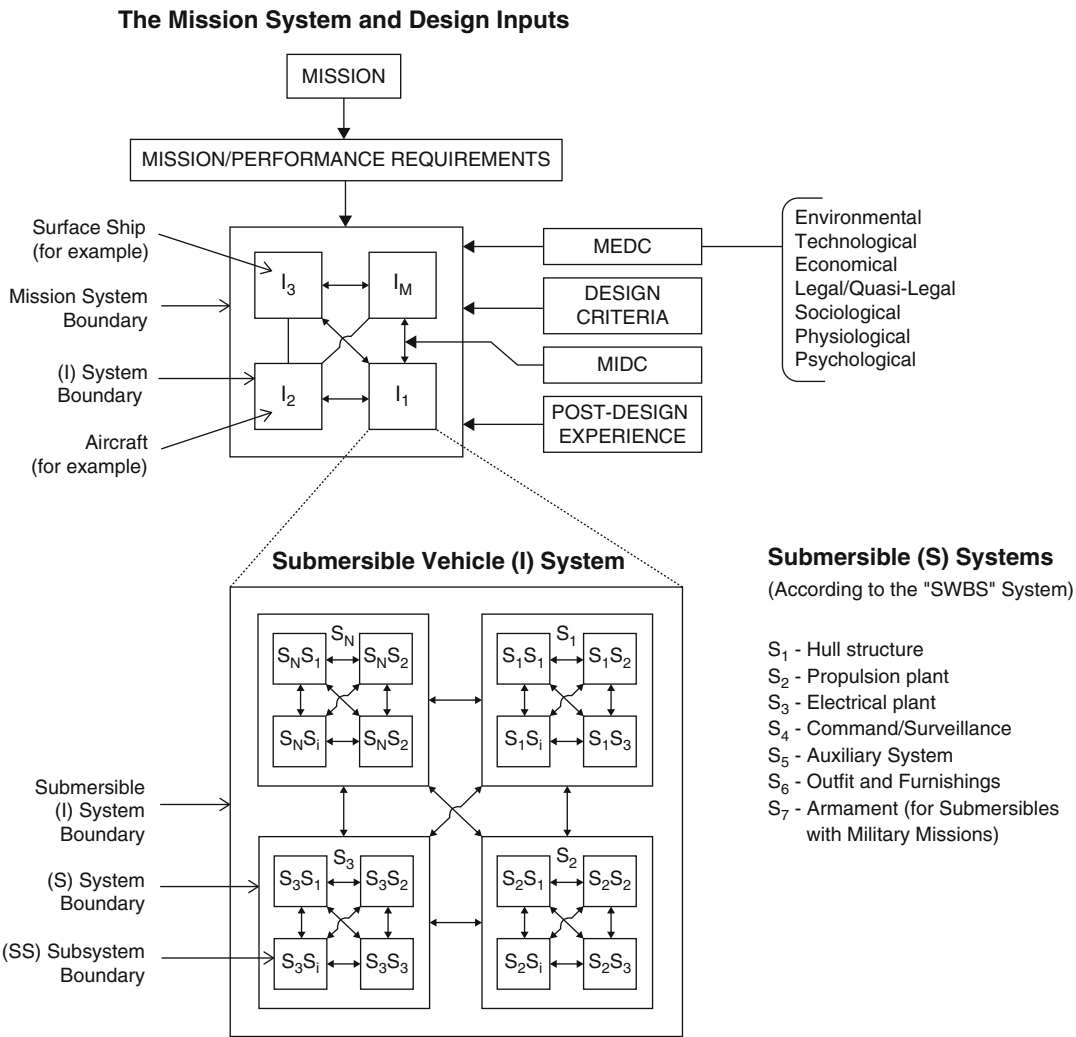
Design of a submersible is the creation of a plan for the construction of a submersible. Designing often necessitates considering the aesthetic, functional, economic, and sociopolitical dimensions of both the design object and design process. It may involve considerable research, thought, modeling, interactive adjustment, and redesign. More specifically, design is a specification of a certain type of submersibles which is manifested by a user and intended to accomplish certain missions in a particular environment using a set of primitive components, satisfying a set of requirements, subject to certain constraints.

Scientific Fundamentals

Basic Definitions About the Design Process

A mission system is any object or process, or group of objects or processes, created to serve some useful purpose or mission, the mission being defined by a set of mission requirements (Allmendinger 1990). It is often convenient to picture a system as isolated from its surroundings by a boundary for the purpose of studying its internal behavior and interaction with its surroundings. Figure 1 provides such a diagram, in this instance showing a mission system designed to serve one or more underwater missions. Note that the mission system boundary encloses M individual or (I) systems, each enclosed by its own boundary, illustrating the aforementioned fact that the mission is accomplished by a submersible supported by other systems. The mission system diagram in Fig. 1 provides an example of these (I) systems in which it is assumed that mission requirements dictate the needs, among others, for the submersible's transportation by air and on the surface of the sea as well as for its launching, recovery, and support while submerged. As shown, (I_1) is the submersible, (I_2) is a transport aircraft, (I_3) is a surface support ship, and the last individual system, whatever it may be, is designated (I_M) (Allmendinger 1990).

Mission system design or the design of any of its individual systems including the submersible generally includes the three processes: *pre-design*, *design*, and *post-design* processes. The predesign process involves the potential user/owner of the system to be designed and is concerned with the development of the primary input to the design process – the *mission statement* and *mission requirements*. The design process conceives the mission system that is feasible and meets the mission requirements in as optimal a manner as possible. It is divided into *basic*, *contract*, and *detail design* phases – basic design being subdivided into *conceptual* and *preliminary design* stages. Finally, the post-design process is composed of three activities which are discussed briefly – construction and *alteration* of “new” and “existing” individual systems of the mission system, test and *evaluation*, and *operation*.



Design of Submersibles, Fig. 1 The mission system and its subsystems (Allmendinger 1990)

Experiences gained from these activities form, collectively, important “feedback” input to the design of future systems.

The design of many mission systems begins with most of the potential (I) systems already existing and some nonexistent. Existing systems require from no to extensive alterations to convert them to (I) systems, within a particular mission system, with commensurate amounts of design effort, construction, and costs involved. Obviously, the extent of alterations required is a primary consideration in choosing between candidate (I) systems. Nonexisting (I) systems,

of course, must be designed and constructed “from scratch” and are often the most costly of these systems.

The design of any (I), (S), or smaller system also begins with at least some of its individual systems existing. These existing systems, requiring no or small modifications, are often referred to as “off-the-shelf” items. Their use significantly reduces costs.

Design of any mission system, and not necessarily any of its individual (I) systems, should achieve the design goal of *satisfying a given set of mission requirements in as optimal a manner*

as possible considering the design constraints acting. This statement introduces two new topics of *design optimization* and *design constraints*. Optimizing a mission system's design is usually based on maximizing its *cost-effectiveness* (CE) to the extent possible given the constraints acting. This process may vary from relatively simple to complex depending on the nature of the mission system. In general, it requires extensive economics and experiential data and the use of sophisticated techniques.

The design of the mission system should lead to the selection of its (I) systems and their characteristics which tend to reduce the number of unproductive days. In this regard, important considerations include (1) basic design philosophy, (2) speed of transit to, from, and between stations or operating areas, (3) endurance of the support system, (4) maintenance and repair facilities, and (5) limiting weather/sea-surface conditions for operation and survival.

Consideration (1) focuses on the virtues of keeping (I) system components as simple and rugged as possible, thereby promoting system safety and reliability. As will be seen, this philosophy encourages the use of the design criterion that designs, particularly of critical systems, be based on state-of-the-art technologies – that is, on proven technologies. The contributions of this philosophy to increasing 5 days per year, as well as to operational safety, are obvious.

Consideration (2) focuses on the speed requirement for the (I) system providing transportation usually a surface ship. This speed should be as high as feasible, and the ship's seakeeping characteristics should permit its maintenance under reasonably adverse sea conditions. This consideration may also lead to the choice of more than one system for transportation, particularly if mission requirements emphasize high in-transit speeds or short response times. The use of an aircraft to fly a submersible and associated gear to a distant port to be placed on board a surface ship is a case in point.

Consideration (3) involves the ability of the support system to remain at sea for a specified length of time, which is usually given as a mission or performance requirement. It is concerned with characteristics such as fuel, freshwater, and provisions and stores capacities and habitability. Habitability has to do with all aspects of design relating to human comfort and is directly related to endurance of personnel and the level of work performance. It increases in importance as endurance increases.

Consideration (4) includes facilities that are vital to the success of the mission as well as to reducing the unproductive, on-station time facilities that maintain the submersible in a state of operational readiness. They may be an integral part of the support system or another (I) system closely associated with the submersible. In the latter instance, these facilities are housed in portable vans which are outfitted at the shore facilities and secured to the deck of the support ship for the cruise. In both instances, the facilities include personnel trained in specific areas of technology such as mechanical and electronic technicians. These persons, including the pilot, are referred to as the "submersible's crew" to differentiate them from the "ship's crew" and other groups of personnel on board.

Consideration (5) involves two environmental factors: weather conditions, with visibility and wind forces being principal concerns, and sea-surface conditions as measured by sea-state numbers from 0 to 9, severity of conditions increasing as these numbers increase. One or both factors, if adverse, can cause a serious increase in unproductive in-transit and on-station time as well as endangering survival of any transport-support system subjected to these elements. Consequently, the design or choice, or both, of (I) systems should consider ways and means of reducing this unproductive time to the extent possible. The three (I) systems primarily involved are the *transport-support*, *submersible*, and *launch-recovery* system.

There are numerous design constraints imposed on the mission system and its individual systems. They may be placed into one of three categories: *mission external design* constraints (MEDCs), *mission internal design* constraints (MIDCs), and *design criteria* (DC).

Mission external design constraints (MEDCs) are the constraints lying outside, or external to, the mission system boundary, indicating that they are independent of all mission considerations: that is, they exist irrespective of the mission system in question. MEDCs imposed on system design may come from a few or several of the following sources: (1) environmental, (2) technological, (3) economical, (4) legal/quasi-legal, (5) political, (6) sociological, (7) physiological, and (8) psychological. The designer's control over MEDCs varies from none to considerable.

Mission internal design constraints (MIDCs) are the constraints lying inside, or internal to, the mission system's boundary or inside the boundary of any (I) or lower-level system. These sources are the interactions between the systems that influence each other's designs.

Design criteria (DC) are the criteria which furnish standards for judgments regarding design procedures and for selecting system components. The use of specified structural formulations and the requirement that critical system components be based on state-of-the-art technology are examples. The basis for optimization, either cost-effectiveness or minimum weight, may also be considered as a design criterion. These criteria may be generated by the user/owner, the designer, or classification certification bodies.

Basic Introduction to the Design Process

The design and associated processes are diagrammed in Fig. 2 in chronological order as the (1) predesign, (2) design, and (3) post-design processes. In this section, they are introduced briefly from the perspective of the mission system to provide an overview of the entire subject.

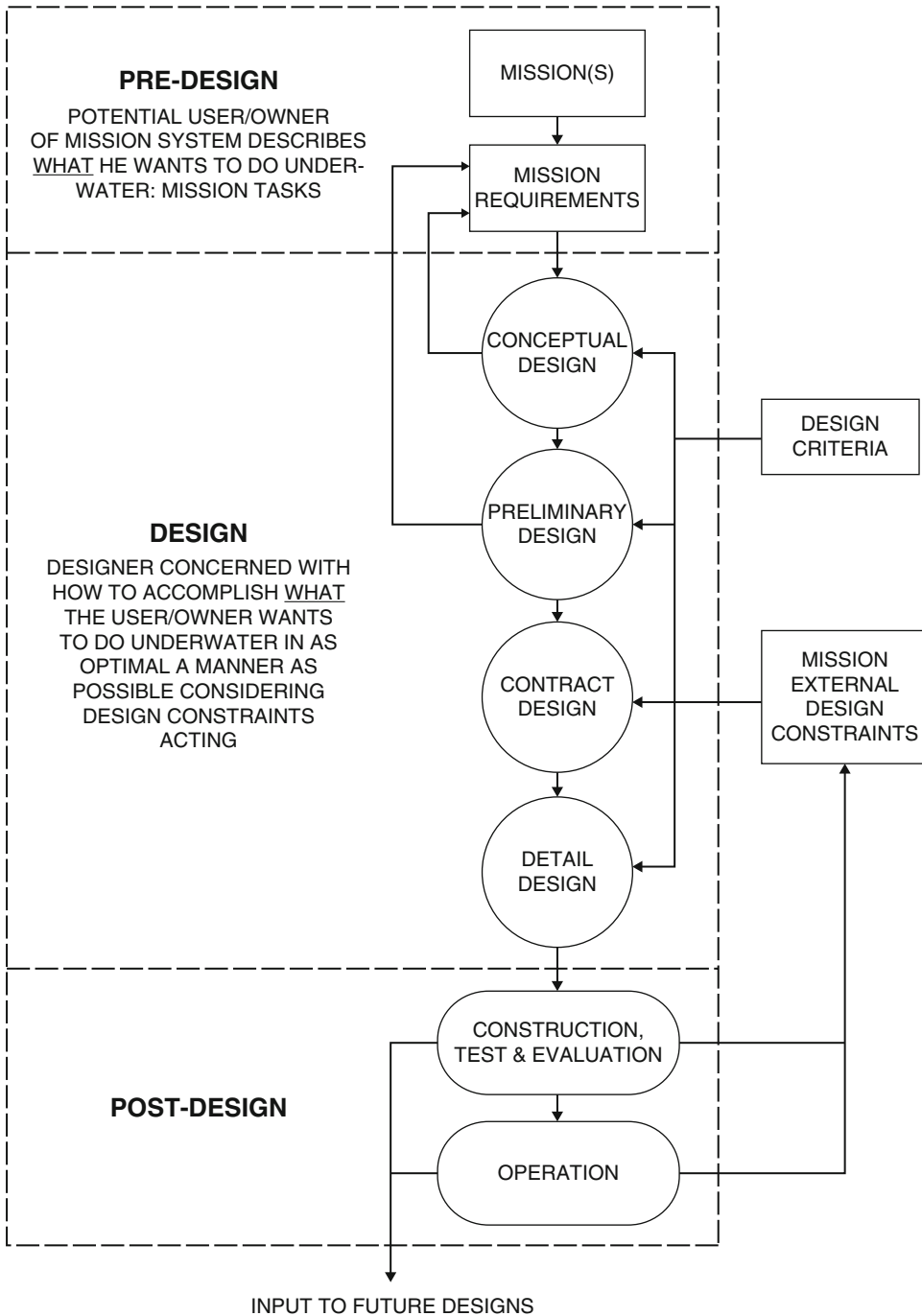
The predesign process leads to the formulation of the basic input to the design process the mission (s) statement(s) and associated mission *requirements*. It is undertaken by the potential user or owner of the mission system to be designed, independently or in consultation with the system's designer.

The mission is a short, concise statement of what the user/owner wants the submersible to accomplish underwater. Missions may be categorized in several ways. The most general way is by the titles of the user community being served: (1) industrial/commercial, (2) military, (3) scientific, or (4) recreational. A subcategorization may be based on terms describing the general nature of the underwater tasks to be performed, for example, (1) *inspection*, (2) *survey*, (3) *monitoring/sampling*, (4) *construction*, (5) *maintenance/repair*, (6) *search*, (7) *salvage*, (8) *rescue*, and so forth. Mission system design may be based on single or multi-mission requirements, the latter including two or more categories of underwater tasks.

The design process is composed of three design phases: (1) *basic design*, (2) *contract design*, and (3) *detail design*. The levels of labor, preparation time, and costs associated with each phase increase exponentially as the process progresses. Design creativity and flexibility are essentially limited to the basic design phase – the mission system, including its submersible system, being well defined at the beginning of contract design.

Basic design is the primary design phase concerned with *how best* to accomplish *what* the potential user/owner of the mission system wants to do underwater as set forth by his mission requirements. In general, this phase involves:

1. Development of performance requirements, or (I) system capabilities, from the mission requirements
2. Determination of the principal characteristics of the (I) systems required to achieve these capabilities, which enable new construction systems to be designed and existing systems to be selected and altered if necessary



Design of Submersibles, Fig. 2 Design and associated processes (Allmendinger 1990)

3. Estimation of capital and operating costs of (I) systems
4. Identification of one or a few mission systems from among a series of alternatives which optimally meet performance requirements to the extent possible considering design constraints acting.
5. Refining and firming up characteristics and cost estimates of the (I) systems composing the optimum conceptual design(s)

The *conceptual design stage* of basic design is concerned with the first four of these functions – function four, herein, is based on cost-effectiveness as the optimization criterion. The *preliminary design stage* of this phase is concerned with the fifth function.

Conceptual design, diagrammed in Fig. 3, is the first attempt to translate all the user/owner's mission requirements into performance requirements and characteristics of the (I) systems composing the mission system. It is often viewed as consisting of *feasibility studies* and *completion* of the *optimum conceptual design(s)* once they are identified by these studies. Feasibility studies are concerned with the development of those performance requirements and associated system characteristics which have significant impact on the cost-effectiveness optimization process. Completion of the optimum design(s) involves developing other major performance requirements and characteristics those which are essential in defining (I) systems but which do not have significant impact on the optimization process.

Feasibility studies create an orderly series of mission system alternatives, all of which meet the mission requirements and are technically feasible, and select one or a few of these alternatives as the conceptual design(s) to be carried on into preliminary design. These alternatives are shown in Fig. 3 as $MS_1, MS_2, \dots MS_i$, where "i" can be any small to large number depending on the simplicity/complexity of the mission and optimization techniques used. As indicated in the figure, each alternative is composed of a unique combination of (I) systems. Each (I) system, in turn, has unique capabilities, characteristics required to achieve these capabilities, and costs associated with providing these characteristics. Certain unique (I) systems may appear in more than one combination, but individual combinations are not duplicated in other alternatives. Mission system alternatives may be generated by (1) varying performance requirements with the cascading effect of varying (I) systems' capabilities, characteristics, and costs and (2) varying (I) systems' capabilities, characteristics, and costs in ways each of which meet a specific set of performance requirements. Alternatives generated in this manner contain information necessary to conduct the

search for the optimum, cost-effective mission system(s).

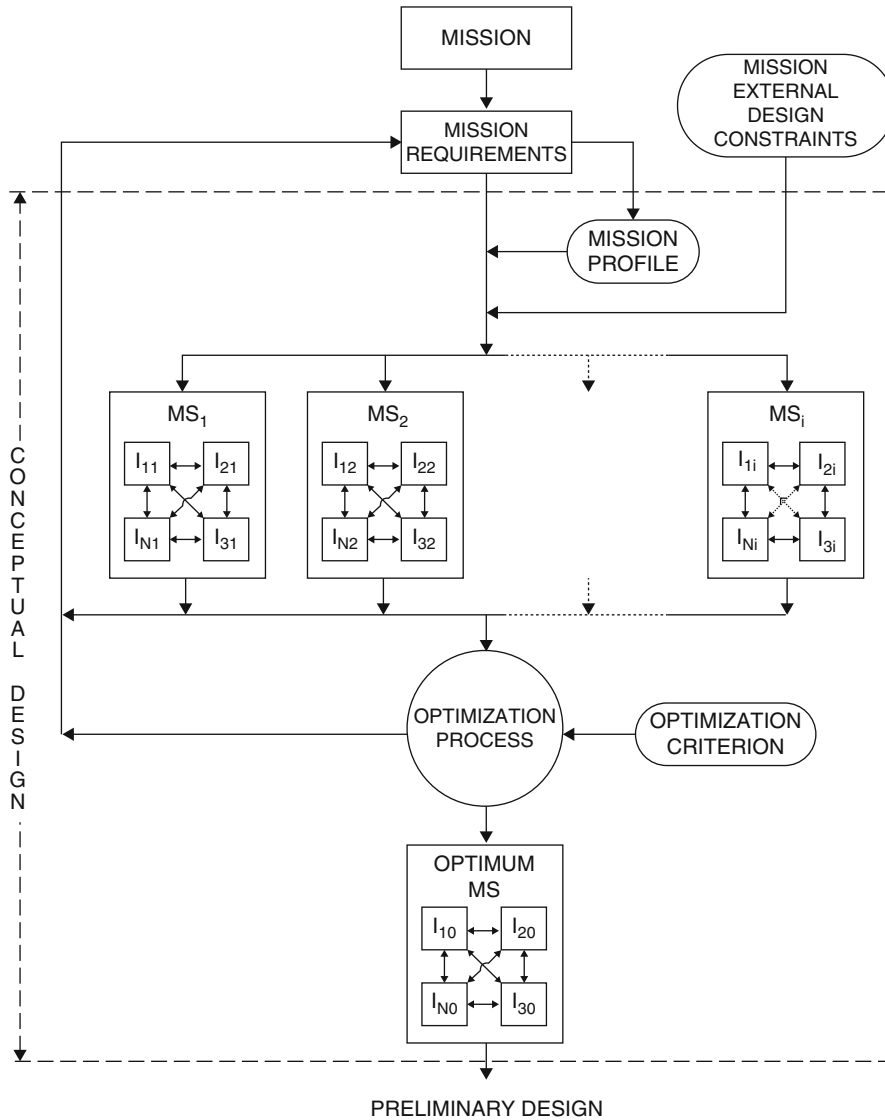
Combinations of (I) systems forming mission system alternatives may vary in *number*, *type/capabilities*, and *status*. These features are related to functions required to accomplish mission tasks, the time frame for completing these tasks, if critical, and economic considerations. A *mission profile*, derived from mission requirements, is useful in providing functions and time frame information – its usefulness increasing as the complexity of the mission increases. This profile is a chronological listing of all events occurring from the mission's beginning to end and including time frame data where necessary.

Preliminary design is the second stage of basic design and is concerned with this phase's fifth function. Starting with baseline data provided by the selected conceptual design(s), it refines and firms up the major characteristics of (I) systems, the MIDCs they exert on each other, and cost estimates associated with them. In summary, it provides a precise engineering definition of the mission system and assurance that the design goal can be attained.

Contract design requires yet further refinement of design and additional detail. It yields contract plans and specifications necessary for interested parties to bid on the construction of new (I) systems or the alteration of existing (I) systems. It also provides contractual documents for the construction and alteration work.

Specifications delineate quality standards of material and workmanship as well as setting forth performance expectations for the (I) systems and their subsystems. They also describe tests and trials which must be performed successfully before acceptance of the systems.

Detail design is the final phase of the design process and entails the development of detailed working plans from which the (I) systems are constructed or altered. In one sense, it is really not a design phase since all the creative design effort is done in the preceding phases, the design being unequivocally defined prior to entering this phase. It does, however, require the greatest amount of work of all the phases and is often undertaken by entities building or altering the system.



Design of Submersibles, Fig. 3 Conceptual design of mission system (Allmendinger 1990)

The post-design process is composed of three activities: construction/alteration, test and evaluation, and operation. Although this publication is not directly concerned with these activities, there are two reasons for including them in this overall perspective: (1) They constitute important sources of input to new designs and feedback to the design in question, and (2) they are subject to certain existing MEDCs and will influence the future impact of MEDCs on following designs.

The above introduction mostly came from Allmendinger (1990) which was based heavily on Busby (1976). More detailed introduction can be found from Busby (1990) and Funnel (1999).

Historical Development of Submersible Design

Design of shallow water submersible may be started at very early days. While it is difficult to pinpoint the advent of the first submersible,

Askew (2001) attributed this to Cornelius van Drebbel who constructed a vehicle in 1620 under contract to King James I of England. It was operated by 12 rowers with leather sleeves, waterproofing the oar ports. It is said that the craft navigated the Thames River for several hours at a depth of 4 m and carried a secret substance that purified the air, perhaps soda lime? However, Kohnen (2009) attributed the first “submarine” to the American Turtle, built in 1772 by David Bushnell during the American Revolution. This particular submersible did not have any practical achievement; however, it is the first to have all systems needed for making a submarine, ballast and trim control, tubes for ventilation, and vertical and horizontal propulsion, and demonstrates classic ingenuity paired with military motivation. However, design of the deep manned submersible came from the idea of William Beebe and Otis Barton’s bathysphere which is actually not a submersible (Hotta et al. 2001). Barton and Beebe set a world record on August 15, 1934 near Bermuda by descending with their bathysphere to a depth of 923 m (Beebe 1935). Basically, it was just a steel sphere with one entry hatch and three large (8.0 in.) windows. These are made of fused quartz, 3 in. thick which were the largest pieces ever made at the time. The use of a cable to lower down the sphere provided many limitations such as depth and maneuverability. Kohnen (2009) pointed out that the following features of the bathysphere were unattractive:

1. The bathysphere was lowered by a tether which could cause discomfort by surface motion of the mothership (Piccard 1956).
2. The possibility of cable breakage caused by load-induced strain (Piccard 1956).
3. The quartz material for the windows was brittle and could fracture (Piccard and Dietz 1961).

The bathysphere cannot be described as a deep manned submersible since it does not have a moving ability, and due to self-weight of the cable, the maximum depth of deployment was only around 3000 m. In order to penetrate deeper in the ocean, people needed to seek a solution without a cable.

This problem was overcome by August Piccard who applied a balloon principle to the design of deep manned submersible. The Swiss-designed but mostly Italian-funded Trieste was an important bridge between the early US submersibles and the American-built deep submersibles that followed its purchase and operation by the US Navy. The catalytic effect of the Navy Trieste program was a major factor in the initiation of deep submergence technology in the United States. On January 23, 1960, the bathyscaph *Trieste*, piloted by Jacques Piccard and Don Walsh, made one successful dive to the depth of 10,913 m of the Challenger Deep. The details of this exploration can be found in many articles (e.g., Walsh 2009; Forman 2009; Kohnen 2009) and even books (e.g., Piccard and Dietz 1961; Busby 1976; Ballard and Hively 2002).

The technical achievements made in both *Trieste* and *Archimede* were great. These included (Kohnen 2009) (1) the underwater balloon principle that is still used even today, (2) the window design of acrylic that has become more or less the standard, and (3) a full ocean depth manned cabin for three persons. However, their drawbacks were also obvious (Kohnen 2009): (1) it was heavy and bulky; (2) it had poor maneuverability; (3) there was danger of fire when pumping gasoline in and out; and (4) swimmers did not enjoy when gasoline leaked into the water.

Due to the occurrence of solid buoyancy material (Allen 1955), the manned submersible became much smaller. Alvin was the most famous (Kaharl 1990; WHOI 2018), while other large depth manned operational submersibles include Nautille of France (Lévêque and Drogou 2006), MIR I and II of Russia (Sagalevitch 2009), Shinkai 6500 of Japan (Takagawa 1995), and Jiaolong of China (Cui 2013).

Since the 1960s, various other types of underwater submersibles were developed such as remotely operated vehicles (ROVs) (Christ and Wernli 2014), autonomous underwater vehicles (AUVs) (Wynn et al. 2014), hybrid ROV and AUV (HROV or ARV) (Bowen et al. 2004), gliders (Rudnick 2015), floats (Richardson and John 2001), etc., and their design philosophy can be found from the references provided.

Similarly, design method is also progressing. Earlier design was mostly experience-based and carried out with a design spiral in series. This is called empirical design. Nowadays, design is carried out more toward the first principle based, and very frequently the uncertainties embedded in the design process are explicitly considered. This is termed reliability-based design. In parallel, the optimization method is progressing, and the optimal design is achieved more globally than previous optimization

methods by introducing the idea of multi-disciplinary design optimization (MDO). For interested readers, one can refer to Pan and Cui (2018).

Key Applications

Submersibles have many uses worldwide, such as oceanography, underwater archaeology, ocean exploration, adventure, equipment testing, maintenance and recovery, and underwater videography. The application of submersibles has been introduced in submersible entry, and here we only emphasized two most important uses.

New Scientific Discovery

Between 1973 and 1974, project FAMOUS (French-American Mid-Ocean Undersea Study) was conducted in the Mid-Atlantic Ridge off the Azores using the French bathyscaph *Archimede* and the US submersible *Alvin*. The project was the first systematic and successful use of deep submersibles for science. They discovered and sampled fresh pillow lavas and lava flows at 3000 m deep in the rift valley, where the oceanic crusts were being created, providing



Design of Submersibles, Fig. 4 Extraordinary, unexpected life forms discovered by *Alvin* in the Galápagos Rift in 1977 (www.whoi.edu)

Design of Submersibles, Fig. 5 The B28RI nuclear bomb, recovered from 2,850 ft (870 m) of water, on the deck of the USS *Petrel*. (Picture from https://en.wikipedia.org/wiki/1966_Palomares_B-52_crash)



visual evidence of plate tectonics (WHOI 2018). In 1977, *Alvin* discovered a hydrothermal vent and vent animals in the East Pacific Rise off the Galápagos Islands at a depth of 2450 m. Discovery of these chemosynthetic animals, which were not dependent on photosynthesis, had a profound impact on biology in the twentieth century (Fig. 4) (Corliss et al. 1979).

Salvage of H-Bomb

In 1966, the US Navy called in *Alvin* to help find a hydrogen bomb accidentally dropped into the Mediterranean Sea. *Alvin* locates the bomb, and after the bomb skids down a slope on the first recovery attempt, it finds it again at a depth of 2,900 ft (880 m). Then, an unmanned torpedo recovery vehicle, CURV-I, became entangled in the weapon's parachute while attempting to attach a line to it. A decision was made to raise CURV and the weapon together to a depth of 100 ft (30 m), where divers attached cables to them. The bomb was brought to the surface by USS *Petrel* (Fig. 5).

Cross-References

- Concept Design
- Design Spiral
- Detailed Design
- Empirical Design
- Hydrodynamic Design
- Multidisciplinary Design Optimization (MDO)
- Optimal Design
- Preliminary Design
- Reliability-Based Design (RBD)
- Structural Design
- Technical Design

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- Serve as a contractual reference document between suppliers and purchasers
- Serve as a guideline for designers, suppliers, purchasers and regulators

And it usually gives requirements for the following:

- Design principles
- Structural categorization
- Material selection and inspection principles
- Design loads
- Load effect analyses
- Design of steel structures and connections
- Corrosion protection
- Foundation design

For marine structures, a design rule or standard is mainly developed by classification societies, which is a third-party non-governmental organization that establishes and maintains technical standards for the construction and operation of ships and offshore structures. Presently the main classification societies in the world include: Lloyd's Register (LR, England), Bureau Veritas (BV, France), Registro Italiano Navale (RINA, Italy), American Bureau of Shipping (ABS, USA), Det Norske Veritas & Germanischer Lloyd (DNV GL, Norway), Nippon Kaiji Kyokai (NKK, Japan), Russian Maritime Register of Shipping (RS, Russia), China Classification Society (CCS, China), and Korean Register of Shipping (KRS, Korean).

In addition, some design rules or standards developed by international or national organizations, committees, and industry associations are also commonly used, e.g., ISO, IEC, Eurocode, NORSOK, etc.

The present chapter mainly addresses rules and standards related to structural design of ocean energy devices. The substructures for offshore wind turbines, wave energy converters, and tidal energy converters are considered.

Design Optimization

► Optimal Design

Design Rules and Standards

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Introduction

A design rule or standard for marine structures provides principles, technical requirements, and guidance for the structural design and the design verification of load-bearing structures subjected to foreseeable actions in marine environments. A design verification represents the confirmation for compliance of a structural design with defined requirements. Marine structures usually include ship structures and offshore structures. When designing a unit where an object standard exists, the object standard for the specific type of unit shall be applied. The main objectives of a design rule or standard are to:

- Provide an internationally acceptable level of safety by defining minimum requirements for structures and structural components (in combination with referred standards, recommended practices, guidelines, etc.)

Design Rules and Standards for Offshore Wind Turbines

The integrated design of an offshore wind turbine and its substructures is in its growth stage.

Reducing its levelized cost of energy (LCOE) is still challenging. How to match the design methods for wind turbines' aerodynamic load & response and the design methods for conventional offshore structure, and refine them, are the key issues for a successful design of offshore wind turbines.

Presently the designer can essentially rely on the following standards/technical specifications for the design of bottom-fixed offshore wind turbines and their substructures:

- IEC 61400-3 (IEC 2009)
- DNVGL-ST-0126 (DNV GL 2018a)
- DNVGL-SE-0441 (DNV GL 2016a)
- ABS #176 (ABS 2013a)

and for the design of floating offshore wind turbines and their substructures:

- DNVGL-ST-0119 (DNV GL 2018b)
- DNVGL-SE-0441 (DNV GL 2016a)
- ABS #195 (ABS 2013b)
- BV NI 572 (BV 2015)
- ClassNK Guideline (ClassNK 2012)

Although there exist design standards for floating wind turbines, the industry is still in an early stage of development; therefore further development and revision of these standards is necessary when the relevant industry experience is available.

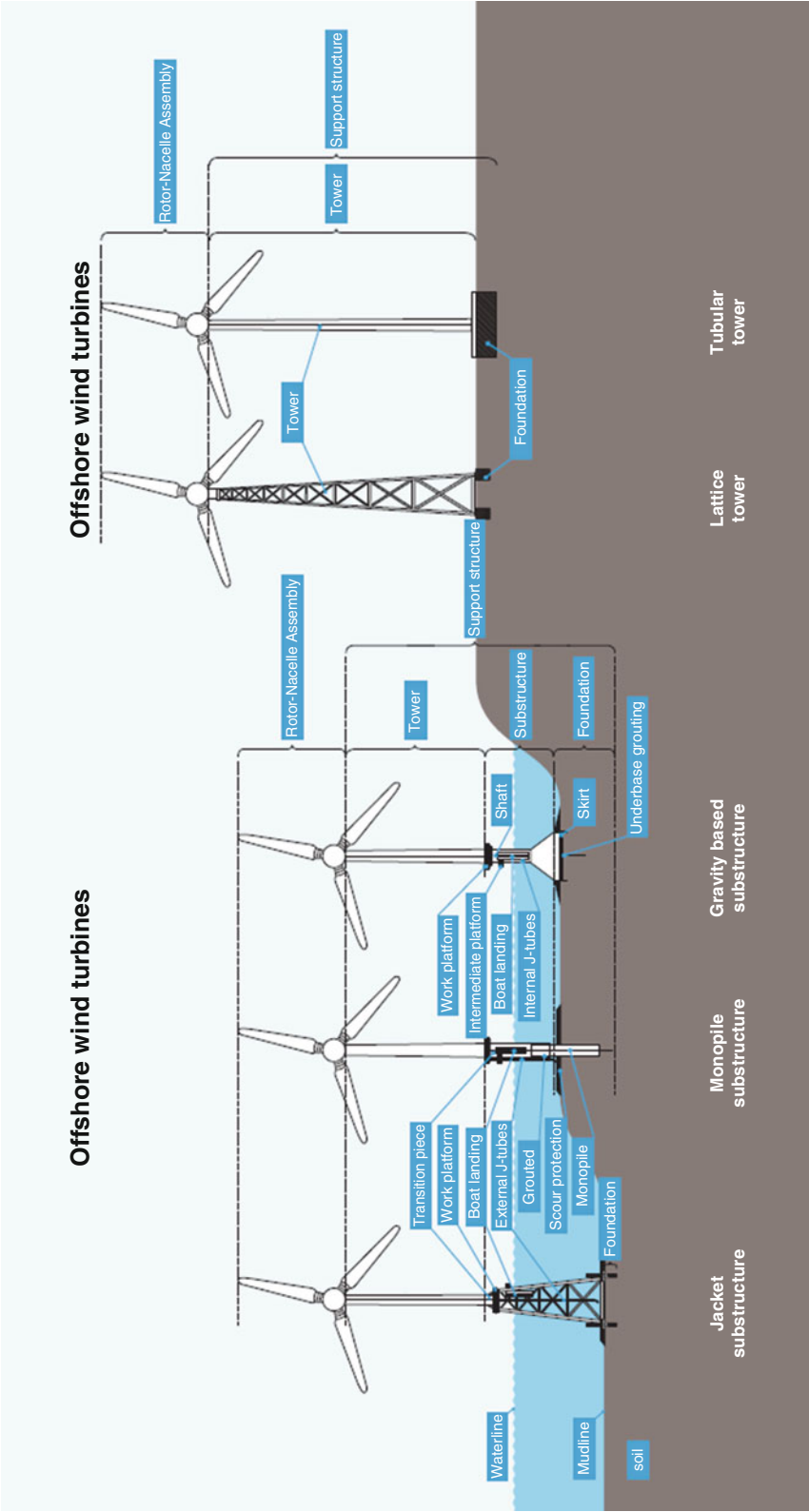
The most common rules to certify offshore wind turbines are the IEC 61400-3 rule and the DNV-GL guidelines. The current edition of the IEC 61400-3 rule considers bottom-fixed offshore wind turbines. In addition, a draft technical specification of IEC TS 61400-3-2 (IEC 2018) for the design of floating offshore wind turbines has been published by IEC, where the first edition of the technical specification will be published in the near future.

DNVGL-ST-0126 is the latest version of the DNV GL standard for guiding the design of support structures of wind turbines, which supersedes the May 2014 edition of DNV-OS-J101 (DNV 2014) and the April 2016 edition of DNVGL-ST-0126. The standard is applicable to all types of onshore and fixed offshore support structures

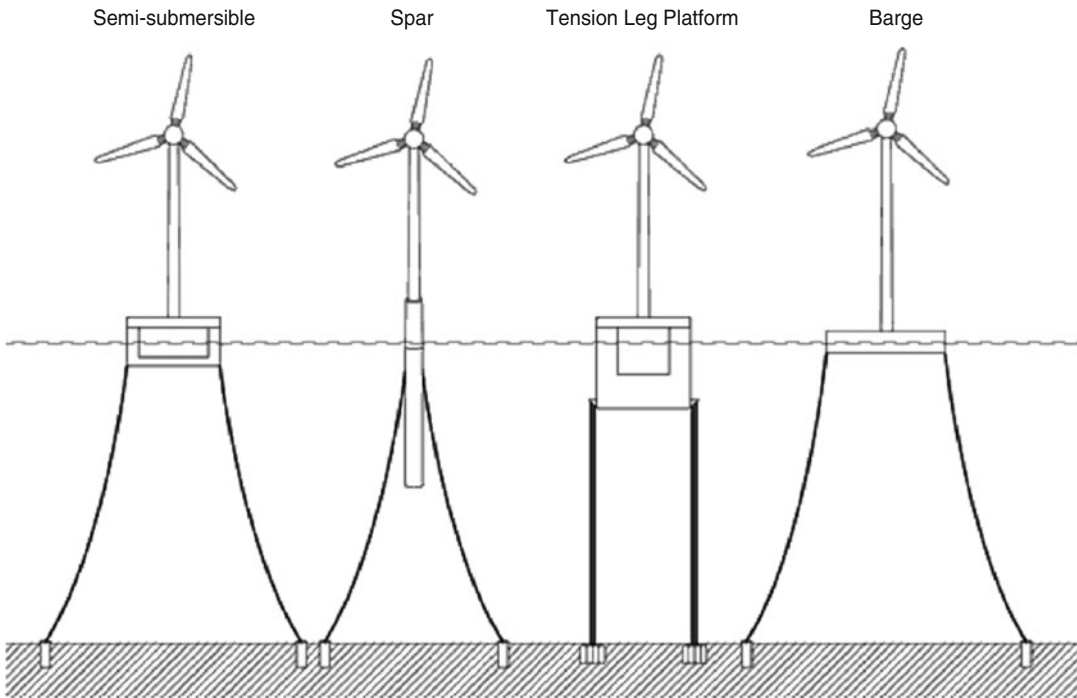
for wind turbines as shown in Fig. 1. It takes construction, transportation, installation, and inspection issues into account to the extent necessary in the context of structural design. Design loads and load combinations to be considered together with this standard shall be determined as described in DNVGL-ST-0437 (DNV GL 2016b). It does not cover design of support structures for substations for wind farms and design of wind turbine components such as nacelle, rotor, generator, and gearbox. For design of support structures for substations, such as converter stations and transformer stations, DNVGL-ST-0145 (DNV GL 2016c) applies. For structural design of rotor blades, DNVGL-ST-0376 (DNV GL 2015a) applies. For structural design of wind turbine components, DNVGL-ST-0361 (DNV GL 2016d) applies.

DNVGL-ST-0119 is the latest version of the DNV GL standard for guiding the design of the support structures and station-keeping systems for floating wind turbines, which supersedes the June 2013 edition of DNV-OS-J103 (DNV 2013). The standard is applicable to all types of support structures and station-keeping systems for floating wind turbines. Figure 2 shows four common floater types for wind turbines. It gives provisions for the floater motion control system and the control system for the wind turbine to the extent necessary in the context of structural design, no matter these systems are separate or combined. It also gives provisions for transportation, installation, and inspection to the extent necessary in the context of structural design. It does not cover design of wind turbine components such as nacelle, rotor, generator, and gearbox. For structural design of rotor blades, DNVGL-ST-0376 applies. For structural design of wind turbine components for which no DNV GL standard exists at present, the IEC 61400-1 standard (IEC 2005) applies.

DNVGL-SE-0441 is a new service specification for type and component certification of onshore and offshore wind turbines. The certification scheme is aligned with the IEC 61400-22 (IEC 2011) scheme for the technical requirements and the certificates. This specification together with referred DNV GL standards and recommended practices represents a completely revised and



Design Rules and Standards, Fig. 1 Definition of wind turbine components (DNV GL 2018a)



Design Rules and Standards, Fig. 2 Four common floater types for wind turbines (DNV GL 2018b)

expanded version of GL-IV-1 (GL 2010) and GL-IV-2 (GL 2012), both with a history of 30 years. It applies to type certification of wind turbines as well as for component certification.

The development for most of the other standards, both national and international, has been aligned with IEC 61400-1 and 61400-3. For example, ABS #176, also an international standard, issued first in 2010 and revised in 2013, is mostly aligned with IEC 61400-3. ClassNK published “Guideline for Floating Offshore Wind Turbine Structures” in July 2012, which applies to unmanned floating structures with a 20-year design life or over and to the tower and rotor nacelle assembly (RNA) as options. The guideline was developed referring to IEC 61400-1 and IEC 61400-3 and the related ClassNK Rules for the Survey and Construction of Steel Ships. The design principle provided in the guideline is to maintain the structural integrity of floating structures throughout their design lifetime without the need for docking and repair, and to consider the absolute minimum of damage stability and one line breaking in order to prevent drifting.

Design Rules and Standards for Wave Energy Converters

Presently there are no rules or standards for the structural design of WEC devices; however the rules, standards, and guidelines existing within the offshore engineering are useful for the design of WECs.

Design of wave energy converters (WECs) is still in its infancy, when compared to wind turbines, solar panels, and even tidal turbines, so it faces significant hurdles to achieve commercial applications. Different concepts are being investigated with different stages of development in the world, which range from concepts to full-size prototype devices and early commercial units. It is common that each WEC has its own method of operation, resulting in different amounts of power production in different wave patterns. In addition, it should be noted that WECs generally differ from other renewable energy devices in terms of the relation between maximum power output and average power output. Specifically, under design environmental conditions, wind, tidal, or solar

power devices will produce consistent power at their rated maximum, while for WECs the average power production will generally be much less than the maximum rated power. This can be misleading to estimate the annual power output for a WEC through the rated power. Up to now, most device developers of WECs rate their devices through the maximum power output (SI-Ocean 2012).

In order to assess the different technologies on a level playing field to allow for a fair and standardized comparison, and develop a uniform method to evaluate both a WEC and the location of installation for different sectors of the industry, the establishment of standards is important for WECs development, and significant efforts are currently underway. Several international collaborations have been initiated to establish rules and standards for rating the power of WECs, which should lead to standardized methodologies for assessing different devices so that fair comparisons can be made. Moreover, guidelines, rules, and standards should also provide detailed procedures and methodologies for design, testing, and operation of wave energy converters.

The International Electrotechnical Commission (IEC) established the Technical Committee (TC) 114 on Marine Energy: Wave, Tidal and other Water Current Converters in 2009. Through a collaborative international effort, a set of new technical specifications (precursors to standards) for the emerging field of marine energy systems have been published since 2011, and more tasks are being carried out by the IEC-TC-114 (2014). Two most advanced technical specifications (TS) for resource assessment and mooring system of WECs were published in 2015:

- IEC TS 62600-101 (IEC 2015a)
- IEC TS 62600-10 (IEC 2015b)

CarbonTrust (2005) commissioned DNV GL for preparing a “guideline on design and operation of WECs” in 2005. The main objective of this document is to provide interpretation and guidance on the application of the various existing standards and guides to WECs devices and it included many aspects on design and operation of WECs. It should be noted that this document

should be regarded as a guideline instead of a standard.

DNV (2008) published the principles and procedures for certification of tidal and wave energy converters in 2008 and made two amendments in 2011 and 2012. The document is built on experience and relevant standards in the oil and gas industry, but also includes the best practices in this particular area. It is supposed to cover all types of tidal and wave energy converters.

In general, the technologies for wave energy conversion are still immature, and many devices are in different stages of development, while new concepts are continuously being proposed. To guide the rational development of WECs, some best practices have been recommended. Unlike the standards, the best practices are regarded as guidelines on how to transfer WECs from the concept stage to the commercial application, and avoid the most common failures.

The design basis and the cost basis for wave and tidal energy systems are discussed in a document entitled “Development of Recommended Practices for Testing and Evaluating Ocean Energy Systems, Summary Report” published by Nielsen (2010), which follows the Implementing Agreement for a Co-operative Programme on Ocean Energy Systems (OES-IA) created by the International Energy Agency, Annex II. Similarly, many deliverables from the EquiMar project have been published on its website <http://www.equimar.org/>, where two of the deliverables with the topics “Best Practice for Tank Testing of Small Marine Energy Devices” and “Protocols and Guidance for Device Specification and Quantification of Performance” may be more relevant for WECs. Other relevant information can also be found from the MariNet project (http://www.fp7-marinet.eu/joint-activity_standardisation_best-practice.html) and the EMEC website (<http://www.emec.org.uk/standards/>).

Design Rules and Standards for Tidal/Ocean Current Turbines

The development of the technologies for tidal/ocean current turbines falls between that of offshore wind turbines and that of WECs, where a

full commercialization is possible. The commercial impacts on developers become greater as the size of turbines and scale of planned arrays increase. Therefore more attention is paid to the reduction of risk and the processes through which risk is controlled and mitigated, which should be the core in the development of new standards/technical specifications (TS) in the future. In addition, a clear structure and guidance on how to move from prototype through to type and project certification should be also set clearly in new standards/technical specifications, keeping the elements of the risk-based approach used in technology qualification and their earlier editions.

Presently the designer can essentially rely on the following standards/technical specifications for the design of tidal/ocean current turbines and their substructures:

- IEC TS 62600-201 (IEC 2015c)
- DNVGL-ST-0164 (DNV GL 2015b)
- DNVGL-SE-0163 (DNV GL 2015c)
- BV NI 603 (BV 2013)

Although there exist design standards for tidal/ocean current turbines, the industry is still in an early stage of development; therefore further development and revision of these standards is necessary when the relevant industry experience is available.

IEC TS 62600-201 is a new technical specification (TS) published by IEC Technical Committee 114 (IEC TC-114) in 2015. It concerns methods and procedures for determining, objectively and reliably, the scale and character of the tidal energy resources in a region. The documentation is intended to become international standard, once sufficient industry experience in its application has been developed.

DNVGL-ST-0164 is a new DNV GL standard for guiding the design, construction, and in-service inspection of tidal turbines. The documentation covers design and construction of structures (including blades, rotor, nacelle, supporting structure, and foundations), machinery, safety, controls and instrumentation, and electrical systems. It takes transportation, deployment, retrieval, and inspection issues into account to the extent necessary

regarding their impact in the design and construction aspects.

DNVGL-SE-0163 is a new service specification for prototype, type, project, and component certification of tidal turbines or tidal turbine arrays. It replaces the DNV GL standards DNV-OSS-312 (DNV 2008) on the tidal subject and GL IV-14 (GL 2005) on the ocean current turbine subject. The documentation covers all types of turbines and their support structures, fixed or floating, and all types of substation (s) including support structure(s), power cables, and subsea connectors. It also covers the project certification of an array of tidal turbines.

Generally, the technologies of tidal/ocean current turbines are becoming mature. The commercial impact on developers will play an important role in the future. The key to further development of the industry is the reduction in cost and the control of risk. For tidal turbine developers, the new standards/technical specifications should provide a clear scope for type certification which is key to open up commercial opportunities for volume production. Project developers themselves will be interested both in what a “type-certified” turbine means but also in certifying their own projects to satisfy their investors and insurers that they have controlled their risks, which should be also reflected clearly in the new standards/technical specifications.

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Design Spiral

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Synonyms

Concept design; Contract design; Detailed design;
Empirical design; Optimal design; Preliminary
design; Sequential engineering; Technical design

Definition

Design spiral mostly describes the construction process of submersible design, including AUV, ROV, HOV/HROV, ARV/MS, Glider, Deep Tow

System, Lander, ADS, DSRV, Rescue Bell, and Deep float design. Design spiral is often used in preliminary design process of submersible up to submersible tender. It elaborates sequence of design steps and uses sequential cyclical approach and iterative process with increasing complexity to result in single feasible submersible with economically reliable tender. It is rule-based and is subject to univariate systematic variation with constraints.

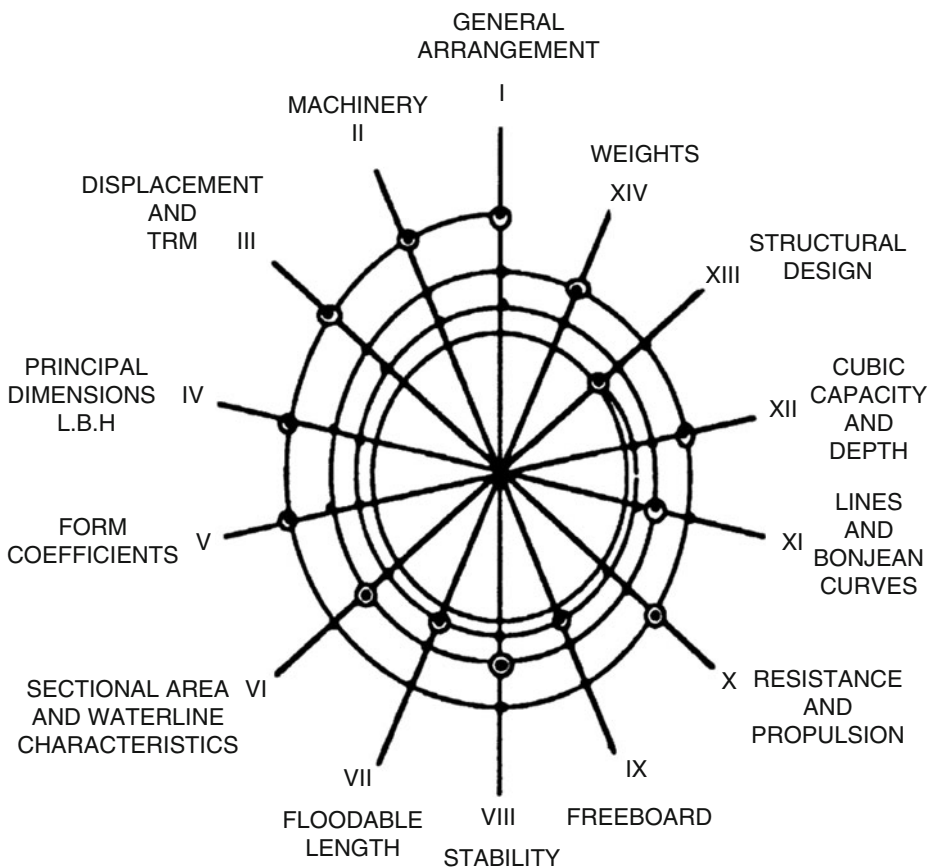
Scientific Fundamentals

Basic Definitions About the Design Process

Evans introduced the concept of design spiral in 1959, which represents the sequential and iterative aspects of the process.

Spiral is used to depicts sequential, cyclical decision process, which uses Class rules as basis and can result in optimal, low weight submersible structure. At every intersection of a spoke with a spiral, a calculation must be performed to update the structure, thus can lead to a feasible design. In the early stage of submersibles design, spiral is often used. It elaborates sequence of design steps and uses sequential cyclical approach and iterative process with increasing complexity. Spiral design is an iterative process, which is rather difficult to capture in a model. The design spiral in Fig. 1 is a traditional way of “capturing” the ship design process. As the design progresses, it converges to a single point. Each baseline is a new iteration of the design.

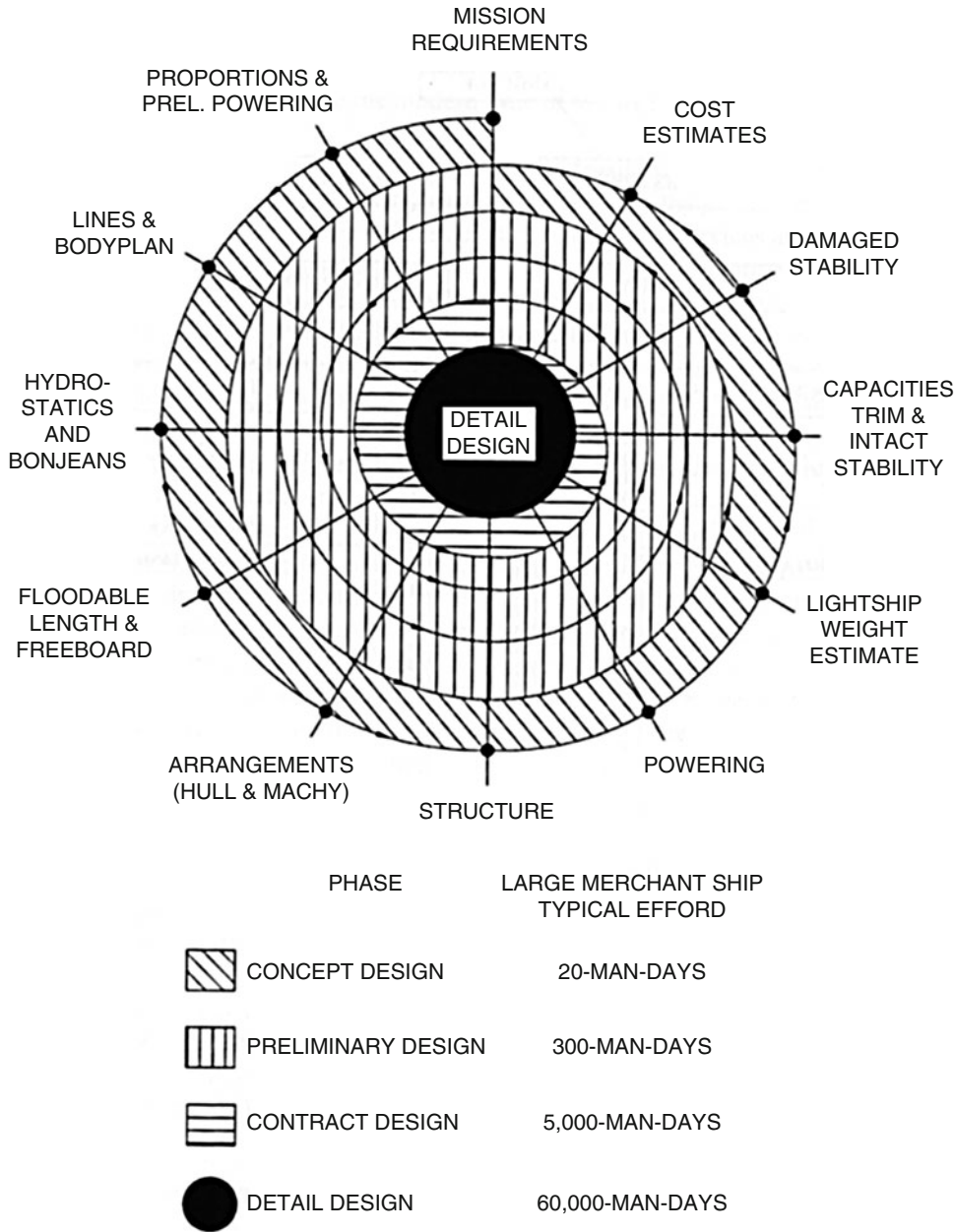
Many of the (largely iconographic) representations of the preliminary ship design process,



Design Spiral, Fig. 1 An early typical design spiral (Evans 1959)

such as those in the 2009 IMDC report, show the iterative nature of that process, which is seen as one of the advantages of the design spiral representation. This iterative characteristic even applies to most of the other, largely sequential, representations of the process. This feedback process can be seen in both the classical sequential ship design synthesis (Fig. 2) and in the more

integrated approach behind the DBB approach adopted. From the 12th preliminary ship design representation in the 2009 IMDC SoA design methodology report, that in both the numerical sequential and the architecturally integrated (DBB) representations the output is fed back to the input, once a ship design balance has been achieved. This reconsideration of the input



Design Spiral, Fig. 2 A simple numeric ship sizing iterative sequence with feedback (Andrews and Dicks 1997)

(requirements) is entirely consistent with the preferred systems engineering approach of Requirements Elucidation propounded by the second author (Andrews 2012). So there is both feedback at a gross level of the whole ship design process, which is often seen to be consistent with achieving one revolution of the design spiral, but this is also seen to occur at each component step (such as the purely numerical balance of weight and space of Fig. 3). These are subtleties not easily modelled by a single design spiral.

The other aspect that the design spiral is considered to model, or at least acknowledge, is the interactive nature of ship design. This is said to persist through out the process, such that any change in a subcomponent system has knock on effects across a range of other supporting ship systems. Thus simply in sizing fuel tanks on a naval ship, Rawson and Tupper's classic text book (1976) shows what is called a "design influence diagram" to reveal the influence on both the preliminary sizing and the final fuel tank arrangements. So changes in anyone of the influences will lead to alterations in this set of fuel tanks. Thus there are a host of interacting consequences with most of the ship design variables being interdependent to a significant degree. The set of tanks will then themselves have influences on, typically, the stability of the ship requiring changes to ship dimensions, weight balance, and structural scantlings, which will further alter the design downstream across a whole range of other component subsystems.

Thus these two aspects of ship design, the iterative and the interactive, need to be reflected in any model or representation of initial ship design, such as the design spiral, if it is to be a valid representation of the ship design process. In particular, such representations should indicate how a design evolves due to major changes in requirements or choices in design style in order to achieve a balanced design.

Basic Introduction to the Process of Design Spiral

Perhaps the most comprehensive representation of the ship design spiral is that reproduced at Fig. 4, due to Rawson (1986). This has 27 steps round the spiral, since it includes the set of "prime"

and "additional" hull form parameters, that he considers to be determined in the process. Furthermore, this spiral reverses the usual spiraling inwards to a solution by spiraling outwards, perhaps suggesting the increase in complexity of the solution as it evolves but still (hopefully) converges? Each sweep is said to be a major phase of the whole ship design process, namely, economics, military effectiveness (these cover commercial and naval vessels respectively), concept, feasibility, and finally full design.

A three-dimensional picture of the design spiral, as a tapering corkscrew, for each of the major ship design phases was done to emphasize, unlike the two-dimensional representation with a progressive sequence of closed loops, that real ship design operates with considerable dialogue in and out of the looping process with inputs by the designer and the many stakeholders (Fig. 5).

This 3-D model was intended to describe (any) one phase of the ship design process and as such could require several complete spirals to achieve the appropriate design balance. This model was felt to be consistent with the description of the concept phase and also with the subsequent feasibility phase undertaken after the concept phase and prior to approval to contract for detailed design and construction. The second author has managed projects with year long feasibility studies, which were only considered to be complete after three full cycles of design balance, each of increasing design fidelity.

Thus strong interactions are suggested throughout the spiraling process with a series of external influences or constraints. These constraints were described as respectively:

- Directly on the design – seen as direct impositions from the customer/users/specialists/classification societies, for aspects such as reduced manning.
- On the design process – such as sources of design data and tools/methods/standards, which might well alter the spiral sequence and lead to the process jumping back and forth, also some of these can be influenced by the design team.
- Originating from the design environment – these are wider constraints, such as the

ASSUMPTIONS

Final ship design solution will not demand major equipment changess

The level of complement machinery, services, structure etc. will be in line with current solutions

The degree of complexity, structural philosophy, standards of accommodation, upkeep, margin philosophy implied

Conventional wisdom on form parameters, shafting, machinery redundancy, etc.

Level of operation (endurance), maintenance philosophy, navy or company manning philosophy and organizational structure

Systems operating and upkeep philosophies, redundacny, etc.

Applicability of hydrodynamics data (triplet, methodical) series), usage of auxiliaries, philosophy on tank allocation (Stability, longitudinal balance, proximity)

Assumed structural design philosophy and material choice, implied configuration solution (% access), design point, stability philosophy

Judgement of 'Satisfactory Balance'

STEPS IN PROCESS

Payload

Total internal volume

First shot at displacement

Selection of machinery

Complement

Auxiliary power and services

Tank volume

Overall displacement and internal volume

Reiterate until displacement and volume balance

SOURCES

Historical data on equipment or estimates by equipment designers (weight, space, services, complement)

Based on past practice from payload volume

Measures of the density of the current ships of that category

Power/speed/displacement plots
Machinery data (volume, mass, auxiliary power, complement)

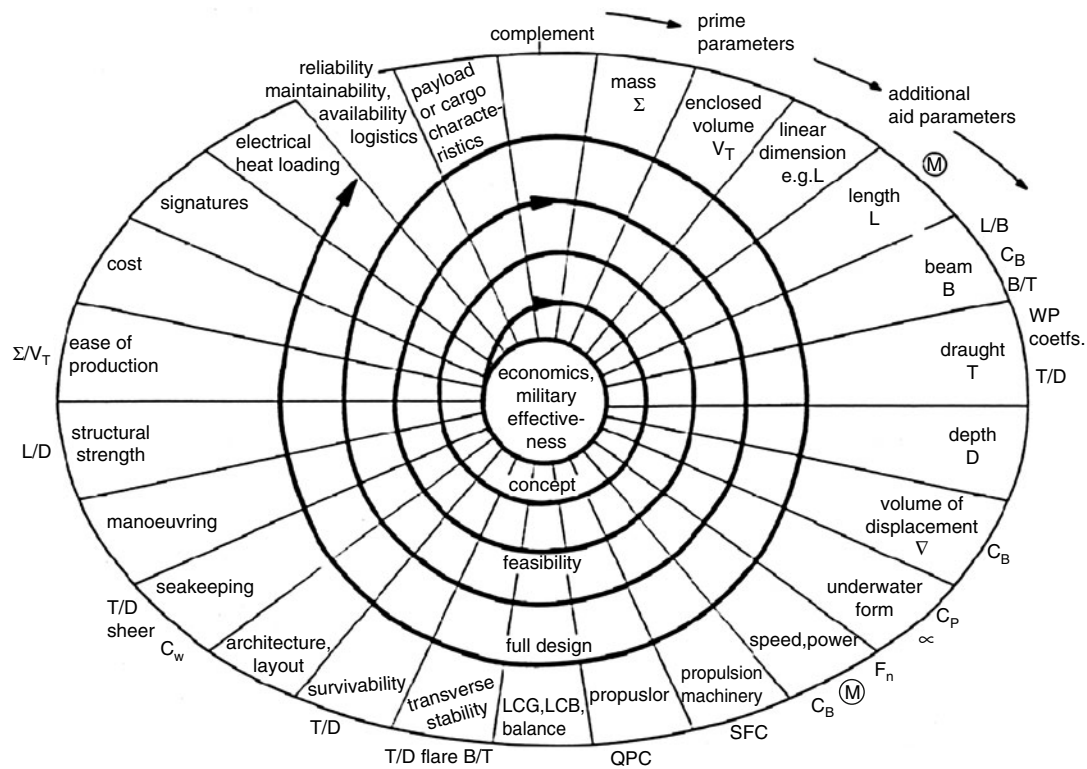
Either overall figure displacement dependant or given by the sum of complement for machinery, payload and remainder

Equipment data
Data book of historical demand related to ship size complement with step changes for discrete range of equipment

Endurance calculation (hydrodynamics data).
Hotel fuel consumption
Fresh water, lub oil, voids

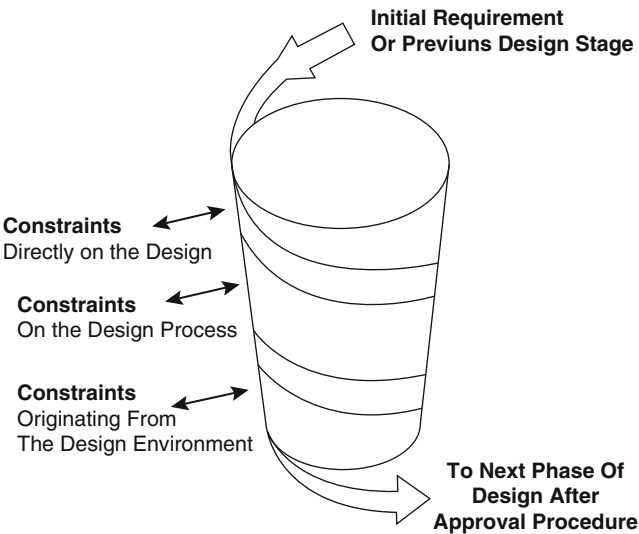
Sum of payload, mcy, services outfit, structure
Margins on items, growth, 'Board'
Superstructure volume as proportion

Design Spiral, Fig. 3 Design spiral (Evans 1959; see Taggart 1980)



Design Spiral, Fig. 4 A ship design process model (Rawson 1986)

Design Spiral, Fig. 5 3-D representation of the ship design spiral (Andrews 1981)



education and training of the designers, government legislation (e.g., environmental/safety), economic climate, major organizational changes, all of which can be highly influential but over which the design team has no influence.

Having remarked that the design spiral takes many forms in trying to represent a complex process, it is now appropriate to consider nonlinearity in the evolution of most ship designs and whether this fact limits the value of the design spiral in modelling the design process.

Historical Development of Design Spiral

The first use of the design spiral in ship design is usually attributed to Harvey Evans (1959) in outlining the sequence for midship structural section design. His spiral has nine steps that were shown to be repeated several times – hence the spiral representation, as shown in Fig. 6.

This example was followed by several versions representing the whole ship design process

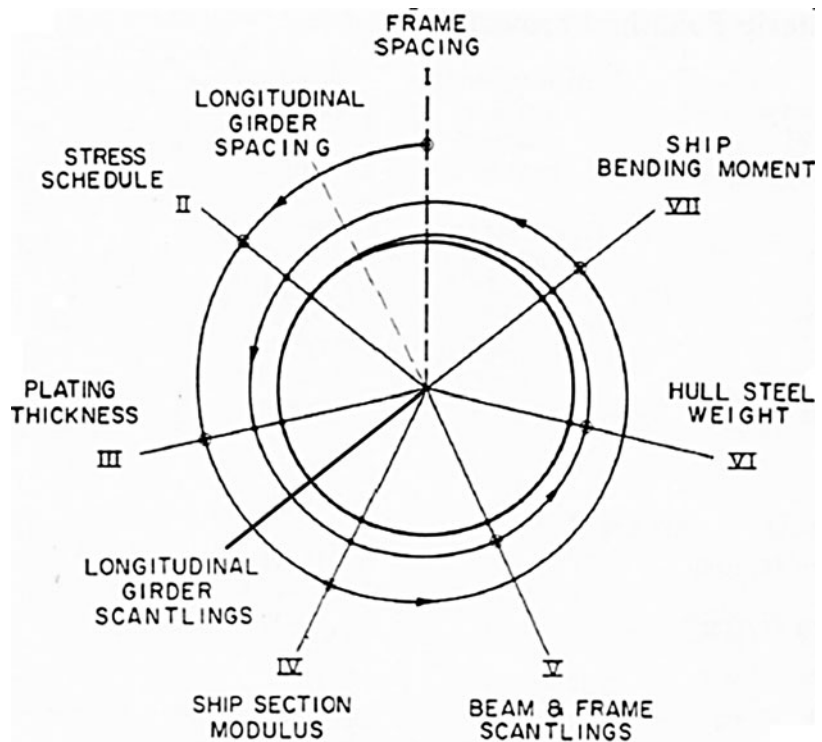
(such as the second, third, and ninth examples reproduced in the 2009 IMDC SoA design methodology report (Andrews et al. 2009)). These show the same steps being undertaken at each sweep of the spiral (but clearly each time more comprehensively). Rawson and Tupper (1973) even remark that not all the twelve steps, which they show, “are required to be developed in depth at every iteration.”

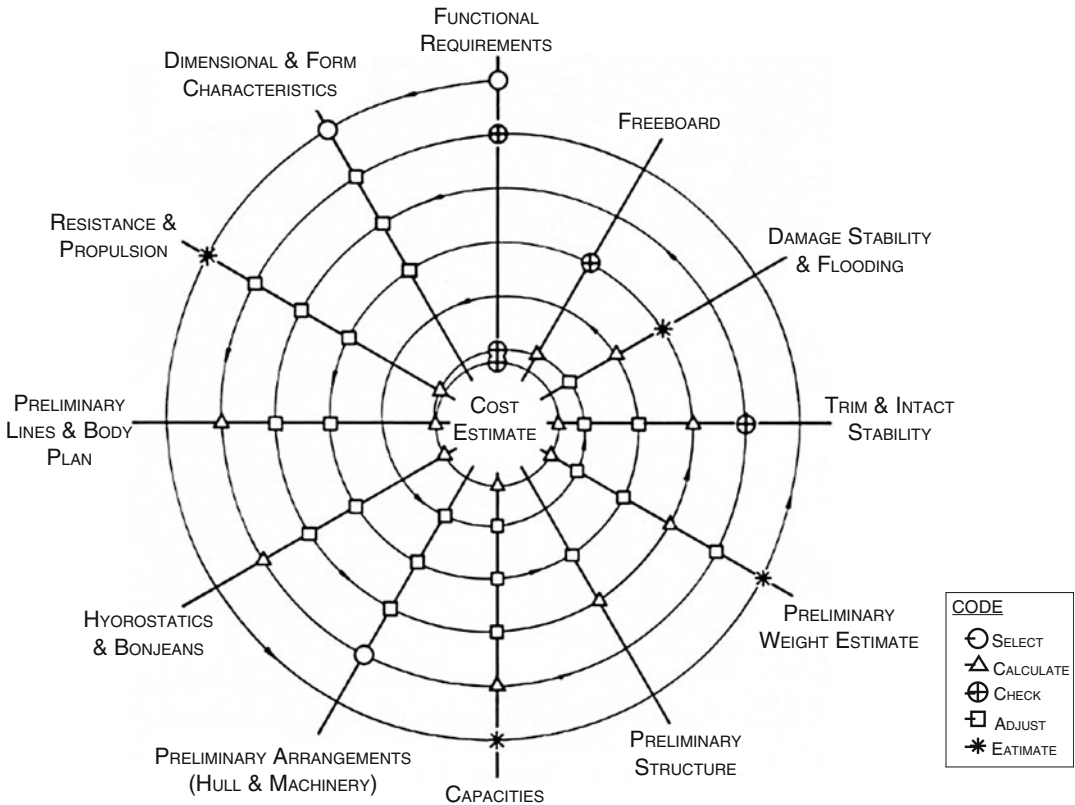
Several years later, in 1965, in order to get reliable cost estimate, MarAD Proposed a preliminary design spiral shown in Fig. 7. It takes sequential, cyclical approach in rigid sequence, which is based on rules and takes univariate systematic variation with constraints. Thus result in single feasible ship with lowest building and fuel cost.

It is considered debatable whether each of the above major design phases should be shown as only one sweep. Eames and Drummond (1977) (the fifth example in the 2009 SoA design methodology report) in their concept exploration paper show three concept

Design Spiral,

Fig. 6 Midship section structural design diagram





Design Spiral, Fig. 7 Preliminary design spiral (MarAD 1965)

stages, each as a complete sweep (called “concept exploration, concept development, and concept validation”) and with ten steps or “spokes.” These follow the same sequence for each of their concept stages and the process starts with a “basis ship” (rather than even outline requirements) and ends with “contract design.” The latter suggest this is modelling a process typical of a commercial shipyard’s bid response to a ship owner’s indicative design, rather than the more protracted concept phase for naval procurement. An example of the latter was outlined for UK warship procurement (Andrews 1994) with three overlapping stages entitled concept exploration (but quite different to the Eames and Drummond (1977) usage, being much more divergent), concept studies and concept design, all undertaken prior to what is now known as the Initial Gate decision point in the UK Ministry of Defence (UK MoD 1999). It is

also relevant that each of these concept stages do not follow the simple repetitive description of the design spiral, as the 1994 paper demonstrates.

A further variant on the representation of the design spiral was proposed in 1981 by Andrews (see Fig. 5, the sixth example in the 2009 IMDC SoA design methodology report). This gives a three-dimensional picture of the design spiral, as a tapering corkscrew, for each of the major ship design phases.

Key Applications

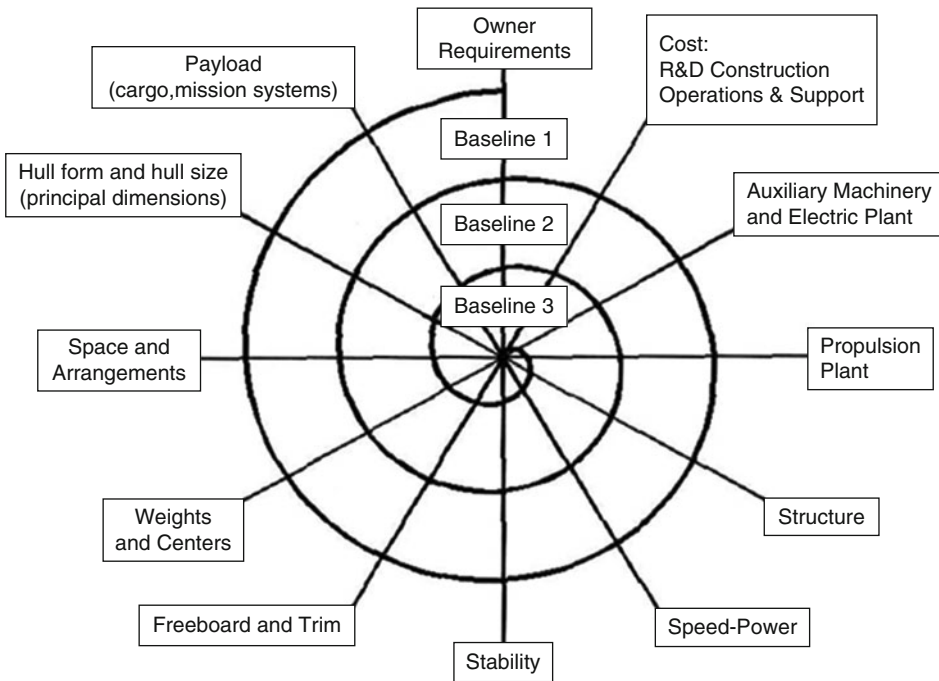
Design spiral has been used in many areas such as ship design, sequential engineering, ROV design, AUV design, preliminary design, etc. Here we list two most important use.

Ship Design

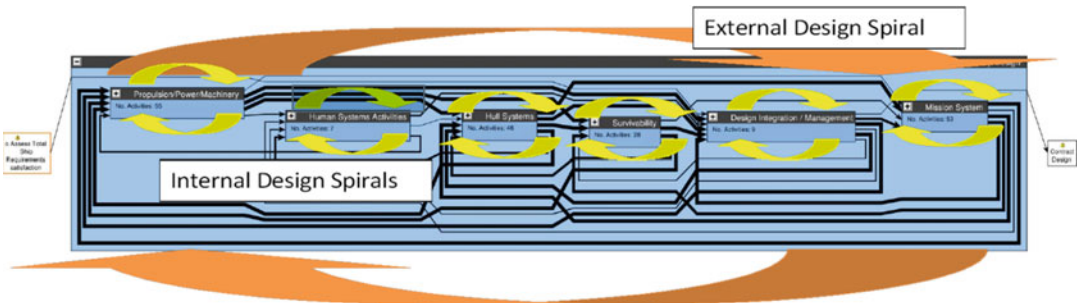
The process of ship design has an iterative nature that is most commonly described by a design spiral that reflects the design methodology and strategy.

The goals for the ship design are not the same for all ship-owners; NAP (Naval Architecture Progress) developed the tools and gained the experience to incorporate all different – conflicting requirements and expectations in its design mission.

The design spiral shown in Fig. 8 shows interactions between different areas whose iterations and trade-offs need to be converged to a baseline. However, the spiral does not show the interaction between the engineering areas as iterations progress. These interactions are shown in Fig. 9. The design has an overall spiral, shown in the orange arrows. Within the spiral, there are internal spirals (yellow arrows) which represent the different engineering groups iterating their own work in that iteration's baseline.



Design Spiral, Fig. 8 Ship design and construction (Thomas Lamb 2003)



Design Spiral, Fig. 9 Ship design spiral

Arrows in the process viewed in Fig. 9 show major information flow. While the arrows capture some of the design interaction, all of the iteration that needs to take place cannot be detailed in a process mode.

ROV Design

Today, with the aid of advanced computer design techniques, the modern ROV has evolved through many iterations of the design spiral shown previously. Today's ROVs are relied upon to perform complex operations offshore, in ever increasing water depths, and have accordingly reached a high level of technical design. These vehicles must also be flexible, that is, they must be capable of being configured for many tasks. This holds true for small and large systems, which are used for a variety of inspection and/or work tasks. Since the goal of the ROV is to accomplish an often-complicated task, its overall capability is usually driven by two major considerations – work requirement and operational water depth – both of which drive the considerations of the design spiral.

The ROV system is a highly interrelated group of subsystems that, when functioning

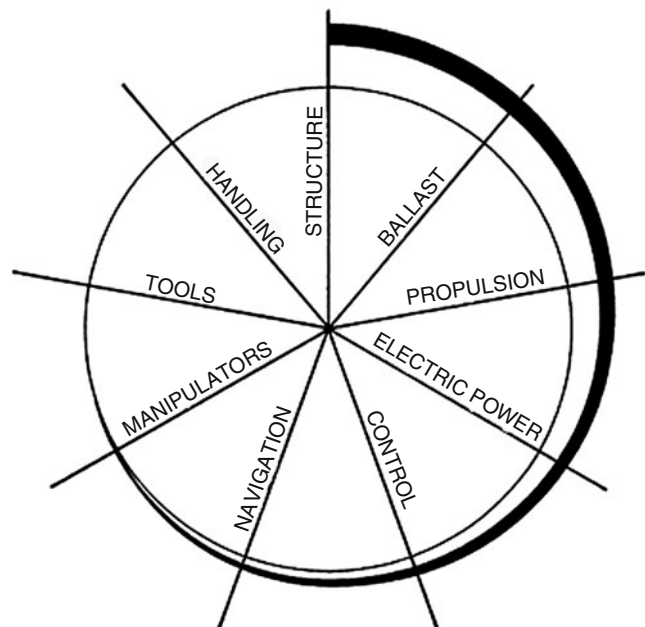
synergistically, provides an impressive subsea capability. Since the characteristics of the vehicles are multivariant, with no unique solution, the design process is necessarily iterative. Because of this highly interdependent relationship, ROV system performance is a delicate balance of design and operational characteristic tradeoffs. The system can be arbitrarily divided into a number of major subsystems. These would generally include:

- Vehicle
- Tools and sensors
- Control/display console(s)
- Electrical power distribution
- Umbilical and tether cables
- Handling system

The interrelationship of such subsystems is graphically illustrated in the “design spiral” shown in Fig. 10, the following figure. For example, an operator may desire a modest increase in the speed capability of an ROV to maneuver in a current condition in a particular geographic area. The additional speed desired will result in the need for additional electrical energy and may

Design Spiral,

Fig. 10 ROV design spiral



generate the need for changes in the thruster/pulsation subsystem. Additionally, the configuration, structure, weight, control system, umbilical/tether size, tether management system, winch, handling system and power supply will all be affected.

Clearly, all of these subsystems and components are affected by what was a seemingly innocuous performance improvement desire. Thus, the designer or user/operator is well advised to be aware of the ROV system's sensitivity to change and carefully consider the serious ramifications of contemplated changes to what may appear to be minor components. The desire for small improvements in performance or the failure of a "minor" component will often have ripple effects through the entire system.

Cross-References

- [Concept Design](#)
- [Design of Submersibles](#)
- [Detailed Design](#)
- [Empirical Design](#)
- [Hydrodynamic Design](#)
- [Multidisciplinary Design Optimization \(MDO\)](#)
- [Optimal Design](#)
- [Preliminary Design](#)
- [Reliability-Based Design \(RBD\)](#)
- [Structural Design](#)
- [Technical Design](#)

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Det Norske Veritas (DnV)

- [Installation of Offshore Pipelines](#)

Detailed Design

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Synonyms

[Concept design](#); [Design spiral](#); [Multidisciplinary design](#); [Optimal design](#); [Preliminary design](#); [Technical design](#)

Definition

Detailed design is the last stage of technical design and it completes the design of the performance, structure, electrical, materials, components, processing, assembly and testing of the submersible system, and forms a complete set of technical design documents and drawings, which can be used as the basis for the next construction, integration, and test.

Scientific Fundamentals

Basic Introduction to the Design Process

The design and associated processes are diagrammed in Fig. 1 in a chronological order as (1) pre-design, (2) design, and (3) post-design processes (Allmendinger 1990). In this section, they are introduced briefly from the perspective of the mission system to provide an overview of the entire subject.

The pre-design process leads to the formulation of the basic input to the design process, that is, the mission(s) statement(s) and associated mission *requirements*. It is undertaken by the potential user or owner of the mission system to be designed, independently or in consultation with the system's designer.

The design process is composed of three design phases: (1) *basic design*, (2) *contract design*, and (3) *detail design*. The levels of labor, preparation time, and costs associated with each phase increase exponentially as the process progresses. Design creativity and flexibility are essentially limited to the basic design phase – the mission system, including its submersible system, being well defined at the beginning of contract design.

The post-design process is composed of three activities: construction/alteration, test and evaluation, and operation. Although this publication is not directly concerned with these activities, there are two reasons for including them in this overall perspective: (1) They constitute important sources of input to new designs and feedback to the design in question, and (2) they are subject to certain existing MEDCs and will influence the future impact of MEDCs on following designs.

Conceptual design is the first attempt to translate all the user/owner's mission requirements into performance requirements and characteristics of the systems composing the mission system. It is often viewed as consisting of *feasibility studies* and *completion of the optimum conceptual design(s)* once they are identified by these studies. Feasibility studies are concerned with the development of those performance requirements and associated system characteristics which have significant impact on the cost-effectiveness optimization process. Completion of the optimum

design(s) involves developing other major performance requirements and characteristics those which are essential in defining systems but which do not have significant impact on the optimization process.

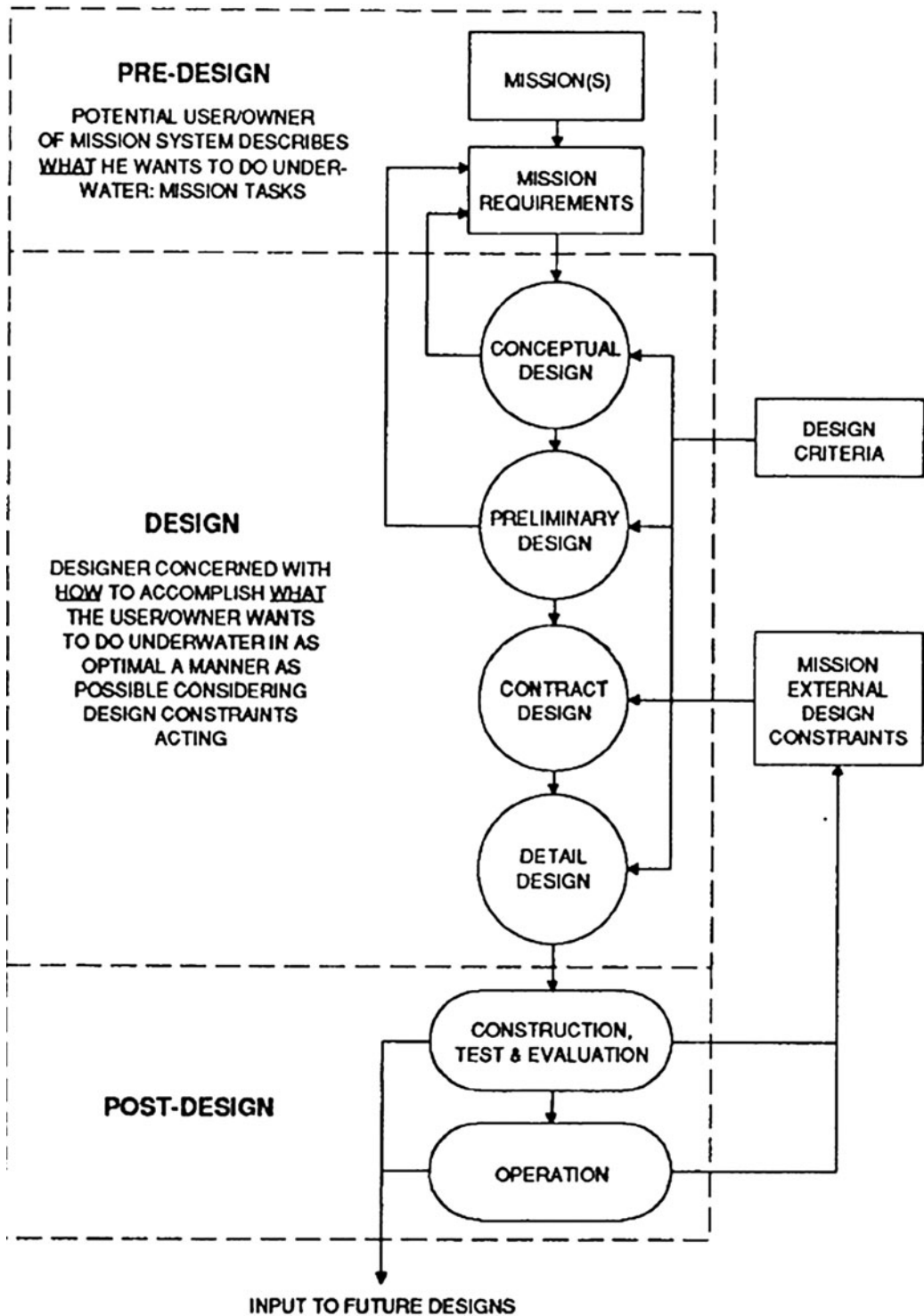
Preliminary design is the second stage of the basic design and is concerned with this phase's fifth function. Starting with baseline data provided by the selected conceptual design(s), it refines and firms up the major characteristics of systems, the MIDCs they exert on each other, and cost estimates associated with them. In summary, it provides a precise engineering definition of the mission system and assurance that the design goal can be attained.

Contract design requires yet further refinement of design and additional detail. It yields contract plans and specifications necessary for interested parties to bid on the construction of new systems or the alteration of existing systems. It also provides contractual documents for the construction and alteration work.

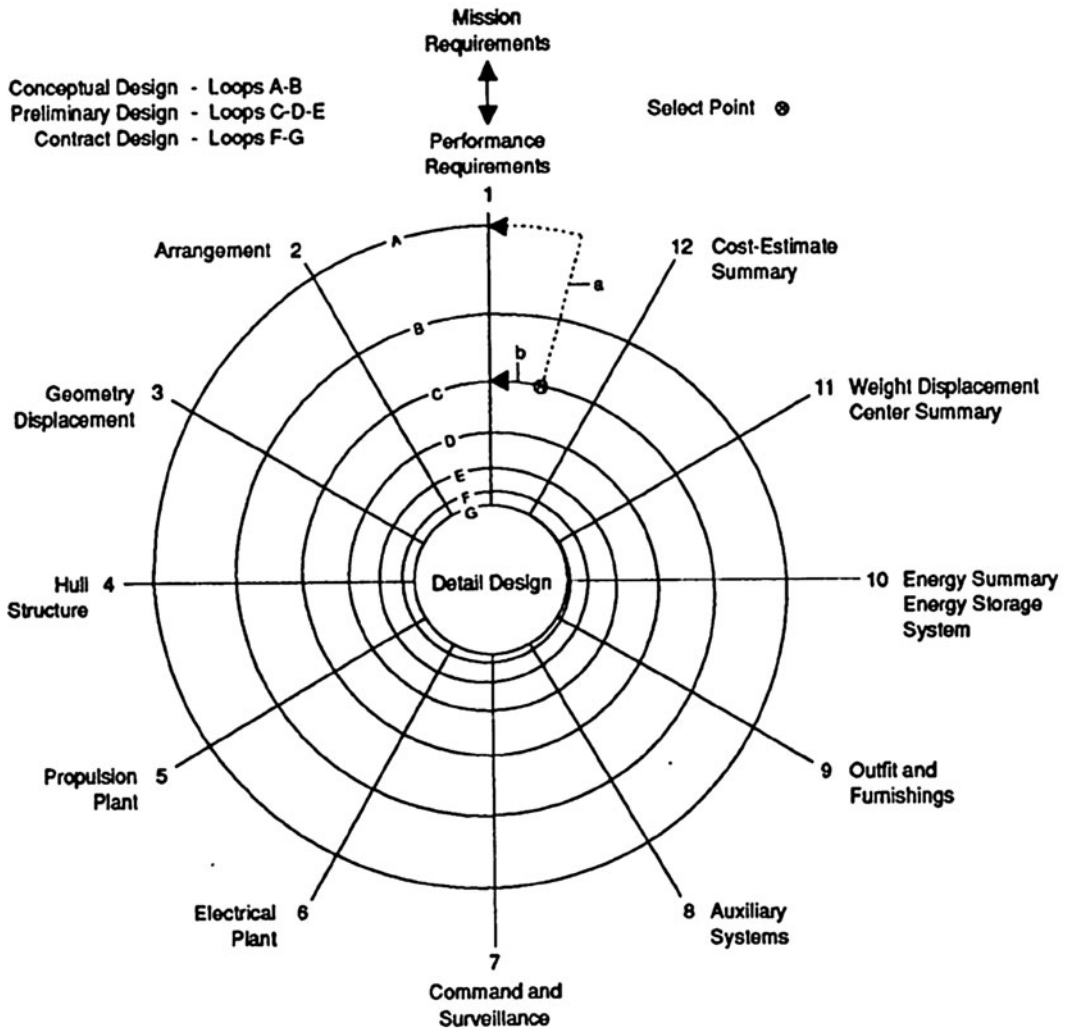
Detail design is the final phase of the design process and entails the development of detailed working plans from which the systems are constructed or altered. In one sense, it is really not a design phase since all the creative design effort is done in the preceding phases, the design being unequivocally defined prior to entering this phase. It does, however, require the greatest amount of work of all the phases and is often undertaken by entities building or altering the system.

Basic Introduction to the Design Spiral

The iterative nature of the overall design process may be modeled by the design spiral shown in Fig. 2. The spiral consists of spokes and loops. The spokes represent design considerations including performance requirements, submersible characteristics, and cost estimates; submersible characteristics are embodied in its arrangement, geometry, six SWBS systems groups, and summaries of energy requirements and weight/displacement centers. The loops indicate design iterations with design's refinement increasing as the loops spiral inward. Outer loop procedures use empirical formulations and estimates based on



Detailed Design, Fig. 1 Design and associated processes (Allmendinger 1990)



Detailed Design, Fig. 2 Design spiral (Allmendinger 1990)

similar design data, whereas middle and inner loop procedures are based on progressively more detailed development of the design considerations utilizing increasingly precise formulations. The intersection of spokes and loops forms points which are identified alphanumerically. Thus, for example, 3B is a point in the spiral at which the geometric characteristics of the submersible are being considered in the second iteration in a more detailed manner than they were initially at point 3A. At 3B, adjustments are made as the result of both the more detailed consideration of the geometric characteristics required and the impact on

geometry of interrelated design considerations for which data were generated at points in the spiral preceding 3B. Adjustments are made at other points in the spiral for the same reasons. For the design to converge to a satisfactory solution of the design problem, these adjustments must become progressively smaller with each succeeding loop through conceptual, preliminary, and contract design. This fact is demonstrated on the spiral by the decreasing spacing between loops as they spiral inward. Convergence is illustrated by the innermost loop becoming a circle, indicating that all design considerations are fully developed and

require no further adjustments. At this point in the spiral, detailed design can begin.

Take the loop A for example:

- 1A. Performance Requirements: Derive performance requirements from the mission requirements pertinent to the submersible's design. These performance requirements should include depth, range, speed, maneuvering, personnel on board, endurance, payload, system requirements, safety features, and limiting sea states.
- 2A. Arrangement: Initially, the submersible may be considered as consisting of the pressure hull (s), envelope, and all other items comprising it. Select types and number of pressure hulls, type of envelope, and location of major items of SWBS system either inside or outside the pressure hull.
- 3A. Geometry – shape and dimensions: Obtain first estimates of pressure hull and envelope geometry and volumes and displacement of submersible.
- 4A. Hull Structure: Fixed ballast and buoyancy are included in this group – these systems have the function of providing for neutral buoyancy, zero trim, adequate stability, and a weight margin in design. At this point, only the amount of fixed ballast for the last function is included, its weight is usually 10% or less of the total displacement. Fixed ballast or buoyancy serving other functions is considered at Point 11, weight displacement center summary.
- 5A. Propulsion Plant: The propulsion power source – which consists almost always of batteries for untethered submersibles – is included in this group. However, for convenience, consideration of this system's sizing data is deferred until Point 10, energy summary.
- 6A. Electrical Plant: The non-propulsion power source for auxiliary and other system's service loads is included in this group. Again, for convenience, consideration of this system's sizing data is deferred until Point 10. The remaining systems of this group, the power distribution and lighting systems, constitute very small percentages of Condition A1 (light ship minus fixed ballast/buoyancy) weight and displacements. Consequently, first estimates of weight and displacement of this group, excluding non-propulsion power source, may be obtained using percentages from similar design data.
- 7A. Command and Surveillance: The total weight and displacement of this group of systems are also very small percentages of Condition A1 weight and displacement. As at Point 6A, first estimates of this group's weight and displacement may be obtained using percentages derived from similar designs.
- 8A. Auxiliary Systems: Certain systems in this group require comment. The auxiliary systems include human system, air gas and miscellaneous fluids systems, static control systems, dynamic control systems, and mission systems.
- 9A. Outfit and Furnishings: This system's weight and displacement are minute percentages of the Condition A1 weight and displacement. In both instances, 1% may be used for first estimate purposes. Space requirements are another matter for items inside the pressure hull, ladders, lockers, hull insulation, seats, and so forth. Even first estimates of these sizing data should be made with great care because they have considerable influence on pressure hull volume and, hence, have a cascading effect on other design considerations.
- 10A. Energy Summary – Energy Storage Systems: At this point, the power-energy requirements for the SWBS systems should be summarized and the sizing data for the electrical and pneumatic energy storage systems estimated, the two forms of energy carried on board almost all untethered, manned submersibles. The summarization procedure for electrical energy storage systems, assumed to be secondary batteries, involves use of a power energy profile.
- 11A. Weight Displacement Center Summary: At this point, summarize weights and displacements of SWBS systems and their loads and achieve the submerged design condition criterion of neutral buoyancy, in which total weight and displacement are equal. Note that the other two design condition criteria of zero trim and adequate positive stability are addressed at Point 11B when estimates of longitudinal and vertical centers of weight and displacement are available.

12B. Cost Estimate Summary: Summarize estimates of first and annual operating costs associated with the SWBS systems selected for a particular design alternative.

Key Applications

Development of Deep Sea Manned Submersible

Manned submersible is an important technical means and equipment for deep sea entry, exploration, development, and protection, which represents the development frontier of the submersible technology. Alvin was the most famous (Kaharl 1990; WHOI 2018) while other large depth manned operational submersibles include Nautile of France (Lévêque and Drogou 2006); MIR I & II of Russia (Sagalevitch 2009); Shinkai6500 of Japan (Takagawa 1995); and Jiaolong of China (Cui 2013).

In 2002, China launched the development of the 7000 m manned submersible Jiaolong. Through the joint research of nearly 100 dominant scientific research institutions in China, through the stages of scheme design, preliminary design, detailed design, processing and manufacturing, general assembly and commissioning, and functional test of water tank, the technical conditions for sea trial were met in early 2008.

From August to October in 2009, May to July in 2010, July to August in 2011, and June to July in 2012, the sea trials tests of 1000, 3000, 5000, and 7000 m were completed, respectively, and the

maximum diving depth reached 7062 m. In 2013, 7000 m manned submersibles have been put into use. More than 150 dives have been completed in the South China Sea, the manganese nodule area in the eastern Pacific Ocean. The cobalt-rich crust areas are in the Western Pacific Ocean, the Mariana Trench, the southwest Indian Ocean, and the northwest Indian Ocean. A large number of biological, mineral, sediment, and rock samples have been obtained, and the high-resolution image data of the seabed have been taken, which provides a basis for the assessment of deep-sea resources and environment important basis. In 2019, the first test dive will be completed after major repair and technical upgrading (Hu and Cao 2019) (Fig. 3).

Alvin Upgrade

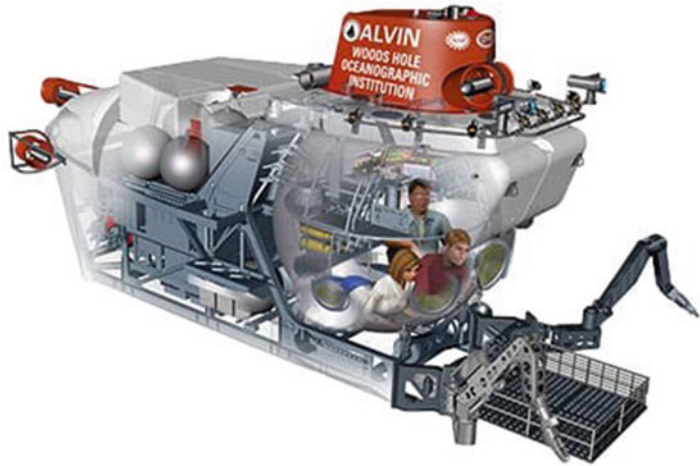
Alvin is the world's longest-operating deep sea submersible. It was launched in 1964 and has made more than 4700 dives, along the way participating in some of the most iconic discoveries in the deep ocean. Throughout 2011, 2012, and into 2013, *Alvin* received a comprehensive overhaul and upgrade funded by the National Science Foundation that greatly expanded its capabilities and will eventually put almost the entire ocean floor within its reach.

The upgrade is taking place in two stages. After Stage 1 was completed in 2013, *Alvin* boasted several new improvements, including: (1) A new, larger personnel sphere with an ergonomic interior designed to improve comfort on long dives. (2) Five viewports (instead of three) to

Detailed Design,

Fig. 3 Jiaolong deep manned submersible (www.cssrc.com)



Detailed Design,**Fig. 4** Alvin upgrade
(www.whoi.edu)

improve visibility and provide overlapping fields of view for the pilot and two observers. (3) New lighting and high-definition imaging systems. (4) New syntactic foam providing buoyancy. (5) Improved command and control system.

Several of these and other improvements to the sub have been designed to withstand descents to 6500 m – the remainder will be upgraded later. As a result, Alvin will initially maintain its current depth rating of 4500 m. In Stage 2, the entire sub will be upgraded to 6500 m depth. In addition, new batteries will be introduced to enable the submersible to stay at depth longer, giving scientists more time to work in unexplored parts of the ocean and putting 98% of the seafloor within their reach (Fig. 4).

New Technology Development

Titan is a Cyclops-class manned submersible designed to take 5 people to depths of 4000 m (13,123 feet) for site survey and inspection, research and data collection, film and media production, and deep sea testing of hardware and software. Through the innovative use of modern materials, Titan is lighter in weight and more cost efficient to mobilize than any other deep diving submersible. A combination of ground-breaking engineering and off-the-shelf technology gives Titan a unique advantage over other deep diving subs; the proprietary Real Time Hull Health Monitoring (RTM) systems provide an unparalleled safety feature that assesses the integrity of the hull throughout every dive. The use of off-the-

shelf components helped to streamline the construction, and makes it simple to operate and replace parts in the field.

Paired with a patented, integrated launch, and recovery platform, Titan is easy to operate in varying sea states using a local appropriately sized ship for the project. In coastal waters this means we do not need a large support ship with a crane or A-frame.

Cross-References

- [Concept Design](#)
- [Design of Submersibles](#)
- [Design Spiral](#)
- [Empirical Design](#)
- [Multidisciplinary Design](#)
- [Optimal Design](#)
- [Preliminary Design](#)
- [Structural Design](#)

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Differential Evolution (DE)

- [Obstacle Avoidance Technology for Underwater Vehicle](#)

Diffusive Equilibration in Thin Films (DET)

- [Underwater Lander](#)

Diffusive Gradient in Thin Films (DGT)

- [Underwater Lander](#)

Digital Signal Processor (DSP)

- [Optical Compass](#)

Digital Tank

- [Numerical Tank](#)

Digital-to-Analog (D/A)

- [Underwater Acoustic Communication](#)

Direct Sequence Spread Spectrum (DSSS)

- [Long Baseline Underwater Acoustic Location Technology](#)

Discrete Element Method (DEM)

- [Maneuverability of Polar Vessel](#)

Discrete-Modules-Based Hydroelasticity Method

- [Hydroelasticity Theory](#)

Displacement Volume

- [AUV/ROV/HOV Hydrostatics](#)

Dissolved Oxygen (DO)

- [Net Structures: Biofouling and Antifouling](#)

Diving of Anchors

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Introduction

The word of “anchor” derives from the Latin form of ancora, where, in a broad sense, anchor is a type of device to stabilize structures, including the offshore anchors as well as soil nails to reinforce

foundations, retaining walls, and slopes. This entry concentrates on the fully embedded offshore anchors, e.g., traditional fluke anchors or more recently plate anchors, used to connect a floating offshore structure to the seabed to prevent it from drifting due to wind, wave, or current.

Floating offshore structures including vessels, drilling rigs, and platforms are subjected to large environmental forces, and most rely on permanent or temporary anchoring systems to withstand these forces. Some of these floating systems, such as floating production storage and offloading (FPSOs) vessels and floating production systems (FPSs), are normally required to remain on station for long periods, while mobile floating systems, such as mobile offshore drilling units (MODUs), use anchoring for temporary mooring to withstand the severest weather conditions. The performance of anchors embedded in the seabed therefore has a great influence on the reliability, integrity, and operational safety of these floating systems. Anchors embedded in the seafloor generate their holding capacity mainly by mobilizing the shear strength of the soil to resist the pulling force. Therefore, the digging or diving depth of anchors is a critical factor to determine the holding capacities of embedded anchors.

Encouraging Anchors to Dive

Anchor History

The evolution of anchors has a closely linked pace to the history of human civilizations, and the first anchor dates from time immemorial. From a chronological standpoint, the development history of anchors includes four distinct periods:

1. The prehistoric period associated with the use of stones to hold vessels by their weight. This kind of stone deadweight anchors were routinely used by the Egyptians.
2. The ancient period (Greco-Roman) when hook anchors first made of wood and gradually stiffened by metal parts were used. Chinese was reported to use hook anchors to stabilize bamboo stocks as early as 2000 BC.

3. The Middle Ages extending to the mid-nineteenth century characterized by a complete absence of innovations and even by the loss of certain attainments.
4. The present period started from 1821 marked by the extremely rapid development of techniques.

The patent for an anchor consisting of two flukes articulated about a pin perpendicular to the shank allowing diving and penetrating into the soil began this era. Along with this chronological line, some typical anchors are introduced as below.

Deadweight Anchors

The ancestor of all anchors is very probably a more or less flat stone pierced with a hole for the passage of a rope, as illustrated in Fig. 1a (Puech 1984). Although this primitive anchor idea was used from thousands of years ago, the mobilization of the holding force by means of a weight placed on the seabed offers such reliable results that this system is still used today in the form of deadweight weighing several thousand tons. Figure 1b shows the deadweight anchor on the North West Shelf of Australia for the guyed flare support tower at North Rankin, where a hollow box is placed on the seabed and then filled with rock and iron ore. Other similar application examples include those used in Wanaea and Cossack fields (Randolph and Gourvenec 2011).

However, this type of pure deadweight design is not efficient, as the resistance force is only a fraction of the weight to be handled. The evolution of anchor develops toward taking advantage of the seabed soil resistance. As illustrated in Fig. 1c, anchor stone fitted with wooden cross-pieces and killick use the combined action of weights and hooks designed to plow the seabed, which helped to increase the efficiency remarkably. The matching modern example is the innovative type of gravity anchor using a buried grillage, beneath a rock-fill or iron ore berm, which was used for the Apache Stag field development on the North West Shelf (Erbrich and Neubecker 1999). As shown in Fig. 1d, the grillage is placed toward the back of the berm but

such that the complete berm must be moved if the grillage starts to fail.

Hook Anchors

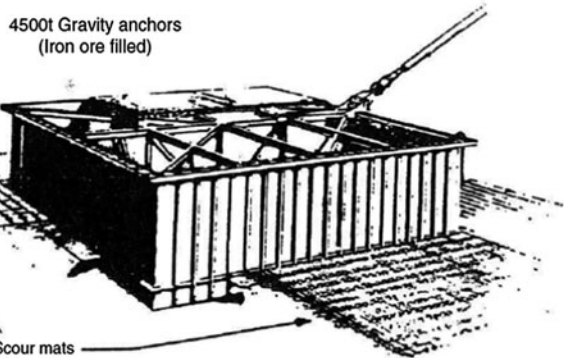
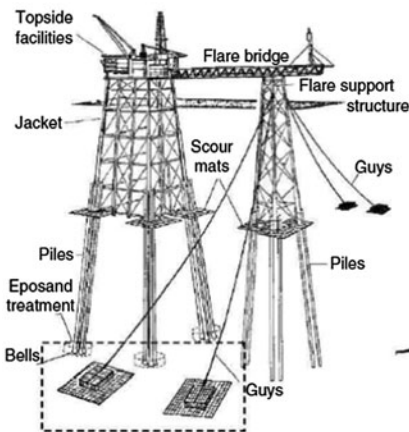
Hook anchors see a big step development toward taking advantage of the seabed soil resistance. The

earliest hook anchors consisted of an assembly of pieces of wood in a “V” shape, where a stone and a piece of bamboo were used by India and China as illustrated in Fig. 2a. The Greeks and Romans strengthened metal parts by stiffening the teeth or flukes with iron plates as shown in Fig. 2b. These anchors reached the summit of their development with the classic stock anchor, and the latest model

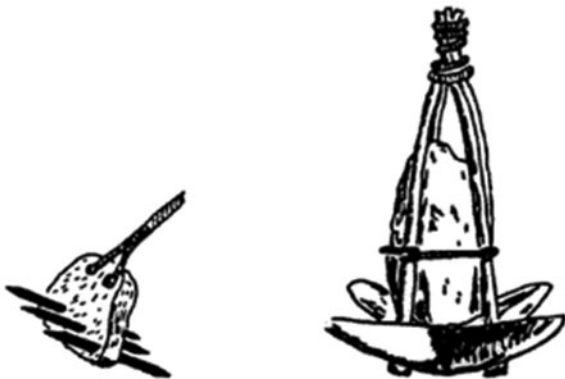
D



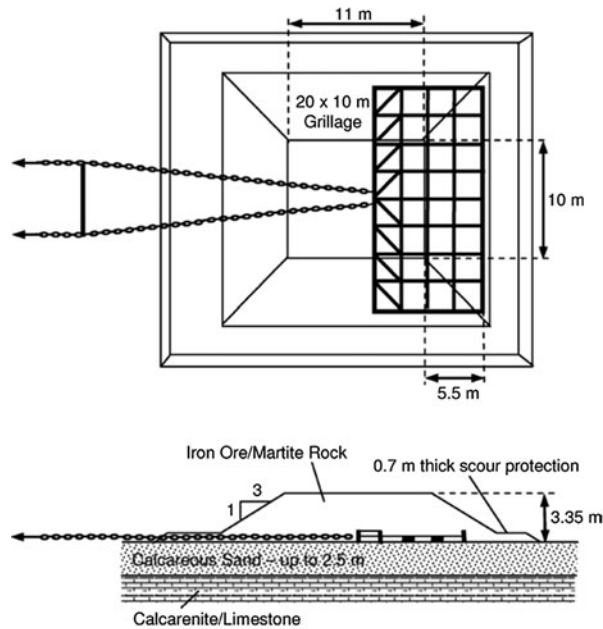
(a) Anchor stone discovered near Hamburg



(b) Box anchors for North Rankin Flare Tower



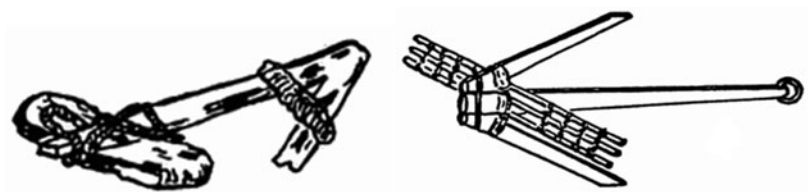
(c) Anchor stone with wooden crosspieces and killick anchor



(d) Grillage and berm anchors

Diving of Anchors, Fig. 1 Deadweight anchors

Diving of Anchors,
Fig. 2 Hook anchors



(a) Wood hook anchors



(b) Wooden hook anchor with stock and lead stiffeners



(c) Classic Admiralty anchor

of the type called the Admiralty Anchor (Fig. 2d) dated from the mid-nineteenth century.

The main feature of these anchors was undoubtedly that they derive their holding power

from embedding into sea floor. However, their efficiency was quite low due to the small fluke area penetrating into the seabed soil. The rapid increase in the weight of warships and

merchantmen in that time led to the development of stronger anchors with shanks and stocks 5–6 m long, which naturally became cumbersome and difficult to handle, despite the possibility of folding down the stock along the shank for stowing (Puech 1984).

Articulated Marine Anchors

Modern marine anchors were heralded by the Hawkins' patent in 1821 to stride to the era of modern articulated fluke anchors, which can be classified in two main categories: stockless anchors and stock anchors. Stockless anchors underwent rapid development to replace the hook-stock anchors in the end of the nineteenth century, which are still widely used today on warships and merchantmen. Figure 3a shows the “Union Universal” 20 t stockless anchor. Articulated stock anchors emerged slightly later with the stock located at the crown at the rear of the anchor as shown in Fig. 3b.

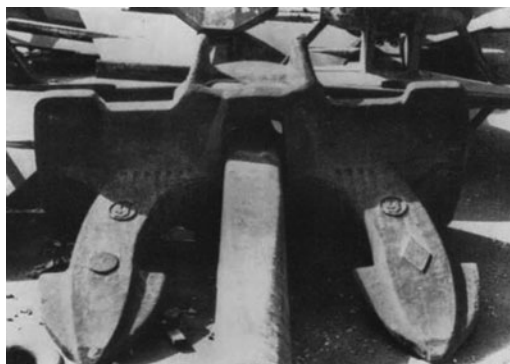
Special Anchors for Offshore Operations

The fast growth of offshore operations essentially related to petroleum exploration and production was accompanied by sweeping changes in anchoring requirements, due essentially to the steadily increasing size of the floating infrastructures. This led to the construction of stockless anchors and

stock anchors weighing up to 20 and even 30 t in air and arose to use anchor-handling vessels to install and retrieve anchors. Around 1970 some manufacturers and inventors, such Vryhof and Bruce tried to develop new products which were better adapted to offshore operating conditions. The spread of petroleum activities in the North Sea and Gulf of Mexico and the variety of soils encountered at the different sites are largely responsible for the huge innovative effort. Today, the evolution of anchoring techniques requires that the temporary mooring of supports mobilizing forces of ~200 t per anchor mooring line can be considered as a standard and reliable operation. Even better performing anchors are already capable of mobilizing forces of ~1000 t for the permanent anchoring of production supports.

Simplified Calculation Model for Embedded Anchors

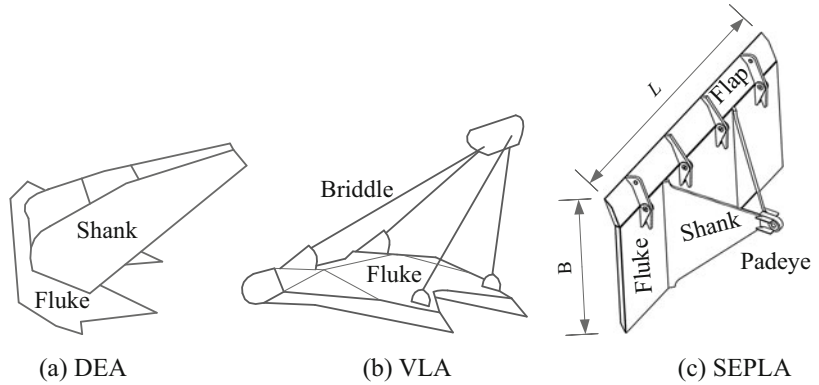
The development of anchor holding capacity formulation involved empirical relationships related to anchor weight. For centuries, it appeared natural to mobilize resistance of the same order of magnitude as the submerged weight. The earliest investigations considering the surface area of the flukes as a significant parameter were conducted in the 1930s. The holding capacity, considered as the direct consequence of anchor-soil interaction,



(a) “Union Universal” 20t stockless anchor



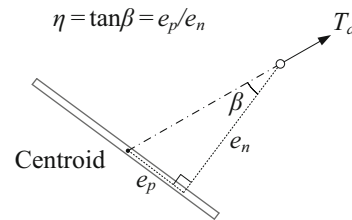
(b) LWT 13t stock anchor

Diving of Anchors,**Fig. 4** Sketches of anchors

is expected to be more intimately related to the bearing area rather than the anchor weight itself.

Figure 4 shows the sketches of a traditional drag embedded anchor (DEA), a vertically loaded anchor (VLA) and a suction embedded plate anchor (SEPLA). The traditional DEA has a fixed fluke rigidly connected to a shank. The angle between the shank and the fluke is pre-determined, though it may be adjusted prior to anchor placement on the seabed. The DEAs have a limitation of taking large vertical loads, and thus the so-called vertically loaded anchors have been developed, where the slim-line shank can be reconfigured relative to the fluke following initial embedment (Murff et al. 2005). The SEPLA is inserted into the tip of a suction caisson, which is then used to penetrate the anchor to the required depth (Dove et al. 1998; Wilde et al. 2001). After retrieval of the caisson, the initially vertical SEPLA is preloaded by tensioning the mooring chain, forcing the anchor to rotate – a process known as “keying.”

For simplification purpose, the soil resistance on the anchor shank is neglected, and the above anchors may be idealized in a generic form as in Fig. 5, where e_n denotes padeye (normal to fluke) eccentricity (the shortest distance between padeye and anchor fluke) and e_p represents padeye (parallel to fluke) offset relative to the fluke centroid. Tian et al. (2014) proposed a definition of padeye offset ratio η as e_p/e_n and highlighted its influence on the tendency for the anchor to dive under load.

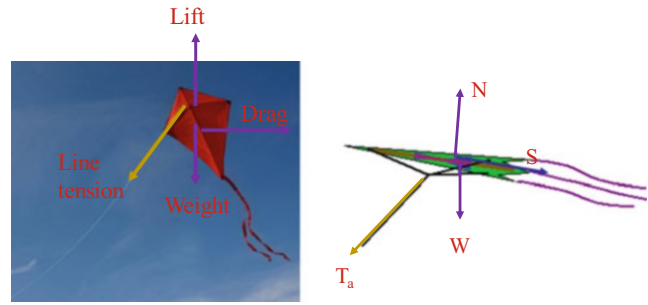
**Diving of Anchors, Fig. 5** Simplified calculation model**Analogous to Flying a Kite**

Although anchors seem to not have much connection to kites, the diving of anchor in soils can be interestingly analogous to flying kites in the air in terms of the mechanism. As shown in Fig. 6, there are four forces, i.e., lift N , weight W , drag S , and tension T_a , acting on the kite. Lift N is the upward force that pushes a kite into the air. Weight W is the downward force generated by the gravitational attraction of the Earth on the kite, which pulls the kite toward the center of the Earth. Tension T_a is the forward force that the line propels a kite in the direction of motion. Drag S is the backward force that acts opposite to the direction of motion. To launch a kite into the air, the force of lift N must be greater than the force of weight W . To keep a kite flying steady, the four forces are expected to be in balance.

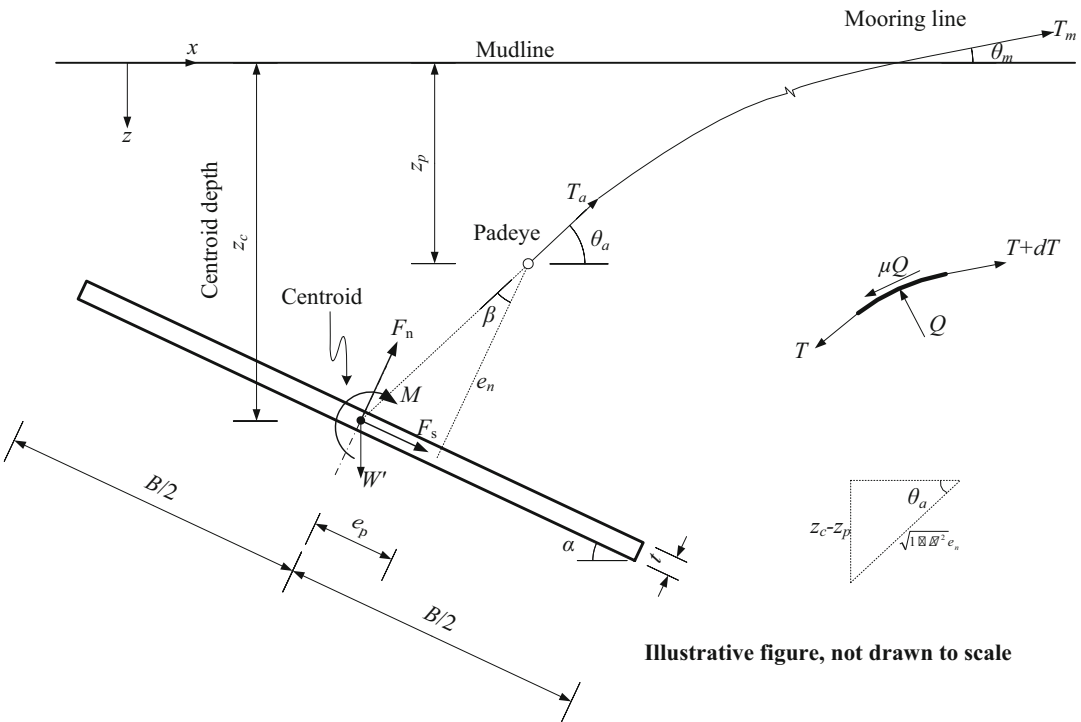
The efficiency of a good kite is largely dependent on the line attachment point and the “attacking angle” that the line connects to the kite. This is essentially the idea of optimizing the padeye position and the offset angle in Fig. 5.

Diving of Anchors,

Fig. 6 The forces acting on a kite



D



Diving of Anchors, Fig. 7 The mechanism controlling anchor diving

Diving Mechanism of Anchors in Soil

Similar as the mechanism in flying a kite, the mechanism dictating whether an anchor dive or not can be shown in Fig. 7. The anchor is simplified as a rectangle with height B , thickness t , eccentricity e_n , and padeye offset e_p . The anchor width is denoted as L , which is the dimension perpendicular to the plane of Fig. 7. The anchor inclination, α , is the angle between the anchor fluke and the horizontal. The anchor effective weight, equal to the difference between the anchor

submerged weight and the soil buoyancy, is denoted as W' and is assumed to act at the anchor fluke. The effective weight is approximated as $(\gamma'_a - \gamma'_s) BLt$, where γ'_a and γ'_s are the submerged unit weight of anchor and soil, respectively. The chain pulling force at padeye is defined as T_a at an angle θ_a to the horizontal. The chain pulling force at mudline is denoted as T_m and the corresponding chain angle θ_m . The tension T_a and effective weight W' result in normal force (F_n), sliding force (F_s), and rotational moment (M) acting at the fluke centroid. A segment of the anchor chain

is illustrated in Fig. 7 with tension T and normal soil resistance Q per unit length following the convention of Neubecker and Randolph (1995). The x - z coordinate system is as shown in Fig. 7 taking z as downward positive.

The dragging in process causes DEAs and VLAs to dive to reach an ultimate equilibrium state when they translate horizontally without diving further. The final equilibrium depth is termed “ultimate embedment depth,” which essentially determines the ultimate holding capacity. For SEPLAs and indeed other anchors, with careful choice of geometry, anchors can be designed to “dive,” i.e., embed deeper (Tian et al. 2014). The ultimate embedment depth z_c for an anchor model as simplified in Fig. 7 can be approximately evaluated using the below equation:

$$\frac{z_c}{B} = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \quad (1)$$

where:

$$a = (1 + \mu^2) \frac{E_n d N_c}{B}$$

$$b = 2N_t \frac{L}{B} \left[\cos \theta_a + \mu \sin \theta_a - e^{\mu(\theta_a - \theta_m)} (\cos \theta_m + \mu \sin \theta_m) \right] + 2(1 + \mu^2) \frac{E_n d N_c}{B} \left(\frac{s_{um}}{kB} - \sqrt{1 + \eta^2} \frac{e_n}{B} \sin \theta_a \right)$$

$$c = 2N_t \frac{L s_{um}}{B kB} \left[\cos \theta_a + \mu \sin \theta_a - e^{\mu(\theta_a - \theta_m)} (\cos \theta_m + \mu \sin \theta_m) \right] + (1 + \mu^2) \frac{E_n d N_c}{B} \left[(1 + \eta^2) \frac{e_n^2}{B^2} \sin^2 \theta_a - 2 \frac{s_{um}}{kB} \sqrt{1 + \eta^2} \frac{e_n}{B} \sin \theta_a \right]$$

where E_n is the effective width multiplier taking 2.5, d is the diameter of the chain bar, and N_c is a bearing capacity factor for the anchor chain taking ~ 7.5 , μ is the chain friction coefficient taking ~ 0.2 – 0.3 . The undrained shear strength, s_u , can be expressed as $s_{um} + kz$ to describe the increasing soil shear strength with depth for offshore sediments. $N_t = T_a / B L s_u$ is the normalized tension force; θ_a is load angle at padeye, explicitly evaluated as

$$\theta_a = \tan^{-1} \left(\frac{n N_{n \max}^q}{p q N_{s \max}^{n/p}} \frac{N_s^{q-1}}{N_n^{q-1}} \right) - \tan^{-1} \eta \quad (2)$$

where parameters $N_{n \max}$, $N_{s \max}$, and $N_{m \max}$ are uniaxial bearing capacity factors and parameters n , p , and q determine defining the soil resistance (see Tian et al. 2015 for details).

The Significance of Anchor Diving and Estimation of the Ultimate Holding Capacity

From primitive stone anchors to contemporary high-capacity anchors, anchoring systems have evolved over many centuries with the most significant design breakthrough driven by offshore oil and gas industry. Today, anchoring system plays a fundamental role in oil and gas exploitation (as the industry advances into deep waters and thus requires floating platforms) as well as in the burgeoning renewable energy industry, which is developing floating wind turbines and wave energy converters to produce electricity at the scale of 10–100 MW. As previously discussed, anchor holding capacity is essentially determined by the embedded depth and thus encouraging anchor diving into deeper soil and knowing the ultimate state are of significant meaning for offshore anchoring system developments.

Cross-References

- [Capacity of Suction Anchor](#)
- [Catenary Anchor Leg Mooring](#)
- [Catenary Mooring](#)
- [Design of Mooring System](#)
- [Drag Anchors](#)
- [Mooring Lines](#)
- [Mooring System](#)
- [Mooring System of Renewable Energy Devices](#)
- [Single Anchor Leg Mooring](#)
- [Single Point Mooring](#)
- [Spread Mooring](#)

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DLC – Design Load Cases

- [Design of Renewable Energy Devices](#)

DLL – Dynamic-Link Library

- [Analysis of Renewable Energy Devices](#)

DNV – Det Norske Veritas

- [Pile Capacity](#)

Doppler Current Profiler

- [Sonar Technology](#)

Doppler Velocity Log (DVL)

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)
- [Integrated Navigation](#)
- [Long Baseline Underwater Acoustic Location Technology](#)
- [Obstacle Avoidance Technology for Underwater Vehicle](#)
- [Sonar Technology](#)

Doppler Velocity Log Dead Reckoning (DVLDR)

- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)

Doppler Velocity Log for Navigation System in Underwater Vehicle

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Synonyms

Autonomous underwater vehicle (AUV); Dead reckoning (DR); Doppler velocity log (DVL); Doppler velocity log dead reckoning (DVLDR); Extended Kalman filter (EKF); Global Positioning System (GPS); Human occupied vehicle (HOV); Inertial measurement unit (IMU); Inertial navigation system (INS); Remotely operated vehicle (ROV); Strap-down inertial navigation system (SINS); Ultrashort baseline (USBL)

Definition

Doppler velocity log (DVL) is a precision instrument, used to measure water current velocities

over a depth range using the Doppler effect of sound waves scattered back from the seabed. The working frequency range of DVLs ranges from 38 kHz to several MHz. It is one of the widely used equipment for underwater vehicles to achieve high-precision navigation.

Scientific Fundamentals

For underwater vehicles, the bottom tracking feature can be used as an important component in the navigation systems. In this case, DVL uses the Doppler shift between the emitted sound wave and the received bottom reflection to measure the relative speed of the vehicle. This kind of instrument has good accuracy, high sensitivity, and can measure longitudinal and lateral speed. Moreover, the velocity of the vehicle is combined with an initial position, compass or gyro heading, and data from the acceleration sensor. The sensor data are combined to estimate the position of the vehicle, typically by using the Kalman filter method. DVL is used to provide navigation information for ships, remotely operated vehicles (ROVs), human occupied vehicles (HOVs), and autonomous underwater vehicles (AUVs).

Overview

With the development of ocean navigation technology, ship velocity was measured using the taffrail log. This consisted of a small propeller that was towed astern, the rate of rotation of which was measured using a counter. In the 1920s, the log uses the Pitot tube and makes a measurement of differential water pressure beneath the ship. In the middle of the twentieth century, the electromagnetic log was developed. In this log, a magnetic field was produced within a streamlined water-tight sensor unit. This field was present in the water, and the motion of the sensor through this field induces a voltage across the sides of the sensor which was proportional to velocity. In the late 1960s and early 1970s, the electromagnetic log had further development.

All the logs so far mentioned made their measurements of velocity relative to the water and not

relative to the seabed. Therefore, their readings were affected by ocean currents. Also, the electromagnetic log and the Pitot tube made measurements within the boundary layer and therefore suffer from further inaccuracies. It was in order to overcome these deficiencies that considerable attention had been given in the following years to the development of next-generation logs, which was the acoustic logs.

The electromagnetic log made using the principle of Faraday's electromagnetic induction made up for the defects of the hydraulic log incapable of measuring the backward speed and low measurement sensitivity. After that, with the development of electronic technology, corresponding improvements were made to the rotary and dynamic pressure log. In the late 1960s and early 1970s, the electromagnetic log had further development.

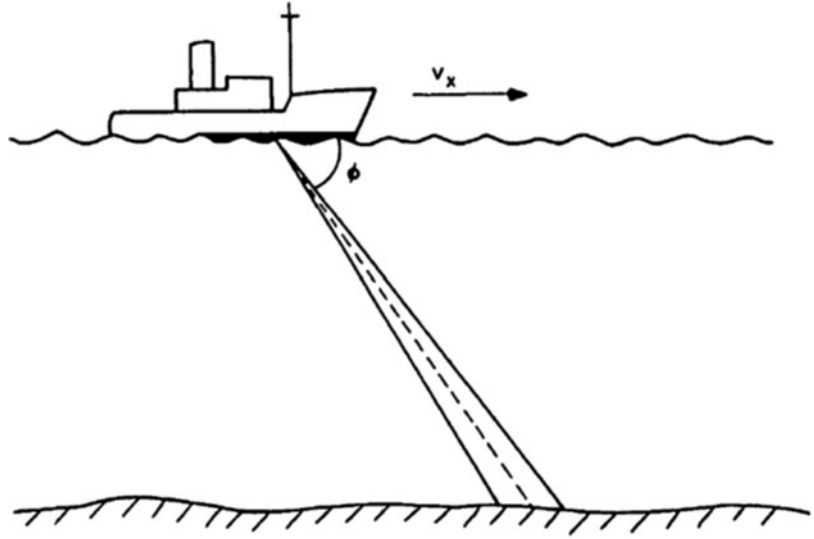
In the 1970s, Doppler velocity log products using the Doppler effect were successfully developed and quickly applied. In the next few years, the Doppler log had been well developed, and the commercial productions included those made by the companies, such as Amatek, Edo-Western, Krupp, Simrad, Furuno, Magnavox, Thomson CSF, True Digital, and Sperry. These logs had a wide range of operating frequencies, from approximately 100 kHz to 1 MHz. Ideally, they operated in the ground tracking mode in which the frequency of a seabed echo was measured to give an estimate of the velocity relative to the seabed. Unfortunately, the maximum depth of operation was very limited and when this was exceeded, they must transfer to the water tracking mode. In that mode the back scatter from within the water column itself was used, usually from quite short ranges, but still far enough away to be outside the boundary layer. This gave the Doppler velocity log a major advantage over the Pitot tube and the electromagnetic log (Denbigh 1984).

General Principles

Consider a vehicle moving with a forward velocity component v_x and radiating an acoustic pencil beam declined at an angle ϕ relative to this

Doppler Velocity Log for Navigation System in Underwater Vehicle,

Fig. 1 Basic geometry of Doppler velocity log.
(Denbigh 1984)



forward velocity vector, as shown in Fig. 1. The beam strikes the sea bed, and, because the ship has a velocity component $v_x \cos \phi$ in the direction of that region of the seabed, the back-scattered signal has a Doppler frequency shift f_D given by:

$$f_D = \frac{2v_x}{c} f \cos \phi$$

where f is the transmitted frequency of the Doppler velocity log and c is the velocity of the sound in the water.

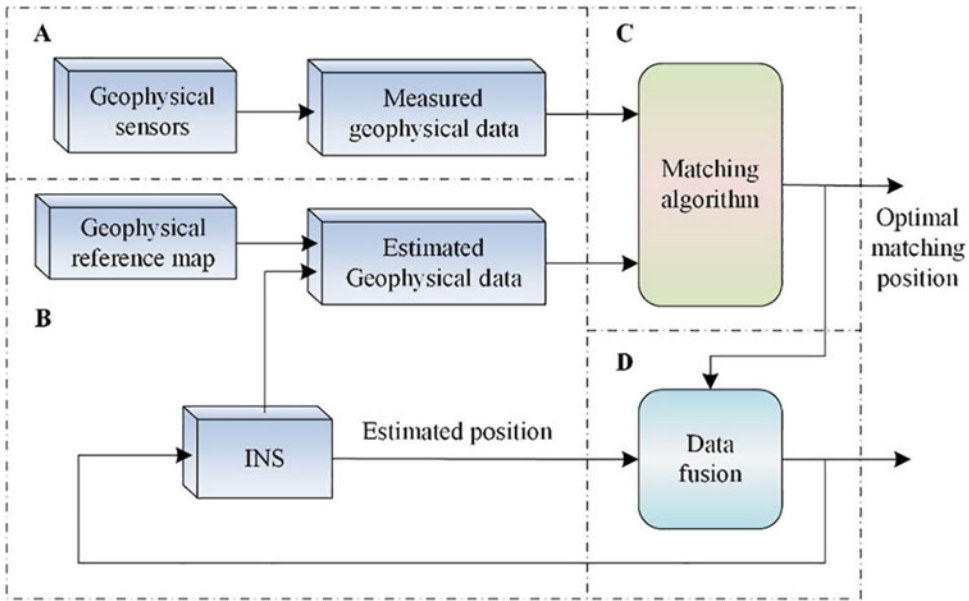
The principle of the Doppler velocity log is to measure the Doppler shift and to deduce the velocity, knowing the declination angle ϕ . The most appropriate value of ϕ is uncertain due to the finite beamwidth, and a major difficulty with the Doppler log is that of constraining the beamwidth to a sufficiently low value. A typical system would have a 3 dB beamwidth of 5 degrees.

In Doppler velocity log, the problem is aggravated by the considerable back scatter from micro-bubbles and particulate matter in the water, and the use of pulses is essential. A tone burst transmission is used, therefore, and an appropriate portion of the return signal is selected for frequency measurement. From the frequency shift, the forward velocity can be deduced using the given equation of f_D .

A major practical problem is that a vertical ship velocity, or heave, also contributes to the Doppler shift. Moreover, pitching is an additional problem as the beam angle ϕ is then no longer constant. The basic technique for overcoming these two effects is to use two beams: one facing forwards and one facing aft, and to measure the difference frequency. This is known as the Janus system. To a first approximation, the increase in frequency in one beam due to pitch and heave is accompanied by a similar increase in frequency in the other beam, so that subtraction of the two measurements gives a frequency which depends only on the required forward velocity. Small errors still remain, however (Denbigh 1984).

Key Research Findings

As the drift errors of inertial sensors are inevitable, the output of the pure inertial navigation system diverges over time. There are many optional sensors that can be used to suppress the inertial measurement units (IMUs) drift, among them DVL is often used to assist the underwater inertial navigation system (INS). The INS/DVL integrated navigation mainly uses the velocity of DVL to restrain the INS accumulation error. Currently, the integration of INS and DVL has even become



Doppler Velocity Log for Navigation System in Underwater Vehicle, Fig. 2 SINS/DVL integrated navigation structure. (Bao et al. 2020)

a standard configuration for AUV navigation (Bao et al. 2020).

In many applications, the DVL-aided inertial navigation system uses a depth sensor and compass as auxiliary sensors. Figure 2 shows a typical integrated navigation structure for AUV. The system framework contains a strap-down inertial navigation system (SINS), error state Kalman filter, aiding navigation sensors, such as DVL, compass, and pressure sensor. In Fig. 2, the error state Kalman filter is used to estimate the SINS navigation errors. The navigation solution and the navigation error estimates are independent of each other; the navigation error estimates are transmitted to the SINS to reset the navigation parameters. Subsequently, the corrected navigation parameters are used as recent initial conditions of the navigation equations (Jalving et al. 2004; Bao et al. 2020).

In underwater navigation, DVL is commonly installed in the bottom of AUV. The DVL malfunctions in the case of large rolls and pitches of AUV. In addition, the data update rate of DVL is low, and DVL usually works in a specific underwater environment, therefore, the application of DVL is occasionally limited. In many underwater

missions, the SINS/DVL is usually combined with GPS, ultrashort baseline (USBL), or other aiding means to enhance the navigation performance and reliability.

An integrated navigation system combining INS and DVL for an AUV named Urashima is presented, in which the feedback gain provided by the Kalman Filter is applied to the velocity error caused by INS and DVL. When the DVL becomes invalid in the water, the system uses only the INS output; therefore, it is more reliable than a dead reckoning system using the DVL. Furthermore, homing sonar and an acoustic receiver are used as navigation assisting devices to further improve the navigation accuracy (Ishibashi et al. 2004; Aoki et al. 2008). A SINS/DVL navigation system considering range measurement for AUV was proposed. Two measurement models based on acoustic range sensors were derived and augmented to the DVL-assisted SINS, respectively. Simulation results with experimental data demonstrate the improvement of the navigational performance and the robustness of the system to the dropout of acoustic sensor signals (Lee et al. 2004a; Lee et al. 2004b; Lee et al. 2007). Considering sea current estimation, DVL with water-



Doppler Velocity Log for Navigation System in Underwater Vehicle, Fig. 3 Explorer DVL mounted to the VideoRay Pro4. (Snyder 2010)

track is integrated with the INS by providing velocity aiding. Experimental results using real AUV data show that the accuracy and robustness of the integrated navigation system are significantly enhanced (Hegrenæs and Berglund 2009). The integration of IMU, DVL, and USBL is used for the Pirajuba AUV navigation, and the fusion of sample data given by the aforementioned sensors is accomplished through an asynchronous extended Kalman filter (EKF) (Zanoni and de Barros 2015).

DVL is a commonly used aiding source for the INS; however, the DVL must lock the seafloor to measure the AUV velocity, and DVL will fail to work in the mid-water zone or on a very rough seabed owing to the loss of bottom track. In some underwater missions, the SINS/DVL requires additional aiding to perform more precise and reliable navigation. Similar to inertial navigation, the DVL-aided INS is a dead reckoning positioning system. The SINS/DVL integrated navigation only reduces the growth rate of the position error with time for AUV and cannot alter the cumulative characteristics of the position error over time.

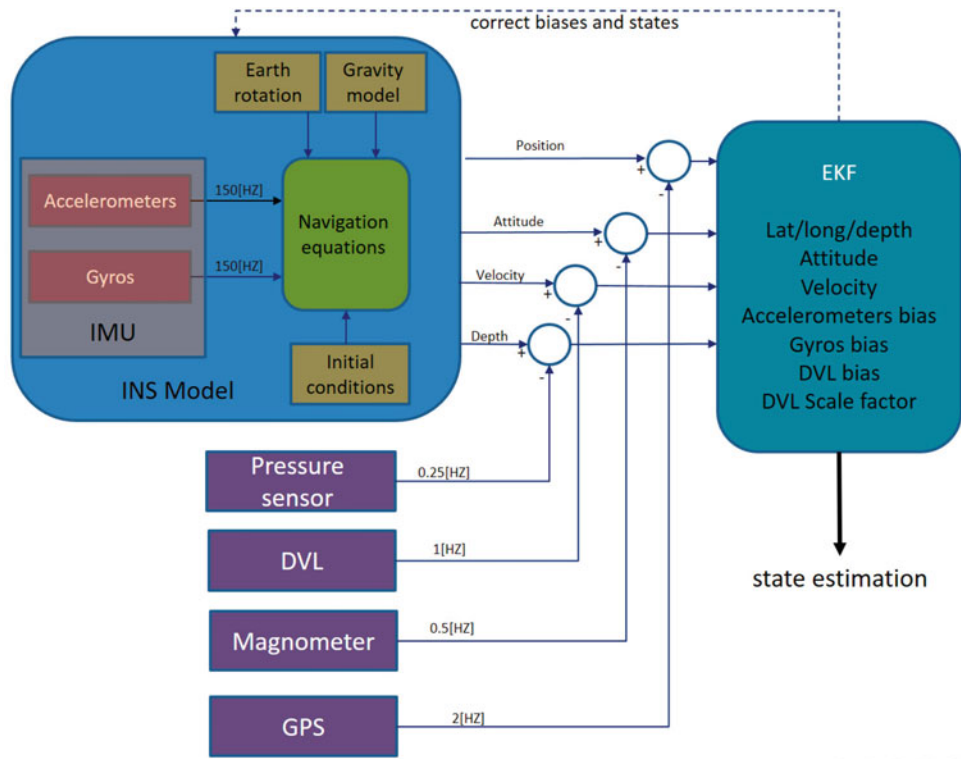
Examples of Application

Snyder (2010) has been working with Teledyne-RDI and VideoRay to integrate and test a self-contained, phased-array DVL on the VideoRay

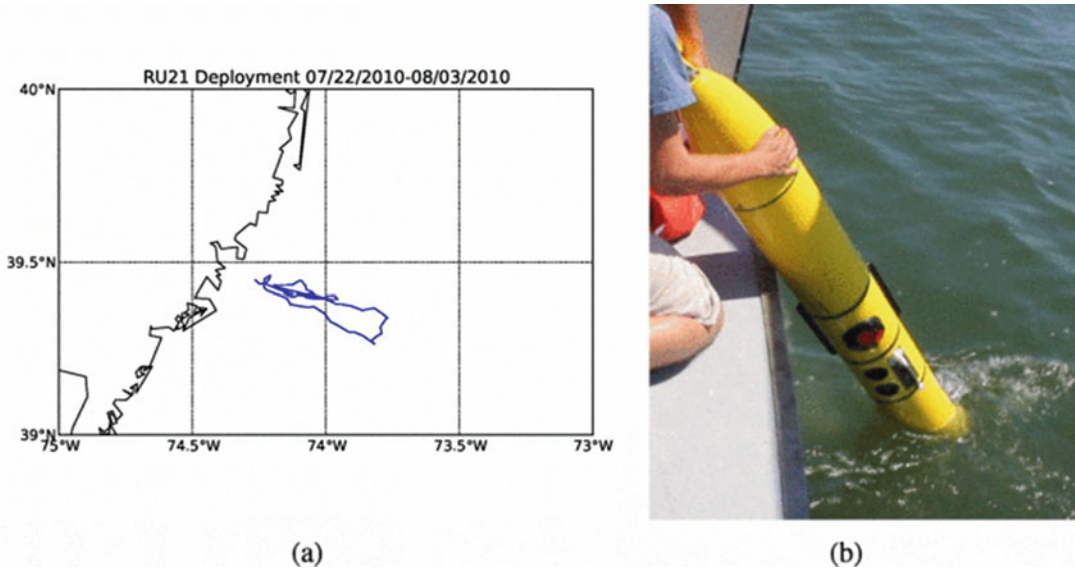
Pro4 mini-ROV, as shown in Fig. 3. The Pro4 has the demonstrated ability to carry the payload of the DVL, combined with the data bandwidth and the power supply to effectively host the navigation system and provide the necessary topside user interface. The integrated system performance in open water on a controlled course and the results of testing this integrated solution in a confined water tunnel are well obtained in the test experiment. Moreover, the application of the solution on a ship hull inspection are demonstrated.

Tal et al. (2017) have extended loosely coupled (ELC) approach for calculating vehicle velocity when only partial DVL measurements are available. This is made possible by using the partial measured raw data from the DVL combined with additional information. In order to examine and demonstrate the improvement of the navigation performance under the ELC approach, Tal et al. develop the TAUV six degrees of freedom (6DOF) simulation, which includes all functional subsystems, as shown in Fig. 4. This simulation consists of the AUV guidance, navigation, and control subsystems and uses a complete hydrodynamic model. Simulation results show great improvement in the navigation performance using the proposed approach compared to the standalone TAUV INS solution.

The Slocum Electric Glider is a buoyancy-driven autonomous underwater vehicle (AUV) capable of long-term deployments typically



Doppler Velocity Log for Navigation System in Underwater Vehicle, Fig. 4 Top level diagram of the TAUV navigation system. (Tal et al. 2017)



Doppler Velocity Log for Navigation System in Underwater Vehicle, Fig. 5 (a) The flight path of a glider deployment from July 22, 2010, to August 3, 2010, off the coast of New Jersey. (b) The vehicle was equipped with a DVL that performed bottom tracking throughout the mission. (Woithe et al. 2011)

lasting 4 to 6 weeks. During mission execution, the vehicle makes use of the global positioning system (GPS) to navigate to its commanded waypoints. GPS, however, can only be used while the vehicle is at the surface. While underwater, the glider uses a simple dead reckoning (DR) algorithm to estimate its location and does not find its true position again until its next periodic surfacing. Since there is limited sensory input to the algorithm, the vehicle's estimated global position can differ significantly from its true position. Woithe et al. (2011) explore the benefits that can be gained if the dead reckoning algorithm makes use of a Doppler velocity log (DVL) to improve a vehicle's location estimates. To gain a quantitative perspective as to how much a DVL may assist in the DR of a glider, Woithe et al. evaluate DVLDR algorithm with a previously flown deployment equipped with a DVL, as shown in Fig. 5. Testing results based on a deployment equipped with the DVL on a Slocum Glider show promising results.

Future Direction of Research

- 1) A small and low-priced DVL targeting micro ROV, UUV and AUV.
- 2) Providing more accurate data for navigation and station keeping.

Cross-References

- [Integrated Navigation](#)
- [Ship Navigation System](#)

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Down Hole Pressure and Temperature (DHPT)

- [Christmas Tree](#)

DP (Dynamic Positioning)

- [Semi-submersible Platform](#)

DP, Dynamic Positioning

- ▶ [Dynamic Positioning in Ice](#)
- ▶ [Numerical Simulation of Ice-Going Ships](#)

Drag Anchor

- ▶ [Mooring Anchor](#)

Drag Anchor, Fluke Anchor, Plate Anchor, or VLA

- ▶ [Drag Anchors](#)

Drag Anchors

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Synonyms

[Drag Anchor](#), [Fluke anchor](#), [Plate anchor](#), or [VLA](#); [FPSO](#); [Offshore Floating Module](#)

Definition

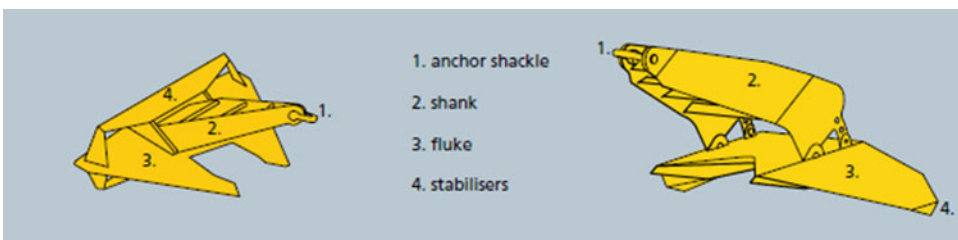
Drag anchors, which are evolved from conventional ship anchors, are usually penetrated into seabed by drag vessels to transmit an applied tensile load mainly through the fluke to a load bearing stratum.

Scientific Fundamentals

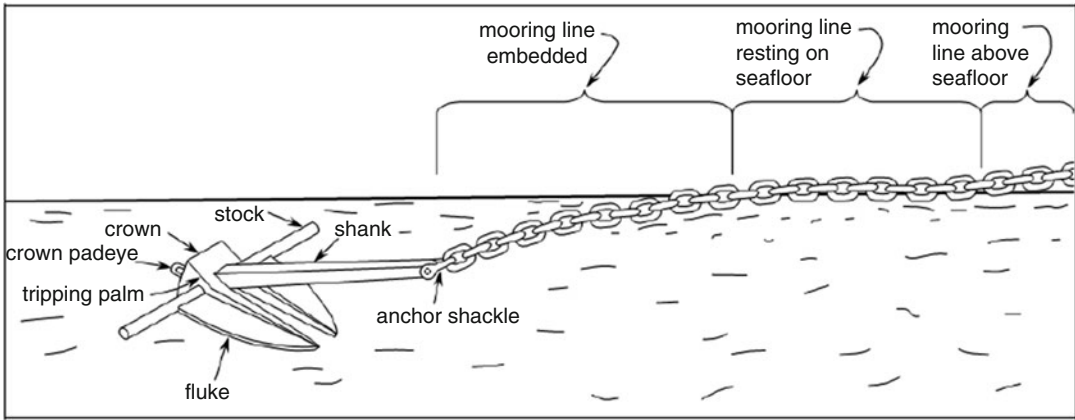
Types of Drag Anchors

Drag anchor is an important component of mooring system in deep water applications. It mainly consists of a broad fluke (providing main resistance, Fig. 1) and a shank (connecting the fluke to the mooring line, Fig. 2). Traditionally, the angle between them is fixed with a potential adjustment before penetration to the seabed but various in different soils, 50° for clay and 30° for sand, etc. (Randolph and Gourvenec 2011). The capacity of the drag anchor is mobilized primarily by the bearing resistance and side resistance on the anchor fluke and the friction along the embedded portion of mooring lines (Fig. 2), usually it can develop a typical holding capacity of tens of its self-weight by penetrating a depths of several times fluke lengths (Gao et al. 2015). However, this traditional fixed-angle fluke anchor is not designed to take the vertical force and hereby is suitable for catenary moorings rather than the taut or semi-taut line moorings in deep water applications.

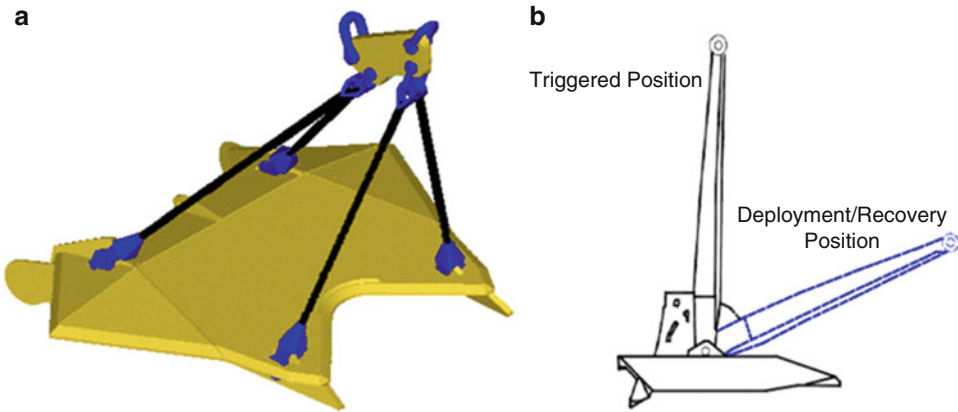
To overcome the limitation of the traditional drag anchor, the drag-in plate anchors, also called vertically loaded plate anchors (VLAs), were



Drag Anchors, Fig. 1 Schematic of Vryhof drag embedded anchor (Vryhof Anchors 2015)



Drag Anchors, Fig. 2 Components of DEAs (Thompson et al. 2012)



Drag Anchors, Fig. 3 Examples of VLAs (a) Vryhof Stevmanta (Vryhof Anchors 2015) (b) Dennla MK3 (Kim 2005)

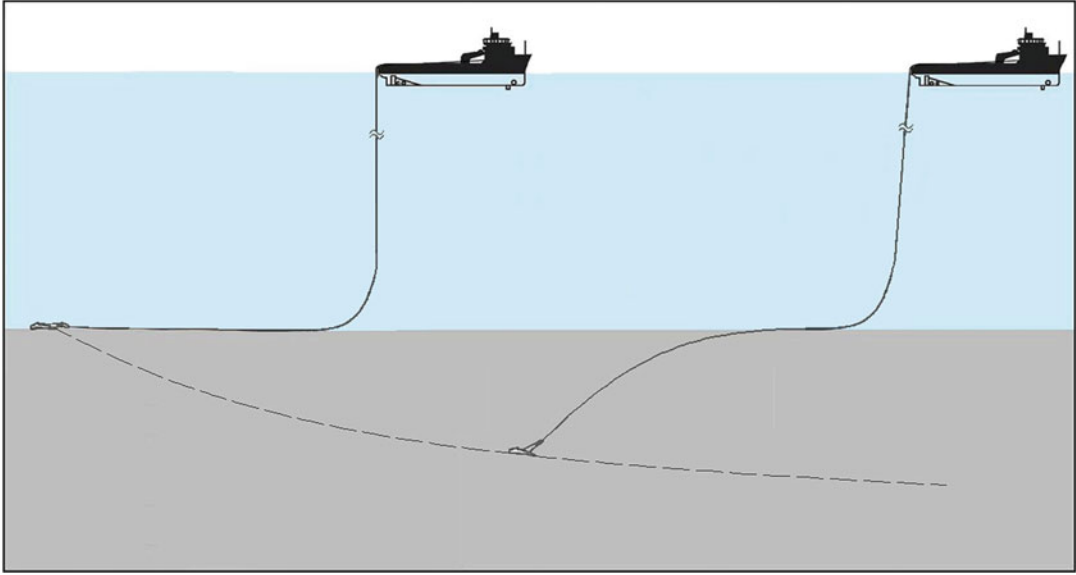
developed to withstand both large horizontal and vertical loads. Figure 3 shows the two commonly used VLAs: Vryhof Stevmanta VLA and Bruce Dennla anchor. It can be seen that the shank of VLA is thinner than that for traditional drag anchor, which makes VLA penetrate deeper into the seabed due to less resistance.

Installation of Drag Anchors

The installation process of the drag anchor can be seen in Figs. 2 and 4. Taking the conventional drag anchor as an example, from the starting point where the anchor shank and the mooring line lying on the seabed horizontally (Fig. 4), the anchor is penetrated into the seabed by the drag

force (Fig. 2) transmitted by the installation line from the vessel (Fig. 4). The installation of the drag anchor finishes once the predetermined installation load is reached.

For the penetration of VLAs, it is quite similar to that for the conventional fixed-fluke drag anchor but with an additional “triggering” or “keying” process at the proof load (e.g., typically 80–100% of the maximum intact mooring line load). By “triggering,” the anchor is rotated such that the mooring line or shank applies a force acting approximately normal to the fluke (Fig. 4). As a result, the anchor acts like a large embedded plate but with a high pull out capacity (Murff and Anderson 2001). It is worthy to note



Drag Anchors, Fig. 4 Sketch of the drag anchor installation (Zhang 2011)

that this “keying” process will cause the embedment loss of the VLAs (Randolph and Gourvenec 2011).

Holding Capacity of Drag Anchors

The holding capacity of a drag anchor depends on the ability to penetrate and reach the target installation depth. The determination of its value at a given depth is relatively straightforward and can follow the design charts based on highly empirical equations in which the ultimate capacity is related to the anchor weight in power form (API 2005; Vryhof Anchors 2015).

$$T_A = a \times \left(\frac{W_a}{C} \right)^b \quad (1)$$

where T_A is the anchor capacity in kips (1 kip = 4.448 kN); a is parameter for capacity in kips; W_a is anchor weight in kips; b is the exponent constant; and C equals to 10 kips.

It should be aware of that the design charts are for ultimate penetration of the anchor. However, anchors are in practice seldom or never installed to their ultimate depth (DNV 2017). Hence, to overcome the limitation of the design charts, further advanced approaches are needed. For the

drag anchor loaded normal to its surface, for example, VLA, which behaves like an embedded plate, the holding capacity could be calculated from the plasticity solution

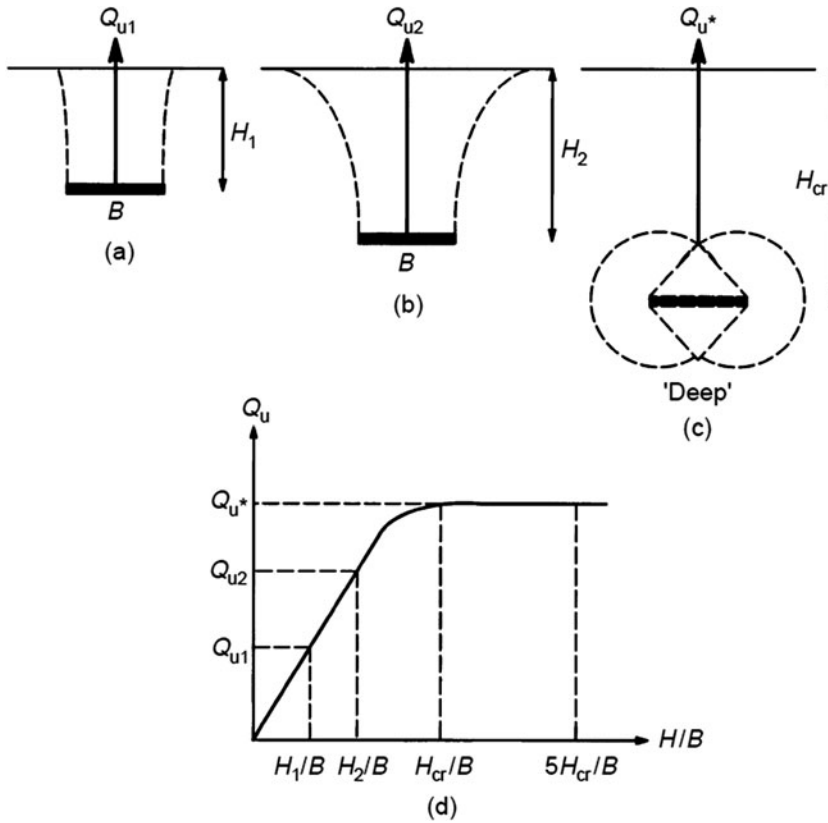
$$T_a = A_f N_c S_u \quad (2)$$

where T_a (i.e., Q_u in Fig. 5) is the holding capacity of the anchor; A_f is the fluke area; N_c is the bearing capacity factor taken as 12–13 for deep flow around anchor, $3 + \pi$ for shallow anchor, the roughness and the shape of the fluke can affect its value (Merifield et al. 2001); and S_u is the local undrained shear strength at fluke level.

For the conventional fixed-fluke drag anchor in engineering practice, its holding capacity is relatively complex due to a relative complex configuration and orientation, and could not be captured precisely by a simple equation in a form of Eq. 2 but would be determined during the prediction of the trajectory, since the conventional fixed fluke anchor mobilizes the holding capacity equal to the load required for installation.

Prediction in Trajectory of Drag Anchor

The installation and final location, which the holding capacity depends on, is the key to identify the



Drag Anchors, Fig. 5 Shallow and deep anchor behavior (Merifield et al. 2001)

working performance of the drag anchor. Methods for prediction in trajectory of drag anchor can be grouped into three categories: limited equilibrium method, plastic limit methods which include the yield envelope method and upper bound method, and analytical method.

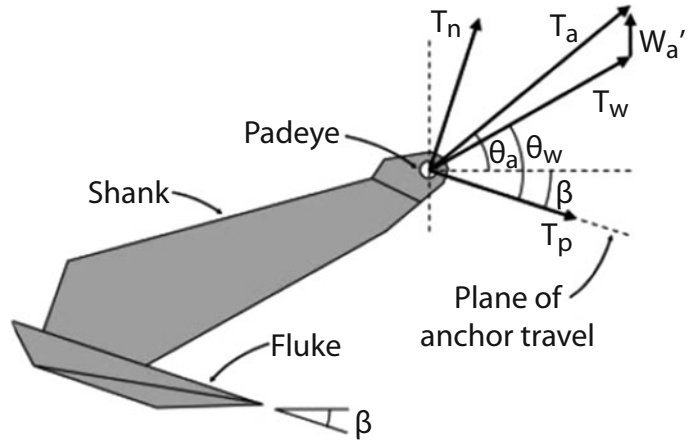
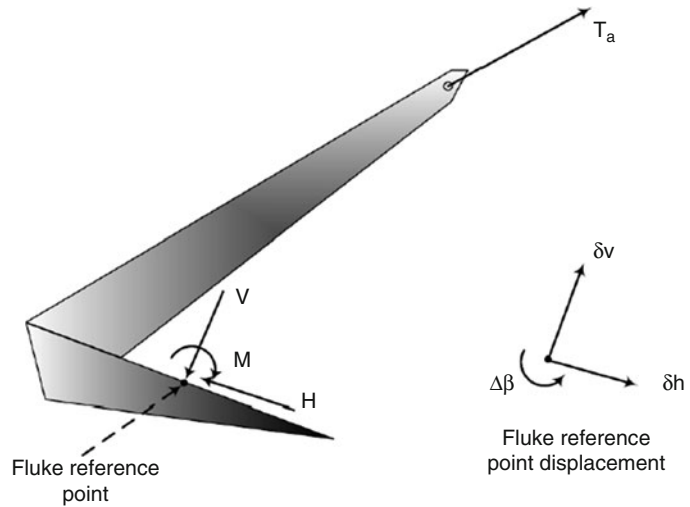
Limit equilibrium method is based on the assumption that when the drag anchor penetrating in and cutting the surrounding soil, the forces on the anchor and anchor line need to fulfill the equilibrium equations (Neubecker and Randolph 1996). In principle, the limit equilibrium method works in the following steps but will slightly change according to different assumptions and force models adopted:

1. From a given starting point (e.g., anchor shank horizontal, embedment zero, initial fluke angle β , advance the anchor through a horizontal increment).

2. Based on the assumption of motion parallel to the previous fluke orientation α , calculate the new anchor depth.
3. Calculate the anchor resistances $T_p = fA_p N_c S_u$ and the anchor capacity T_w if necessary according to different force models and the attached line angle θ_a .

For example, the force model proposed by Neubecker and Randolph (1996) is shown in Fig. 6, in which the fluke anchor θ_w and the form factor f are inherent properties of a drag anchor and the average values are 28° and 1.4, respectively.

4. Establish the moment equilibrium around the padeye and adjust the fluke angle β to fulfill the equilibrium equation
5. Repeat the above steps until the fluke angle $\beta = 0$, that is, the ultimate embedment depth is reached

Drag Anchors,**Fig. 6** Anchor force system for force model (O'Neill et al. 2011)**Drag Anchors,****Fig. 7** Loads and displacements at failure for a simplified drag anchor

Plastic limit analysis method based on yield envelop concept considers a simplified force model of fluke anchor, in which all loads are transferred to the mid-point (reference point) in V-H-M form (Fig. 7) and then carried out the finite element analysis to get a yield/failure envelop with fitted parameters (q, m, n, p):

$$\left(\frac{V}{V_{\max}}\right)^q + \left[\left(\frac{M}{M_{\max}}\right)^m + \left(\frac{H}{H_{\max}}\right)^n\right]^{\frac{1}{p}} - 1 = 0 \quad (3)$$

The method is similar to that of Neubecker and Randolph (1996). The only difference lies in the

application of the yield envelope and associated flow in determining the forces on the fluke and anchor displacements (Wu 2017). By ignoring the influence of shank force on the movement and the elastic displacements, the direction of fluke movement is calculated using the following equations after choosing the incremental distance δh :

$$\frac{\delta v}{\delta h} = \frac{\delta f}{\delta V} / \frac{\delta f}{\delta H}, \quad \frac{\delta \beta}{\delta h/B} = \frac{\delta f}{\delta (M/B)} / \frac{\delta f}{\delta H} \quad (4)$$

where B is the width of the fluke.

Upper bound of plasticity method in the prediction of the trajectory of the drag anchor assumes the anchor will penetrate into the soil volume following the path that the required

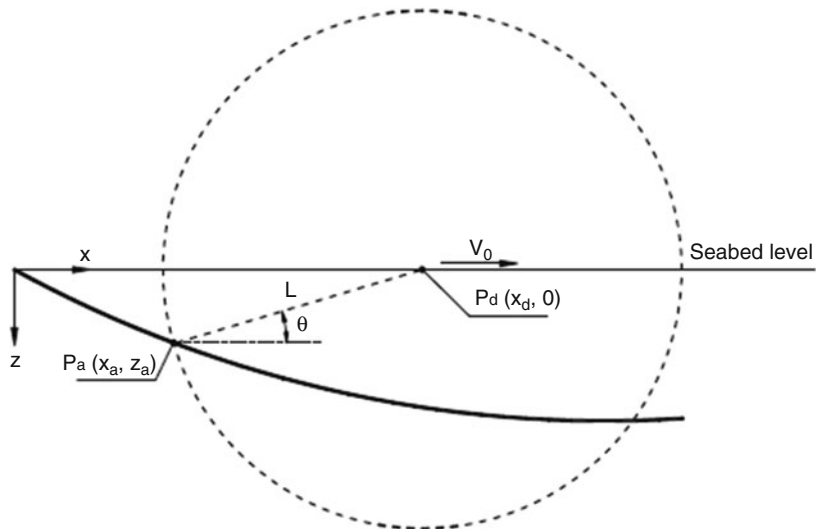
installation force T_a is minimum (Kim 2005). For a given position of the anchor, by equating the external work performed by the installation force T_a at the padeye and the anchor weight W to the internal energy dissipation in the soil, the minimum installation force T_a for a given anchor orientation θ_f can be obtained. By changing the anchor orientation and repeating this process, the locus relating minimum anchor line force $T_{a, \min}$ to θ_f is established. Furthermore, the inverse-catenary chain solution also gives a $T_a - \theta_f$ relation. The intersection of the two curves gives the unique values of anchor force T_a and inclination angle θ_f at a given step in the drag embedment process. Steps can be repeated until the fluke is parallel to the seabed. Note that the inverse catenary chain solution can be different according to different soil profiles (Wu 2017).

Analytic method (Liu et al. 2012) considered the installation of the drag anchor as a process in which the anchor penetrates following a circle path with a moving circle center $P_d (x_d, 0)$ (i.e., dragging by a vessel at a constant velocity, V_0) and the changing radius L , as shown in Fig. 8. Hence, the location of the anchor (x, z) needs to fulfill the equations below:

$$(x - x_d)^2 + z^2 = L^2 \quad (5a)$$

$$z = L \sin \theta \quad (5b)$$

Drag Anchors, Fig. 8 The kinematic model of the anchor



$$x_d = L \left\{ 1 + \cos \theta_a \left[\ln \left(\tan \frac{\theta_a}{2} \right) - \ln \left(\tan \frac{\theta_a - \theta}{2} \right) \right] \right\} \quad (5c)$$

in which θ_a is constant for a confirmed shape (Zhang 2011), the radius L and the angle θ can be determined by the reverse catenary chain solution.

The prediction of the trajectory is still challenging for the applications of drag anchors. Different methods could predict the trajectory quite different. The advantages and limitations of the methods are compared in Table 1 (Wu 2017).

Deep-Waters Applications of Drag Anchors

A summary of typical deep-water applications of drag anchors (including conventional fix-fluke drag anchors and VLAs) for both catenary and taut moorings in Oil&Gas industry is shown in Table 2. Most of the field applications are FPSO located offshore Brazil and Gulf of Mexico. Besides, the semi-submersible platform CNOOC 981 in China also adopted the drag anchor when its operating water depth is below 1500 m. Furthermore, since the application in the first floating spar-type floating windmill HYWIND), the drag anchor is gradually widely used in new energy field as the positioning system in offshore wind turbine as well.

Drag Anchors, Table 1 Advantages and limitations of different methods for anchor trajectory prediction (Wu 2017)

Method	Advantages	Limitations
Limit equilibrium method	The earliest method for anchor trajectory prediction based on geotechnical knowledge The method is based on the calculation of soil resistance on all the elements using given equations	Empirical constants which determined from field and laboratory tests are required Parametric study proved that the assumed factors for the resistance calculation influence the accuracy of prediction
Upper bound method	The method is based on the theory of plastic limit analysis, which is more efficient in solving the collapse problem	The application of the method is complex with requirement of comprehensive optimization analyses
Yield envelope method	Finite element analysis can be used for the yield envelopes study The prediction flowchart can be easily applied	The shallow anchor behavior is ignored
Analytical method	The method can be used for arbitrary fluke section in cohesive and noncohesive soil The method is based on a kinematic model and some mathematical expressions	The accuracy of the results from this method depends on the accuracy of the mathematical expressions, which needs to be improved

Drag Anchors, Table 2 Field applications of drag anchors (Zhang 2011)

Year	Field and type	Location	Water depth (m)	Anchor type	Fluke area (m ²)	Operator
1996	Liuhua 11-1 FPSO	South China Sea	310	Bruce FFTS Mk4	16.4	Amoco
1998	Voador P27 Semi-FPU	Offshore Brazil	530	Stevmanta	11	Petrobras
1999	Marlim South EPS FPSO-II	Offshore Brazil	1215	Bruce Dennla	10	Petrobras
1999	Roncador P36 Semi-FPU	Offshore Brazil	1350	Stevmanta	13	Petrobras
2000	Marlim P38 Semi-FSO	Offshore Brazil	1080	Drag anchors	–	Petrobras
2000	Marlim P40 Semi-FPU	Offshore Brazil	1080	Stevmanta	13	Petrobras

Cross-References

- [Catenary Mooring](#)
- [FPSO](#)
- [Offshore Wind Turbines](#)
- [Semi-submersible Platform](#)
- [Taut Mooring](#)

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Drag Embedment Anchor

- [Catenary Mooring](#)

D-Shackle

- [Mooring Connector](#)

DTF, Detection, Tracking and Forecasting

- [Dynamic Positioning in Ice](#)

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications

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Synonyms

[Crushing strength of sea ice](#)

Definition

Flexural strength of sea ice is the maximum bending stress that the sea-ice material can take. This definition is drawn from the traditional material mechanics. Once the sea-ice strength is surpassed, the failure and fracture of sea-ice plate are followed. The flexural strength is closely connected with many natural phenomenon and engineering properties of sea ice. For example, the breakup, rafting, and ridge building are often observed in polar area and seasonal ice-covered region, when the ice covers are under the actions of wave, current, and wind. In the cold region, the jacket platform with icebreaking cones, artificial island, slope protection, and icebreaker suffer from the collision of the ice cover. These situations generally cause the bending failure mode of sea ice, and the ice loads of the structures are subsequently determined by the flexural strength.

Uniaxial compression strength is one of the fundamental properties of sea ice. It defines the maximum compressive stress that the ice can bear before failure. The compressive strength is an important engineering property since it involved in most ice-structure interaction scenarios, where ice moves against structures.

Sea-ice ductile-brittle transition is one of the characteristics ice properties under compressive loading when the strain rate raises to a critical level. The effect is important because it marks the point where the compressive strength reaches a maximum and, thus, where the load from a given ice feature against another object approaches its highest value (Schulson and Buck 1995).

Scientific Fundamentals

The Influence of Porosity on the Uniaxial Compression Strength

The uniaxial compressive strength is the most common test for two reasons. First, for the many scenarios where ice moves against vertical structures, the ice load is dominated by the uniaxial strength. The other reason is that the uniaxial compression tests are relatively simple to perform.

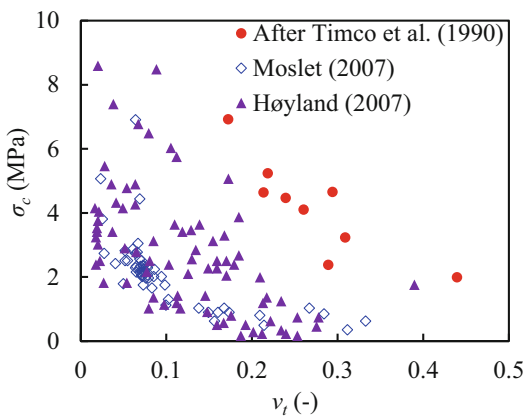
The compressive strength of sea ice depends on many factors: the brine volume, air bubble portion, loading direction (relative direction to grain columns), grain size, strain rate, temperature, and ice type (columnar or granular). In this section, we will focus on the influences of porosity, loading direction, and strain rate.

Natural sea ice contains pure ice, brine volume, and air. They have different capabilities to carry compressive forces. In general, the strength depends on the solid material. In another way, the unsolid part can be presented by the total porosity v_t :

$$v_T = v_b + v_a$$

where v_b is brine volume and v_a is air volume. The method to calculate the v_t is explained by Cox and Weeks (1983).

In Fig. 1, it gives the relationship between total porosity fraction and uniaxial compressive strength. It shows that the overall compressive strength has a negative correlation with total porosity. Compared with solid ice, the liquid brine and air are relatively weak for resisting compressive load. The higher porosity means lower solid material to resist the compressive force and thus results in lower compressive strength.



Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 1 Relationship between total porosity and uniaxial compressive strength under horizontal loading (Timco and Frederking 1990; Moslet 2007)

The Influence of Loading Direction on the Uniaxial Compression Strength

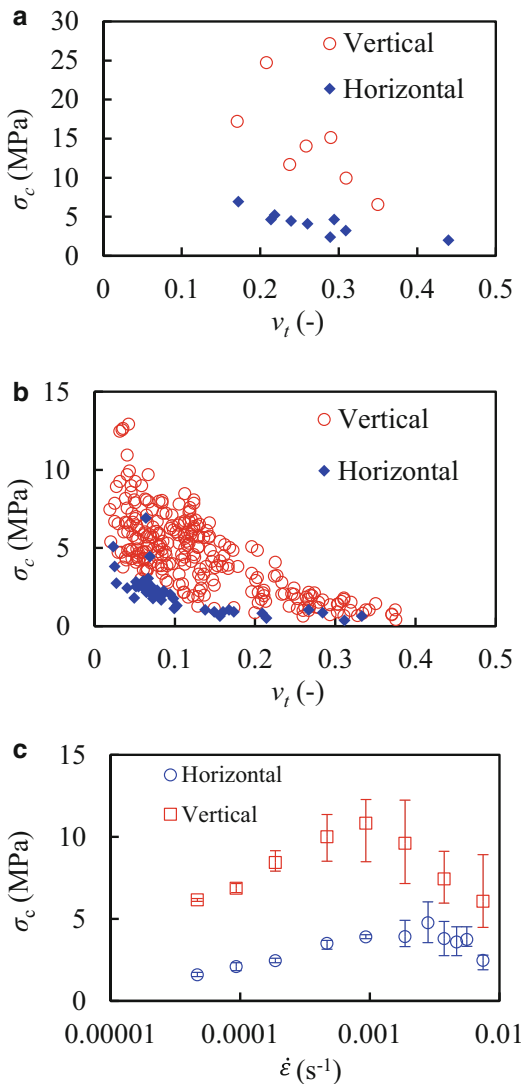
For the unbroken-level ice, the majority part owns columnar structure. The columnar grains make the mechanical properties depend on the angle between loading direction and grains. Due to the anisotropy of the columnar sea ice, the compressive strength may differ significantly when loading horizontally and vertically. The experimental results from different loading direction are given in Fig. 2. It shows that the vertical compressive strength is 4–5 times higher than horizontal compressive strength.

The Influence of Strain Rate on the Uniaxial Compression Strength

Field test on uniaxial compression strength indicated the strain rate or loading speed was one of the dominant factors for the strength value and failure mode. The exact value of strain rate for ductile-brittle transition and the peak value for compression strength might be different because of the variation sea-ice properties (such as microstructure, physical properties, etc.) in different sea area. Figure 3 shows some representative field test results of uniaxial compression strength versus strain rate of sea ice. It shows that the compressive strength increases with strain rate at low strain rate, where the sample fails in ductile manner. At the ductile-to-brittle transition, the strength reaches its maximum value. And the strength decreases with further increasing strain rate. The main mechanism for the different trends depends on the failure procedure as we shall introduce in the following section.

The Other Factors on the Uniaxial Compression Strength

The uniaxial compression strength may also be influenced by other factors such as grain size, environmental history, and ice type. In general, the strength has a positive correlation with the grain size. It is similar to the granular matter,



Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 2 The influence of loading direction on the uniaxial compressive strength. (a) Results from Timco and Frederking (1990); (b) results from Moslet (2007); (c) results from Ji et al. (2020)

since larger grains are more difficult to be rotated or to slide. The experiment results show that the sea-ice strength from different location may be different even if most of the parameters (porosity, strain rate, loading direction) are similar. It proves that the ice growth history may be important for its mechanical properties. The ice type is also important for the compressive strength. The granular sea

ice is normally considered as an isotropic material. The compressive strength of granular ice is between the vertical and horizontal strength of granular sea ice.

The Ductile-to-Brittle Transition of Sea Ice Under Uniaxial Compression

In section “[The Influence of Strain Rate on the Uniaxial Compression Strength](#),” it shows that the compressive strength is sensitive to strain rate. The correlation between strength and strain rate depends on the material behavior. Sea ice consists of liquid, gas, and solid.

Due to the different strain rates of sea ice under different loading rates, there are three different development trends of cracks in sea ice, and three different models of sea-ice damage are also presented in the macro-perspective (Table 1):

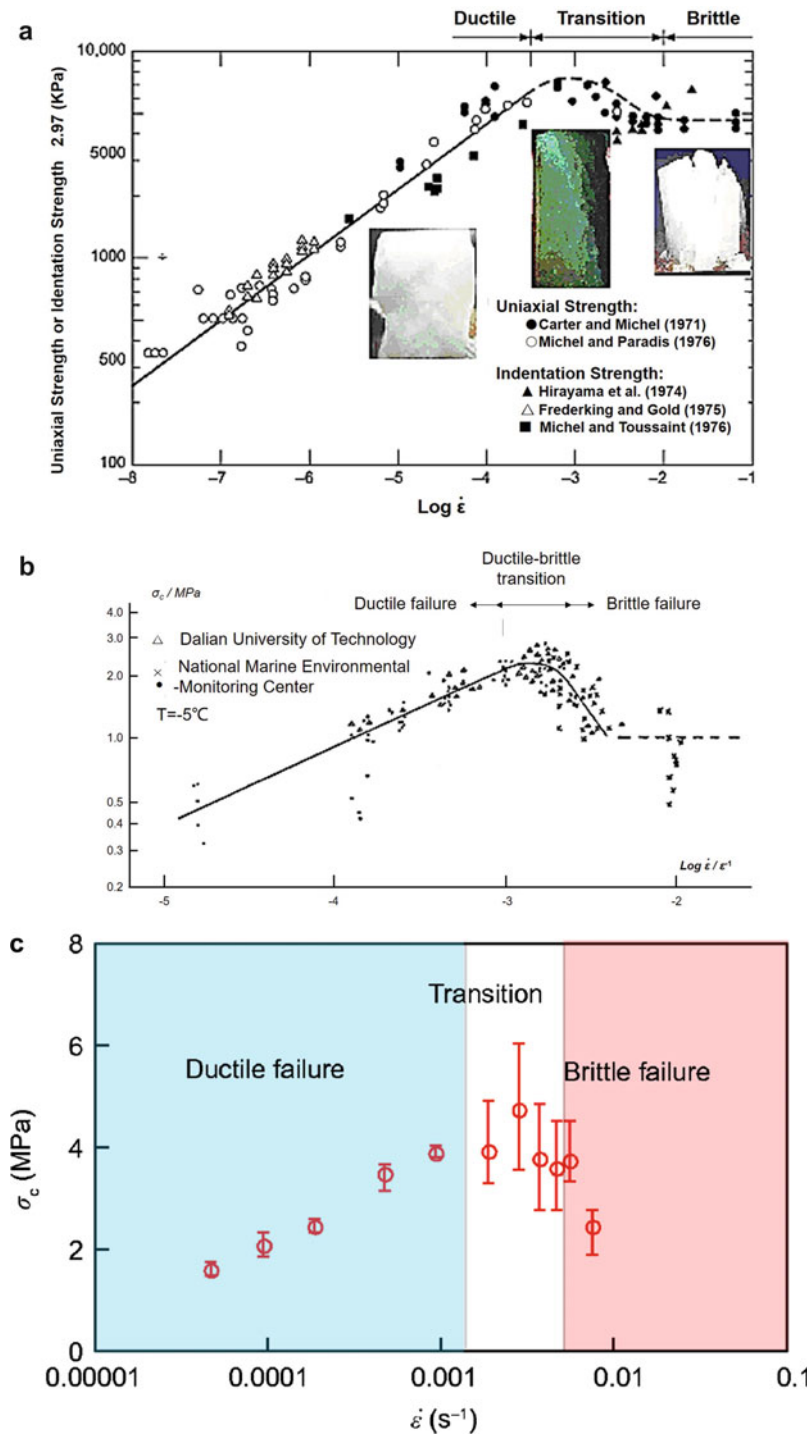
1. When the loading rate is low, the strength of sea ice increases with the increase of the loading rate, and the stress concentration at the crack tip in sea ice is released through creep, which inhibits the crack growth and finally presents a ductile failure mode.
2. When the strain rate is in the ductile-brittle transition, the crack inside the sea ice can grow stably along the principal axis and generate the main crack running through the entire sample, forming split failure, while the sea ice presents a tough-brittle transformation failure.
3. When the loading rate is high, the crack tip of the sea-ice sample cannot grow stably, so the oblique crack is generated and fragmentation is formed, while the sea ice presents a brittle failure mode.

Experimental value of sea-ice ductile-brittle transition could be obtained from the sea-ice compression test under axial action. The simultaneous testing values of axial strain, time history of loading, and indenter displacement were used to reveal the sea-ice behavior of ductile-brittle transition.

How the sea ice would behave during ductile-brittle transition? Let’s take field test on Feb. 6, 2018, in northeast of the Bohai Sea (N 40°07’16’’,

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications,

Fig. 3 Uniaxial compression strength versus strain rate of sea ice. (a) Representative results during 1971–1976. Basically from Yue et al. (2009) and Guo (2009). (b) 1989–1990 winter of Liaodong Bay in China. Basically from Yue and Li (1997). (c) Field test on Feb. 6, 2018, in the Baisha Bay of northeast of the Bohai Sea (N 40°07'16", E121°57'36"). Seawater salinity in the test site was about 33.2, which was higher than the average value of the Bohai seawater salinity, 30. Results from Chen et al. (2018)



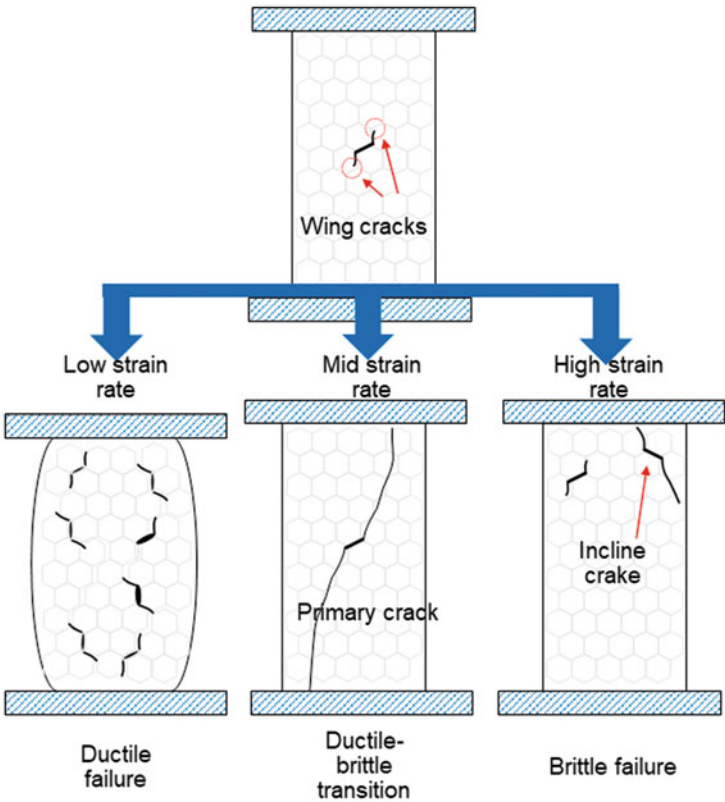
E121°57'36'') as an example. Figure 3 shows results of relationship between failure mode and compressive strength of sea ice. Figures 4, 5, and 6 show the multi-racks of sea ice under compression and the sea-ice fracture mode from ductile to spitting to spalling by the strain value increasing.

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Table 1 Physical mechanism for lock-in ice force and steady vibration during ice-structure interaction process

Loading phase	Relatively ice-structure velocity	Physical mechanism		
		Microscale: sea-ice internal micro-cracks	Mesoscale: ice material crushing property of ice	Macroscale: ice edge fracture behavior (contact with structure)
Loading phase	Keep in the same direction, with gradually decreased value	The number of micro-cracks increases as the ice load continues to increase not propagating due to the passivation effect at the crack tips	Under the front part of ductile-brittle transition	Be continuously compressed as complete ice body but not fracture
Loading-unloading transition	Relative rest	Crack saturation and prepare to run through	Under the transition part of ductile-brittle	Prepare to fracture to ice blocks
Unloading phase	Opposite direction	Primary crack run through	Under the latter part of ductile-brittle transition	Fracture to ice blocks which was crowded out by the lateral intact ice sheet

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications,

Fig. 4 Micro-crack development and failure mode under various loading/strain speed. Basically from Chen et al. (2018)

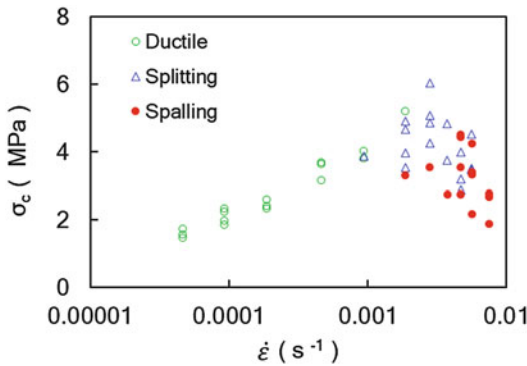


In the case test, the strain rate range for ductile-brittle transition 9×10^{-4} – $5 \times 10^{-3} \text{ s}^{-1}$ and the max value for compression strength was almost 6 MPa (Figs. 6–8).

Key Applications

Uniaxial Compressive Strength for Design Load on Vertical Structure

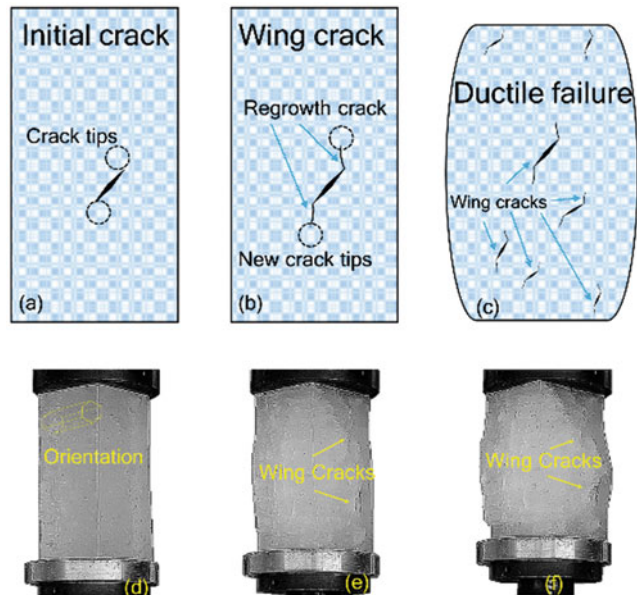
One important application of the uniaxial compressive strength is that it can be used as a



Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 5 The relationship between failure mode and compressive strength of sea ice. Results from Chen et al. (2018)

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 6

The ductile failure processes with multi-racks of sea ice under compression. Results from Chen et al. (2018)



parameter for the calculation of ice action on vertical offshore structures. For the ice floes against vertical structures, the ice is mostly likely to fail by compression between structure and following moving sea ice. The ice load is dominant by the compressive strength. The typical structure encountered sea ice are Molikpaq, Confederation Bridge piers, lighthouses in the Baltic Sea, platforms in the Bohai Sea, and so on.

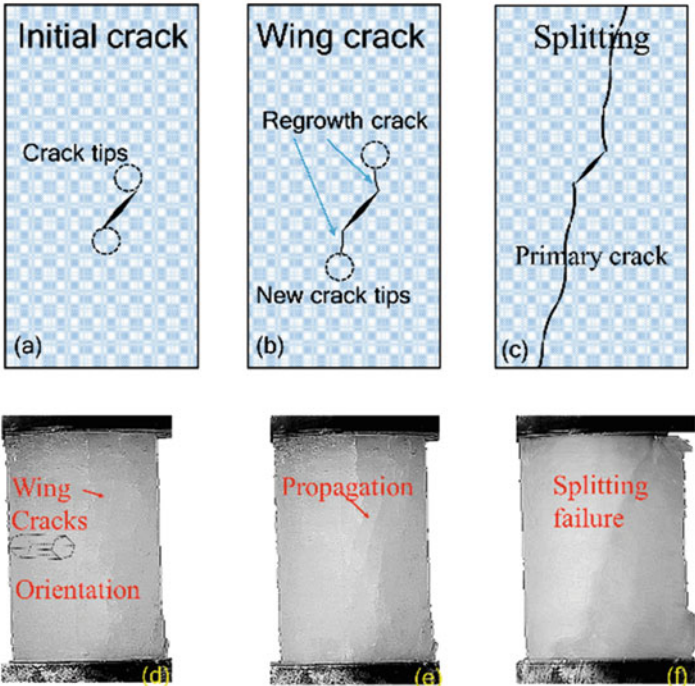
Three Modes of Ice Load and Structural Responses on Vertical Structure

Field measurement on ice-resistant structure and model test reveals that while the ice speed increases, three modes of ice load and structural responses occur: quasi-static (intermittent) mode, lock-in (steady state) mode, and random (continuous crushing) mode (Kärnä and Turunen 1990; Sodhi 1998; Yue et al. 2009; Guo 2009). ISO19906 recommended shows time histories of ice load and the structure's displacements in three different modes, shown as Fig. 9.

In ISO19906, frequency lock-in (depicted in Fig. 9b) was described as the following: It “can occur at intermediate ice speeds as the time-varying ice actions adapt to the frequency of the

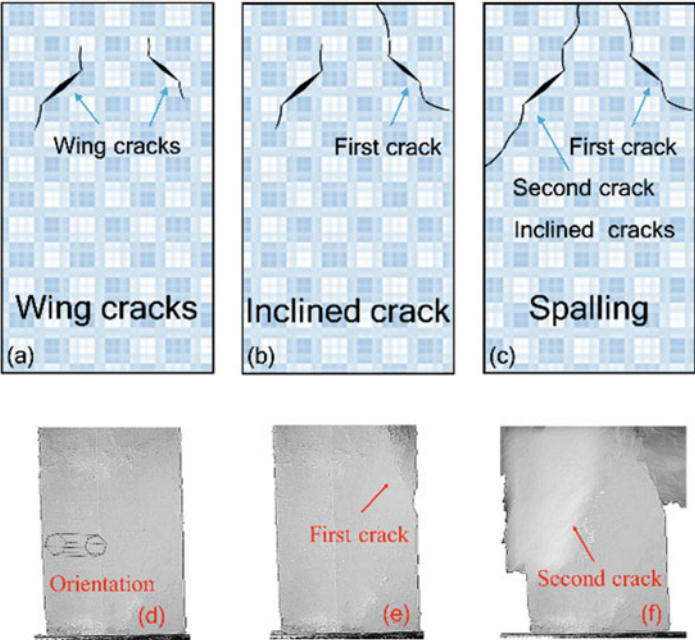
Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications

Applications, Fig. 7 The splitting failure mechanism of sea ice under compression. Results from Chen et al. (2018)



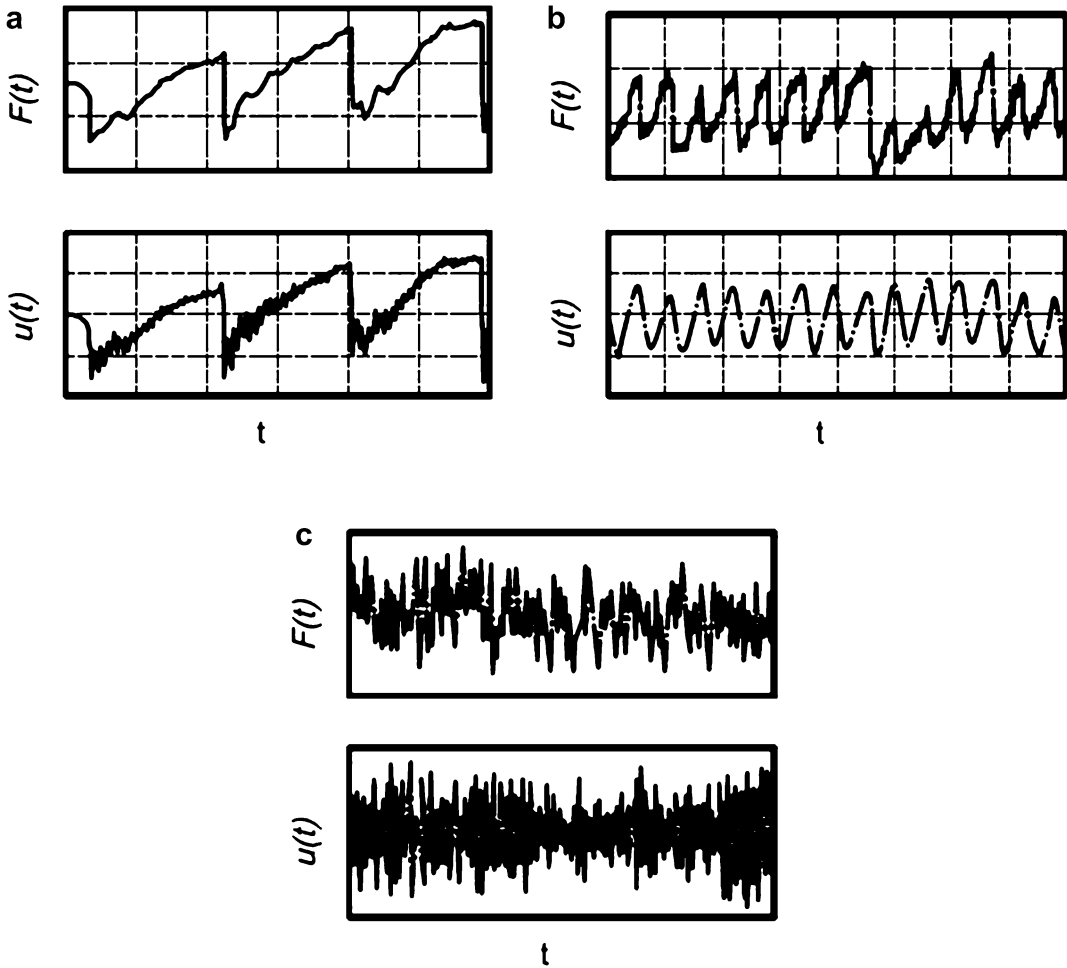
Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications

Applications, Fig. 8 The spalling failure mechanism of sea ice under compression. Results from Chen et al. (2018)



waterline displacements of the structure. The vibrations of the structure are typically sinusoidal in this condition. Similar to intermittent crushing,

the ice-structure interaction exhibits alternating phases of ductile loading and brittle unloading. The time history of the ice action depends on the



Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 9 Modes of time-varying action due to ice crushing

and the corresponding dynamic component of structural response (ISO19906 2019)

characteristics of the ice and the structure.” During the lock-in ice force and simultaneous similar resonant vibration, the frequency of ice load is locked into the vibrating frequency of structure, which is always the structure’s first natural frequency. So the steady vibration induced by lock-in ice force was also called self-excited vibration.

The ice speed for lock-in ice load is always in a special range. Field measurement on the jacket oil platform with vertical legs in the Bohai Sea shows that the ice speed is lower than 2 m/s for quasi-static (intermittent) mode, 2 ~ 4 cm/s for lock-in (steady state) mode, and higher than 4 m/s for random (continuous crushing) mode (Yue et al.

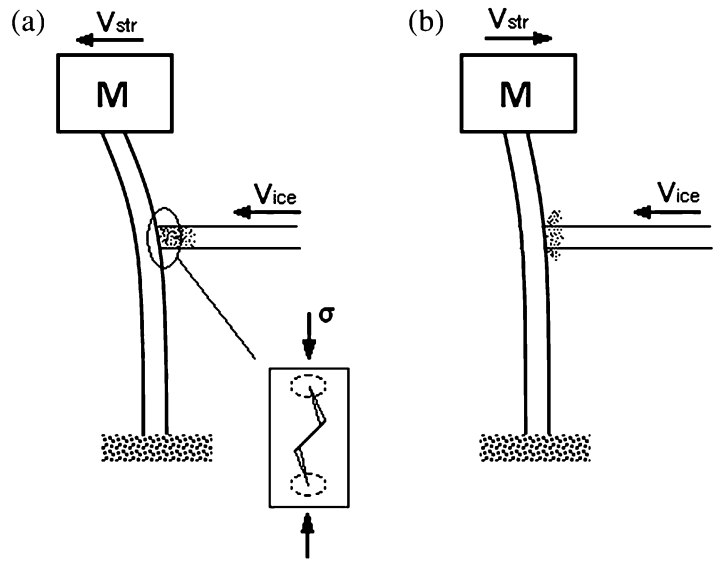
2009). And the time duration for the steady vibration and lock-in ice force could last from around 20 s to more than 600 s (Qu 2006).

Physical Mechanism for Lock-In Ice Force and Steady Vibration

The understanding of the physical mechanism for frequency lock-in ice force and steady-state vibration on vertical structure was in continuous development. In the 1970s and 1980s, the negative damping model (Blenkarn 1970; Maattanen 1983) was proposed. Later, a more complete

Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications,

Fig. 10 Sketch of physical mechanism of ice-induced self-excited vibration: (a) loading phase; (b) unloading phase (Yue et al. 2009)



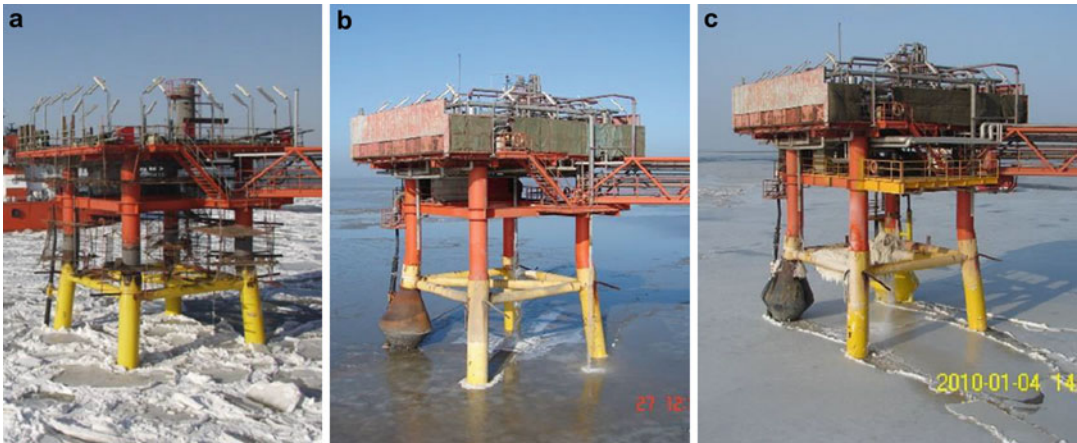
Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 11 Three-leg jacket platform before and after adding

cone in the Bohai Sea. (a) Without cone; (b) after adding cone (Xu et al. 2011)

understanding was proposed, which covers the microscale, development internal micro-cracks of ice; mesoscale, ice crushing properties under different strain rate; and macroscale, ice-structure relative velocity at water level, during acting on vertical ice. This theory was supposed by field measured ice crushing failure process and corresponding ice loads (Sodhi 1998, 2001; Kärnä and Turunen 1990; Yue et al. 2001).

Figure 10 demonstrates the real ice acting process on vertical structures, loading and unloading phase. The ice was moving forward at an almost constant speed (V_{ice}), and the structure was moving as a harmonic motion at an equilibrium position (structure velocity at waterline from $-V_{str} \sim +V_{str}$) (Table 1).

Only if the frequency of each period of loading and unloading phase matched the natural



Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications, Fig. 12 Single-leg jacket platform before and after adding

cone in the Bohai Sea. (a) Before 2004a, without cone; (b) 2004–2005 winter with single cone; (c) 2008–2009 winter with cones on two legs (Xu 2011)

frequency of structure, the repeat phenomenon of micro-cracks deformation in ice and ice force would be stabilized for a while.

Avoiding Ice-Induced Steady Vibration by Adding Cone

Steady-state vibrations were considered to be the most serious problem on ice-resistant vertical structures, which had been ever induced pipeline fracture and flange loosening of some offshore structure. One of the methods to resolve ice-induced steady vibrations was adding icebreaking cones at the water level, thus to change the ice failure mode from crushing to flexural. Several vertical oil jackets were added icebreaking cones to avoid steady vibration and lock-in ice force in the Bohai Sea. And field measurement verified the effect of mitigation of ice-induced vibration and avoidance of steady vibration by adding icebreaking cone (Xu 2011) (Figs. 11 and 12).

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Dunker

► Submarine

Dynamic Analysis Method

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Synonyms

Mooring system; Nonlinear effects; Synthetic lines; The lumped mass method; The rod theory

Introduction

Dynamic analysis method studies the dynamic response of the mooring line under the unsteady external environmental load. Besides, this method is to investigate the design system

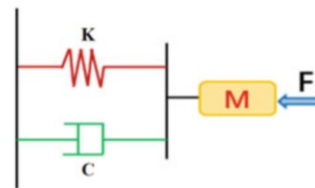
stabilization and to identify the stress of the mooring line is within the allowable stress range, which fulfill the mooring system with the specific mooring requirements. Unfortunately, it is difficult to determine, a priori, whether a floating offshore structure-mooring system will experience important dynamic loading. This determination must be made for each design problem on the basis of environmental site conditions, floating offshore structure size, and mooring arrangement.

Scientific Fundamentals

Dynamic analysis method accounts for the time varying effects due to mass, damping, and fluid acceleration. In this approach, the time-varying fairlead motions are calculated from the floating offshore structure's surge, sway, heave, pitch, roll, and yaw motions. Dynamic methods are used to predict mooring line responses to the fairlead motions. Importantly, dynamic methods include the additional loads from the mooring system other than restoring forces, specifically the hydrodynamic damping effects caused by relative motion between the line and fluid. As a simple model established in Fig. 1, following general equation of motion need to be solved.

$$M \frac{\partial^2 x}{\partial t^2} + C \frac{\partial x}{\partial t} + Kx = F$$

where **M**, **C**, and **K** are the matrices of hydrodynamic mass, damping, and stiffness, respectively, and **x** is the displacement vector. **F** represents the total sum of the forces in equation above.



Dynamic Analysis Method, Fig. 1 Simple model of mooring analysis

There are four primary nonlinear effects that can have an important influence on mooring line behavior (API RP 2SK 2005):

(a) **Nonlinear Stretching Behavior of the Line**

The strain or tangential stretch of the line is a function of the tension magnitude. Nonlinear behavior of this type typically occurs only in synthetic materials such as polyester. Chain and wire rope can be regarded as linear. In many cases, the nonlinearity can be ignored and a linearized behavior assumed, using a representative tangent or secant modulus.

(b) **Changes in Geometry**

The geometric nonlinearity is associated with large changes in shape of the mooring line.

(c) **Fluid Loading**

The Morrison equation is most frequently used to represent fluid loading effects on mooring lines. The drag force on the line is proportional to the square of the relative velocity (between the fluid and the line), hence is nonlinear.

(d) **Bottom Effects**

The interaction between the line and the seafloor is usually considered to be a frictional process and is hence nonlinear. In addition, the length of grounded line constantly

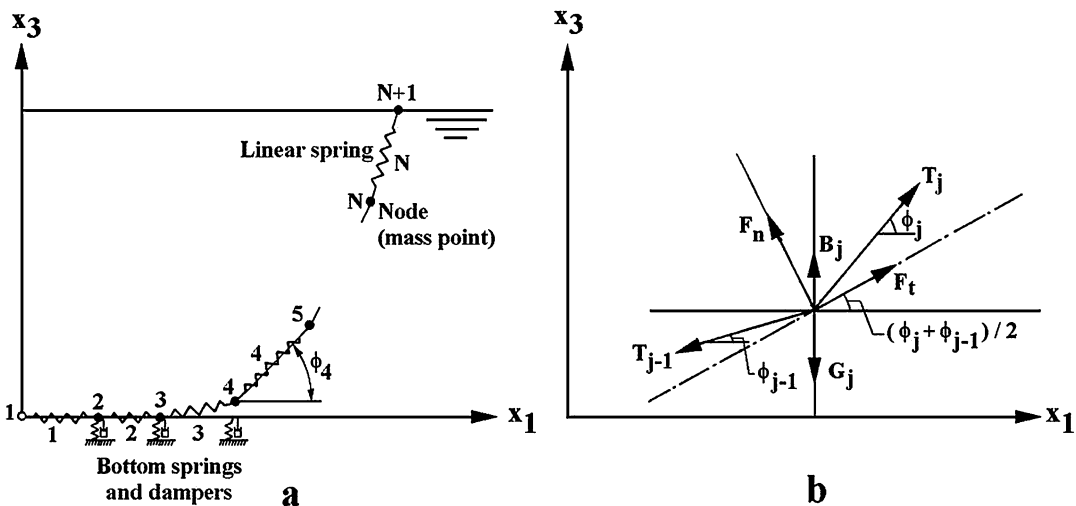
changes, causing an interaction between this nonlinearity and the geometric nonlinearity.

The lumped mass method and the slender rod theory are two typical methods to simulate the dynamic characteristics of mooring lines.

The Lumped Mass Method

Van den Boom (1985) describes a technique, which implies that the behavior of a continuous line is modeled as a set of concentrated masses connected by mass-less springs. This involves the lumping of all effects of mass, external forces, and internal reactions at a finite number of points along the line. By applying the equations of dynamic equilibrium (stress/strain) to each mass, a set of discrete equations of motion is derived. These equations may be solved in the time domain using finite difference techniques. Small effects of material damping, bending, and torsion moments are ignored.

Following the paper of Van den Boom (1985), the discretization of the mooring line can be obtained by lumping all forces to a finite number of nodes – so-called “lumped masses” – as given in Fig. 2a. The finite segments connecting the nodes are considered as mass less springs



Dynamic Analysis Method, Fig. 2 Discretization and definitions for lump mass method

accounting for the tangential elasticity of the line. The line is assumed to be fully flexible in bending directions. The hydrodynamic forces are defined in a local system of coordinates (tangential and normal direction) at each mass, as given in Fig. 2b.

In order to derive the equations of motion of the j^{th} lumped mass, Newton's law is written in global coordinates:

$$([M_j] + [m_j(\tau)]) \cdot \ddot{\vec{x}}_j(\tau) = \vec{F}_j(\tau) \quad (1)$$

where $[M_j]$ is the inertia matrix, $[m_j(\tau)]$ is the hydrodynamic inertia matrix, τ is the time, $\vec{x}_j(\tau)$ is the displacement vector, and $\vec{F}_j(\tau)$ is the external force vector.

The hydrodynamic inertia matrix $[m_j(\tau)]$ can be obtained from the normal and tangential fluid forces by directional transformations:

$$[m_j(\tau)] = a_j^n \cdot [\Lambda_j^n(\tau)] + a_j^t \cdot [\Lambda_j^t(\tau)] \quad (2)$$

where a_j^n and a_j^t represent the normal (superscript n) and tangential (superscript t) hydrodynamic mass:

$$a_j^n = \rho \cdot C_M^n \cdot \frac{\pi}{4} \cdot D_j^2 \cdot l_j \quad (3)$$

$$a_j^t = \rho \cdot C_M^t \cdot \frac{\pi}{4} \cdot D_j^2 \cdot l_j \quad (4)$$

In here, C_M^n and C_M^t are the nondimensional normal and tangential hydrodynamic mass coefficients.

$[\Lambda_j^n]$ and $[\Lambda_j^t]$ are directional matrices given below for the two-dimensional case:

$$[\Lambda_j^n] = \begin{pmatrix} +\sin^2 \vec{\phi}_j & -\sin \vec{\phi}_j \cdot \cos \vec{\phi}_j \\ -\sin \vec{\phi}_j \cdot \cos \vec{\phi}_j & +\cos^2 \vec{\phi}_j \end{pmatrix} \quad (5)$$

$$[\Lambda_j^t] = \begin{pmatrix} +\cos^2 \vec{\phi}_j & +\sin \vec{\phi}_j \cdot \cos \vec{\phi}_j \\ +\sin \vec{\phi}_j \cdot \cos \vec{\phi}_j & +\sin^2 \vec{\phi}_j \end{pmatrix} \quad (6)$$

$$\vec{\phi}_j = \frac{\vec{\phi}_j - \vec{\phi}_{j-1}}{2} \quad (7)$$

The nodal force vector, $\vec{F}$, contains contributions from the segment tension, T , the drag force, $\vec{F}_D$, buoyancy and weight, $\vec{F}_W$ and soil forces, $\vec{F}_S$:

$$\begin{aligned} \vec{F}_j(\tau) = & T_j(\tau) \cdot \vec{\Delta x}_j(\tau) - T_{j-1}(\tau) \\ & \cdot \vec{\Delta x}_{j-1}(\tau) + \vec{F}_{D_j}(\tau) + \vec{F}_{W_j}(\tau) \\ & + \vec{F}_{S_j}(\tau) \end{aligned} \quad (8)$$

where $\vec{\Delta x}_j(\tau)$ is the segment basis vector $(\vec{x}_{j-1} - \vec{x})/l_j$, in which l_j is the original segment length.

The drag force, $\vec{F}_D$, may be derived from the normal and tangential force components:

$$\vec{F}_{D_j}(\tau) = [\Omega(\tau)] \cdot \vec{f}_{D_j}(\tau) \quad (9)$$

$$f_{D_j}^n(\tau) = \frac{1}{2} \rho \cdot C_D^n \cdot D_j \cdot l_j \cdot u_j^n(\tau) \cdot |u_j^n(\tau)| \quad (10)$$

$$f_{D_j}^t(\tau) = \frac{1}{2} \rho \cdot C_D^t \cdot D_j \cdot l_j \cdot u_j^t(\tau) \cdot |u_j^t(\tau)| \quad (11)$$

$$\vec{u}_j(\tau) = |T_j(\tau)| \cdot (\vec{c}_j - \vec{x}_j(\tau)) \quad (12)$$

where

$\vec{f}_{D_j}$ = drag force in local coordinates

$\vec{u}_j$ = relative fluid velocity in local coordinates

$\vec{c}_j$ = current vector in global coordinates

ρ = fluid density

D = characteristic segment diameter

l = segment length

C_D = nonlinear (quadratic) 2-D damping (or drag) coefficient

C_D^n = normal drag coefficient

C_D^t = tangential drag coefficient

The directional matrices $[\Omega_j(\tau)]$ and $[T_j(\tau)]$ are used to transform the global drag forces and velocities into local drag forces and velocities:

$$[\Omega_j] = [T_j] = \begin{pmatrix} -\sin \vec{\phi}_j & \cos \vec{\phi}_j \\ \cos \vec{\phi}_j & \sin \vec{\phi}_j \end{pmatrix} \quad (13)$$

C_M is a 2-D nondimensional hydrodynamic mass coefficient. C_D is the 2-D quadratic drag coefficient. The hydrodynamic mass and drag coefficients C_M and C_D as they are used in the so-called Morison equations.

Van den Boom (1985) has derived the fluid reactive force coefficients a^n, a^t, C_D^n , and C_D^t from forced oscillation tests and free hanging tension tests with model chain and wire sections. A volumetric diameter, defined by $d_c = 2\sqrt{\nabla/(\pi l)}$ (with ∇ is segment volume and l is segment length), proved to be an accurate parameter in the dimensionless hydrodynamic coefficients. From the model tests it was concluded that frequency-independent coefficients could be used for normal mooring chains and wires.

Dynamic seabed reaction forces do not affect the behavior of the line and can be modeled as critical damped springs to prevent numerical instabilities. Tangential soil friction forces may be of importance when the line part on the bottom is extremely long and transverse soil reactive forces may be of importance for 3-D problems. Both soil effects are neglected here, thus the vertical force component, F_{s3j} , equals:

$$\begin{aligned} F_{s3j} &= -b_j \cdot \dot{x}_{3j} - c_j \cdot x_{3j} & \text{for } x_{3j} < 0 \\ F_{s3j} &= 0 & \text{for } x_{3j} > 0 \end{aligned} \quad (14)$$

The time domain relations between nodal displacements, velocities, and accelerations are approximated by finite difference methods such as the Houbolt scheme which is described by Bathe and Wilson (1976):

$$\begin{aligned} \vec{x}_j(\tau + \Delta\tau) &= \frac{1}{6\Delta\tau} \cdot \left\{ 11\vec{x}_j(\tau + \Delta\tau) - 18\vec{x}_j(\tau) + 9\vec{x}_j(\tau - \Delta\tau) + 2\vec{x}_j(\tau - 2\Delta\tau) \right\} \\ \vec{\ddot{x}}_j(\tau + \Delta\tau) &= \frac{1}{\Delta\tau^2} \cdot \left\{ 2\vec{x}_j(\tau + \Delta\tau) - 5\vec{x}_j(\tau) + 4\vec{x}_j(\tau - \Delta\tau) - \vec{x}_j(\tau - 2\Delta\tau) \right\} \end{aligned} \quad (15)$$

Or:

$$\begin{aligned} \vec{x}_j(\tau + \Delta\tau) &= \frac{5}{2}\vec{x}_j(\tau) - 2\vec{x}_j(\tau - \Delta\tau) \\ &+ \frac{1}{2}\vec{x}_j(\tau - 2\Delta\tau) \\ &+ \frac{1}{2}\Delta\tau^2 \vec{\ddot{x}}_j(\tau + \Delta\tau) \end{aligned} \quad (16)$$

The segment tension $T_j(\tau + \Delta\tau)$ is derived from the node positions by a Newton-Raphson iteration using the additional equation for the constitutive stress-strain relation:

$$\begin{aligned} \vec{\psi}_j(\tau) &= l_j^2 \\ &\cdot \left\{ \vec{\Delta x}_j(\tau) - \left(1 + \frac{T_j(\tau)}{E_j \cdot A_j} \right)^2 \right\} \end{aligned} \quad (17)$$

$$\begin{aligned} \vec{T}^{k+1}(\tau + \Delta\tau) &= \vec{T}^k(\tau + \Delta\tau) \\ &- [\Delta\psi^k(\tau)]^{-1} \cdot \vec{\psi}^k(\tau) \end{aligned} \quad (18)$$

where $\vec{\psi} = (\psi_1, \dots, \psi_j, \dots, \psi_N)$ is segment length error vector, $\vec{T}^k = (T_1, \dots, T_j, \dots, T_N)$ is tentative segment tension vector at the k^{th} iteration, and $\Delta\psi = \partial\psi/\partial T$ is the length error derivative matrix obtained from (16) to (17).

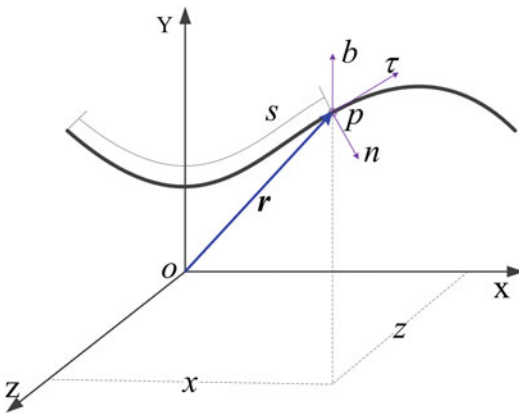
For each time step, the system of (18) should be solved until acceptable convergence of $T^k(\tau + \Delta\tau)$ is obtained. The initial tentative tension can be taken equal to the tension in the previous step. Each node j is connected to the adjacent nodes $j - 1$ and $j + 1$, hence (18) represents a three-diagonal $(N \times 3)$. The so-called Thomas algorithm may efficiently solve such equations. Van den Boom (1985) claims that – certainly for engineering applications – the Lumped Mass

Method (LMM) does provide efficient and accurate predictions of dynamic line motions and tensions.

The Slender Rod Theory

The slender rod theory is a famous beam model used for static and dynamic analysis of the mooring lines and risers in the ocean engineering. It is developed by Garrett (1982) in the nonstretched configuration and improved by Pauling and Webster (1986) into the small stretched case. The small stretched rod model is widely used to make programs, such as CABLE3D, WINPOST. And Chen et al. (2001) made an improvement on the stretch which is simulated with an inserted spring without bending stiffness. As we all known, there is bending stiffness in the polyester line. Webster et al. (2012) showed an improvement on the large stretch based on the nonstretched rod model with bending stiffness.

The mooring line in the space can be described with the radius vector, $\mathbf{r}(s)$, and arc length, s , in the global coordinate directly, see Fig. 3. The Garrett's rod theory and Paulling and Webster's small stretched rod model are created in the global coordinate. The large stretched rod model will be provided in the same way. In Fig. 3, τ , n , b are the unit tangent vector, the unit normal vector and the unit binomial vector, respectively. Webster et al.



Dynamic Analysis Method, Fig. 3 The coordinates of the rod model

(2012) give the motion equation and the constrained stretch equation for the large stretched rod as:

$$\left. \begin{aligned} &\gamma_{ikm}(s) \tilde{M}_{njm} \ddot{u}_{kj} + \\ &\alpha_{ikm}(s) \frac{B_m}{(1 + \varepsilon_m)^4} u_{kn} + \\ &\frac{\beta_{ikm}(s)}{1 + \varepsilon_m} \tilde{\lambda}_m(t) u_{kn} \end{aligned} \right\} = \mu_{im}(s) q_{mn}(t) + \tilde{f}_{in}^{nbc} \quad (19)$$

$$\frac{1}{2} \beta_{ikm}(s) u_{in} u_{kn} - \frac{1}{2} (1 + \varepsilon_w)^2 \eta_{wm}(s) = 0 \quad (20)$$

where ℓ is the element length in the reference configuration, B is the static bending stiffness of the element in the reference configuration, $\tilde{M}$ is the mass matrix, u is the position and slope of the two nodes of the element, ε is the strain, $\tilde{\lambda}$ is the Lagrangian factor, q is the distributed force, $\tilde{f}_{in}^{nbc}$ is the natural boundary condition, $n = 1..3$ is the index for the directions, $m, w = 1..3$ are the index for the points including the first node, the middle point and the last node in the element, $i, j, k = 1..4$ are the index for the positions and slopes, other coefficients are defined as:

$$\begin{aligned} \alpha_{ikm}(s) &= \int_0^\ell a_i''(s) a_k''(s) p_m(s) ds = \frac{\alpha_{ikm}(\xi)}{\ell^3} \\ \beta_{ikm}(s) &= \int_0^\ell a_i'(s) a_k'(s) p_m(s) ds = \frac{\beta_{ikm}(\xi)}{\ell} \\ \gamma_{ikm}(s) &= \int_0^\ell a_i(s) a_k(s) p_m(s) ds = \ell \gamma_{ikm}(\xi) \\ \mu_{im}(s) &= \int_0^\ell a_i(s) p_m(s) ds = \ell \mu_{im}(\xi) \\ \eta_{wm}(s) &= \int_0^\ell p_w(s) p_m(s) ds = \ell \eta_{wm}(\xi) \end{aligned} \quad (21)$$

The control equations can be solved with the finite element method with the numerical algorithm. In the dynamic case, the Moulton method is chosen as its explicit property and good performance. The first and second order derivative of the unknowns with respect to the time are:

$$\dot{\mathbf{d}} = \left\{ \sum_{n=1}^3 \sum_{k=1}^4 \frac{d(u_{kn}(t))}{dt}, \sum_{m=1}^3 \frac{d(\tilde{\lambda}_m(t))}{dt} \right\}^T \quad (22)$$

$$\ddot{\mathbf{d}} = \left\{ \sum_{n=1}^3 \sum_{k=1}^4 \frac{d^2(u_{kn}(t))}{dt^2}, \sum_{m=1}^3 \frac{d^2(\tilde{\lambda}_m(t))}{dt^2} \right\}^T \quad (23)$$

The control equations can be expressed in the matrix form as:

$$\tilde{\mathbf{M}}\ddot{\mathbf{d}} + \mathbf{C}\dot{\mathbf{d}} + \mathbf{K}_{\text{dynamic}}\mathbf{d} = \mathbf{F}_{\text{dynamic}} \quad (24)$$

in which the control equations are divided into two parts including the motion equations and the stretch constrained equations. Each matrix can be expressed as the addition of the two parts $\tilde{\mathbf{M}} = \tilde{\mathbf{M}}^{[1]} + \tilde{\mathbf{M}}^{[2]}$. For the motion equations, the matrixes are:

$$\tilde{\mathbf{M}}^{[1]} = \begin{bmatrix} \gamma_{ikm}(s)M_{njm}(t) & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \quad (25)$$

$$\mathbf{C}^{[1]} = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \quad (26)$$

$$\mathbf{K}_{\text{dynamic}}^{[1]} = \begin{bmatrix} \alpha_{ikm}(s)B_m & \beta_{ikm}(s)u_{kn}^{(0)} \\ +\beta_{ikm}(s)\tilde{\lambda}_m^{(0)} & \\ \beta_{ikm}(s)u_{in}^{(0)} & -\frac{\eta_{wm}(s)}{E_0A_w^t} \left(1 + \frac{\tilde{\lambda}_w^{(0)}}{E_0A_w^t}\right) \end{bmatrix} \quad (27)$$

$$\begin{aligned} \mathbf{F}_{\text{dynamic}}^{[1]} &= \mathbf{F}_{\text{dynamal}}^{(1)} + \mathbf{F}_{\text{dynall}}^{(0)} \\ &= \begin{bmatrix} \mu_{im}(s)q_{mn}^{\text{dynamic}}(t) + f_{in}^{nbc} \\ \mathbf{0} \end{bmatrix} \\ &- \begin{bmatrix} \alpha_{ikm}(s)B_m u_{kn}^{(0)} + \beta_{ikm}(s)\tilde{\lambda}_m^{(0)} u_{kn}^{(0)} \\ \frac{1}{2}\beta_{ikm}(s)u_{in}^{(0)}u_{kn}^{(0)} - \frac{1}{2}\eta_{wm}(s) \left(1 + \frac{\tilde{\lambda}_w^{(0)}}{E_0A_w^t}\right)^2 \end{bmatrix} \end{aligned} \quad (28)$$

where the upper symbol (0) means the time t_n , the (1) means the time $t_{n+1} = t_n + \Delta t$, Δt is the timestep. $\mathbf{C}$ is the structural damping matrix. In the rod system, there are different types of dampings including damping from fluid, damping from ocean bottom, damping from material, and structural damping. The first two dampings are implemented as the external forces like drag force and friction force. The damping from material is calculated with the visco-elastic construction equation as the creeping and relaxation. The last damping is written as the damping matrix $\mathbf{C}$.

For the constrained equations, the matrixes are:

$$\tilde{\mathbf{M}}^{[2]} = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \quad (29)$$

$$\mathbf{C}^{[2]} = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \quad (30)$$

$$\mathbf{K}_{\text{dynamic}}^{[2]} = \begin{bmatrix} \mathbf{0} & \mathbf{0} \\ \beta_{ikm}(s)u_{in}^{(0)} & -\frac{\eta_{wm}(s)}{E_0A_w^t} \left(1 + \frac{\tilde{\lambda}_w^{(0)}}{E_0A_w^t}\right) \end{bmatrix} \quad (31)$$

$$\begin{aligned} \mathbf{F}_{\text{dynamic}}^{[2]} &= \mathbf{F}_{\text{dynamal}}^{(1)} + \mathbf{F}_{\text{dynall}}^{(0)} \\ &= \begin{bmatrix} \mathbf{0} \\ \mathbf{0} \end{bmatrix} - \begin{bmatrix} \mathbf{0} \\ \frac{1}{2}\beta_{ikm}(s)u_{in}^{(0)}u_{kn}^{(0)} - \frac{1}{2}\eta_{wm}(s) \left(1 + \frac{\tilde{\lambda}_w^{(0)}}{E_0A_w^t}\right)^2 \end{bmatrix} \end{aligned} \quad (32)$$

The Moulton method is applied and finally gets the corresponding iterative equations in the time domain as:

$$\begin{aligned} &\left(\tilde{\mathbf{M}}^{(\frac{1}{2})} + \mathbf{C}^{(\frac{1}{2})} \frac{\Delta t}{2} + \mathbf{K}_{\text{dynamic}}^{(\frac{1}{2})} \frac{\Delta t^2}{4} \right) \ddot{\mathbf{d}} \\ &= \mathbf{F}_{\text{dynamal}}^{(\frac{1}{2})} + \mathbf{F}_{\text{dynall}}^{(0)} - \left(\mathbf{C}^{(\frac{1}{2})} \dot{\mathbf{d}}(t_0) + \mathbf{K}_{\text{dynamic}}^{(\frac{1}{2})} \dot{\mathbf{d}}(t_0) \frac{\Delta t}{2} \right) \end{aligned} \quad (33)$$

in which the upper symbol $(\frac{1}{2})$ means the median time $t_n + \frac{1}{2}\Delta t$.

Cross-References

- [Mooring System](#)
- [Static Analysis Method](#)
- [Synthetic Lines](#)
- [The Lumped Mass Method](#)
- [The Rod Theory](#)

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Dynamic Behavior and Fatigue

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Dynamics of Flowline/Riser Systems

Definition

Pipeline and riser systems are one of the offshore production equipment that plays a vital role in the

transportation of oil and natural gas from subsea wellhead to topsides of the floating units/vessels. Basically, risers are conductor pipes that connect floaters on the sea surface and the wellheads at the seabed. They experience different sources of dynamic loadings coming from wave, currents, VIV, hydrodynamics, vessel motions, and slugs which cause instabilities and vibrations of the system.

Scientific Fundamentals of Riser Structures

Riser Description and Configurations

There are mainly two kinds of risers which are rigid risers and flexible risers. A hybrid riser is the combination of these two.

Rigid Risers

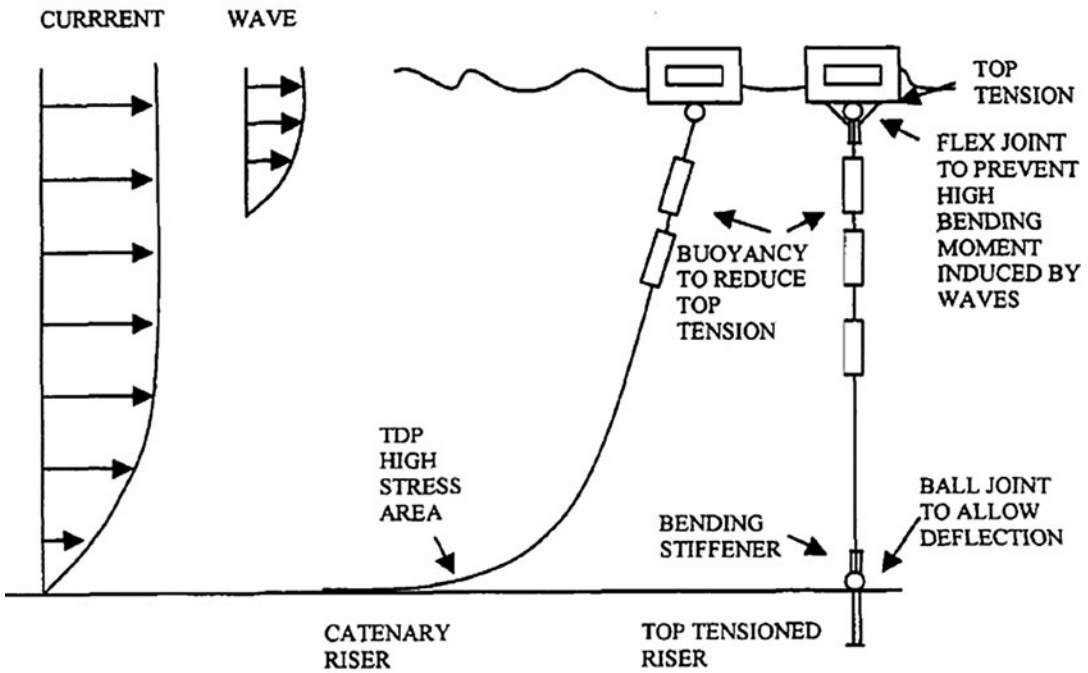
Steel catenary risers (SCR) and top tensioned risers (TTR) are various forms of rigid riser systems. In practice, the top tensioned risers are deployed in shallow waters, but as design for higher water depth is accounted, a new design practice led to the adoption of SCR as shown in Fig. 1 (Bai and Bai 2005).

TTR is very sensitive to heave motion due to wave and current loads because rotation at the top and bottom connections is limited and requires a heave compensator between the riser top and the floater. However, if the top tension is reduced, it will induce larger bending moment along the riser most especially if the riser is operated in an environment with strong current. If it happens the effective tension becomes negative (i.e., compression), then Euler buckling will occur.

The SCR riser is self-compensated for the heave motion, i.e., the riser is lifted off or lowered on the seabed. The catenary riser is sensitive to environmental loads, i.e., wave and current due to the normally low effective tension in the riser.

Flexible Risers

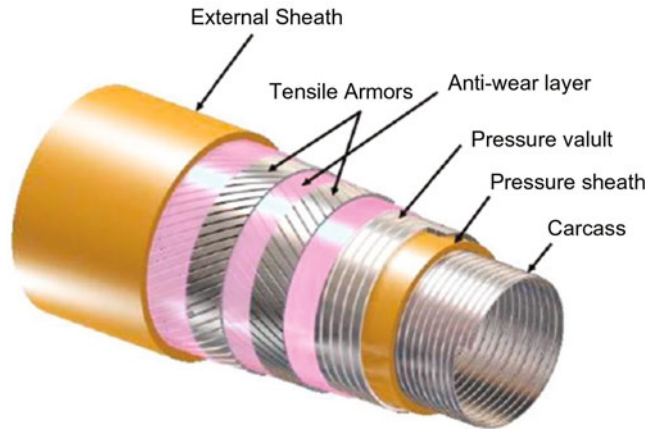
Flexible risers unlike rigid risers are flexible pipes that offer a robust means of producing oil and natural gas in harsh environments. They are capable of accommodating a wide number of



Dynamic Behavior and Fatigue, Fig. 1 Top tensioned risers and steel catenary risers and their components (Bai and Bai 2005)

Dynamic Behavior and Fatigue, Fig. 2

Description of layers in an unbounded flexible pipe (Mac and Sicsic 2018)



different conditions ranging from shallow and deep waters to horizontal and vertical movements; hence, it is suitable for use with floating facilities. Flexible risers come as a layered structure (Fig. 2) with various materials having different functions.

Each layer of a flexible riser performs the following functions:

- The external sheath prevents ingress of seawater to the annulus.
- The tensile armors provide tensile strength to support axial loads and comprise of two or four layers with opposite laying angles to equilibrate the torsional torque.
- The anti-wear layer prevents friction between armor layers.

- The pressure vault provides internal pressure resistance.
- The pressure sheath contains fluid within the pipe bore and provides the fluid transportation capacity.
- The carcass provides collapse resistance to external pressure.

Designs are made by using a mixture of local and international standards to ensure sufficient margin of safety (API 17B 2014 & 17J 2014).

Flexible risers can be installed in a number of different configurations according to the production requirement and site-specific environmental conditions (Bai and Bai 2005). Static analysis shall be carried out to determine the configurations which are based on:

- Global behavior and geometry
- Structural integrity, rigidity, and continuity
- Cross sectional properties
- Means of support
- Material
- Cost

The six main configurations for flexible risers are shown in Fig. 3.

wave models. In each of the wave simulators, wave-induced water particle velocity and acceleration, surface elevation, and dynamical pressure and pressure gradient of an arbitrary point in space and time are defined mathematically. This will allow the wave simulators to compute the wave kinematics during a time domain dynamic analysis. The 2D regular long-crested waves are useful when studying the effects of extreme waves, while 2D random long-crested waves are used when modeling a complete sea state (Bai and Bai 2005).

Figure 4 shows the parameters that are used when defining a 2D regular long-crested wave propagating in the positive x-direction, where

L : wavelength

H : wave height

d : $Z_s - Z_b$ = still water depth

a : wave amplitude ($H/2$)

T : wave period

g : acceleration due to gravity

t : time

α : phase angle (radians)

x : direction of wave propagation

Wave frequency

$$\omega = \frac{2\pi}{T} \quad (1)$$

Wave number

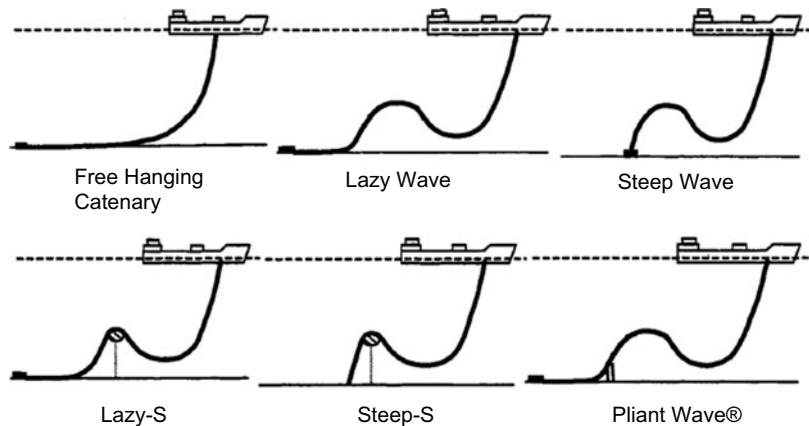
$$K = \frac{2\pi}{L} \quad (2)$$

Hydrodynamic Loads

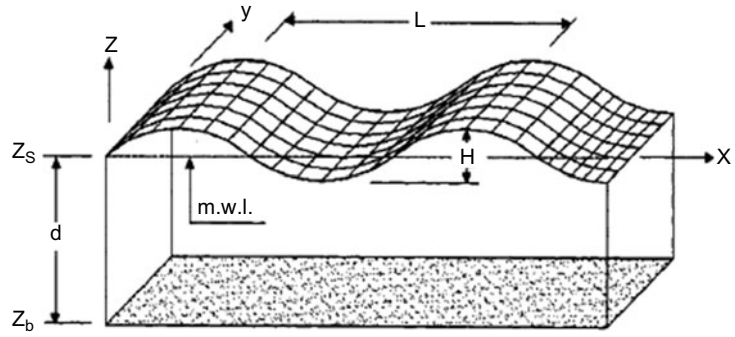
Wave Simulator

Wave simulators can be established using 2D regular long-crested and 2D random long-crested

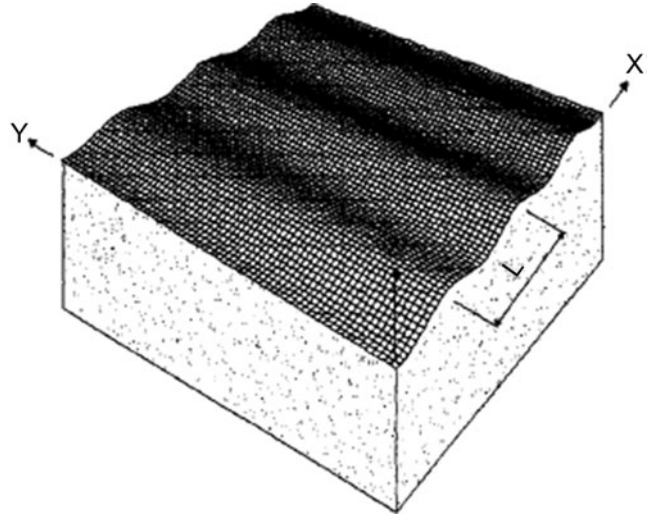
Dynamic Behavior and Fatigue, Fig. 3 Flexible riser configurations (Bai and Bai 2005)



Dynamic Behavior and Fatigue, Fig. 4 Defining parameters for 2D regular waves (Bai and Bai 2005)



Dynamic Behavior and Fatigue, Fig. 5 2D regular long-crested waves (Bai and Bai 2005)



The dispersion relation expresses the relationship between the wave period and wavelength and is given by

$$\frac{\omega^2}{gk} = \tanh(kd) \quad (3)$$

2D Regular Long-Crested Waves

Figure 5 shows the 2D regular long-crested waves which are defined by their wave amplitude and frequency. The following expressions present the wave kinematics:

$$\eta = a \cdot \sin(\omega t - kx + \alpha) \quad (4)$$

Fluid velocity component in the x-direction

$$V_x = \frac{agk}{\omega} \cdot \frac{\cosh(k(d+z))}{\cosh(kd)} \cdot \sin(\omega t - kx + \alpha) \quad (5)$$

Fluid velocity component in the z-direction

$$V_z = \frac{agk}{\omega} \cdot \frac{\sinh(k(d+z))}{\cosh(kd)} \cdot \cos(\omega t - kx + \alpha) \quad (6)$$

Fluid acceleration component in the x-direction

$$a_x = agk \cdot \frac{\cosh(k(d+z))}{\cosh(kd)} \cdot \cos(\omega t - kx + \alpha) \quad (7)$$

Fluid acceleration component in the z-direction

$$a_z = -agk \cdot \frac{\sinh(k(d+z))}{\cosh(kd)} \cdot \sin(\omega t - kx + \alpha) \quad (8)$$

Dynamic pressure

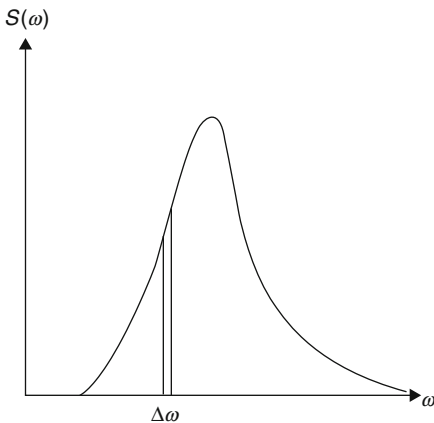
$$P_{dyn} = \rho ag \cdot \frac{\cosh(k(d+z))}{\cosh(kd)} \cdot \sin(\omega t - kx + \alpha) \quad (9)$$

2D Random Long-Crested Waves

The 2D random long-crested wave shown in Fig. 8 is formulated based on the use of a wave spectrum of Fig. 6. Input of significant wave height, peak frequency, etc. (input is dependent on type of wave spectrum) defines the characteristics of the sea state (Bai and Bai 2005). The JONSWAP spectrum can be defined as

$$S(\omega) = a_p g^2 (10)^{-5} \exp\left(-\frac{5}{4} \left(\frac{\omega}{\omega_p}\right)^{-4}\right) \gamma^{a_p} \quad (10)$$

$$a_p = \exp\left(\frac{-(\omega - \omega_p)^2}{2\sigma^2 \omega_p^2}\right) \quad (11)$$



Dynamic Behavior and Fatigue, Fig. 6 Wave spectrum (Bai and Bai 2005)

where

ω : angular frequency

ω_p : angular frequency of spectral peak

g : acceleration due to gravity

a_p : Phillip's constant

σ : spectral width parameter

γ : the JONSWAP peakedness parameter determined from

$$\gamma = \begin{cases} 1, & \gamma < 1 \\ \exp\left(5.75 - 0.367T_{peak}\sqrt{\frac{g}{H_s}}\right), & 1 < \gamma < 5 \\ 5, & \gamma > 5 \end{cases} \quad (12)$$

From the wave spectrum, we can obtain several properties. λ_n denotes the n^{th} (stress) spectral moment defined by

$$\lambda_n = \int_0^\infty \omega^n S_{ss}(\omega) d\omega \quad (13)$$

$H_{1/3}$ is the significant wave height and can be found as

$$H_{1/3} = 4\sqrt{\lambda_0} \quad (14)$$

The characteristics wave period T_w may be estimated as

$$T_w \approx 2\pi\sqrt{\frac{\lambda_0}{\lambda_2}} \quad (15)$$

and the spectral bandwidth parameter ε as

$$\varepsilon = \sqrt{1 - \frac{\lambda_2^2}{\lambda_0 \lambda_4}} \quad (16)$$

By performing an inverse transportation, the wave amplitude (a_i) and frequencies (ω_i) of each wave component are extracted from the wave spectrum. Extraction of amplitudes and frequencies from the wave spectrum is for each wave component done according to

$$a_i = \sqrt{2 \cdot \Delta\omega \cdot S(\omega_i)} \quad (17)$$

where

$$\omega_i = i \cdot \Delta\omega$$

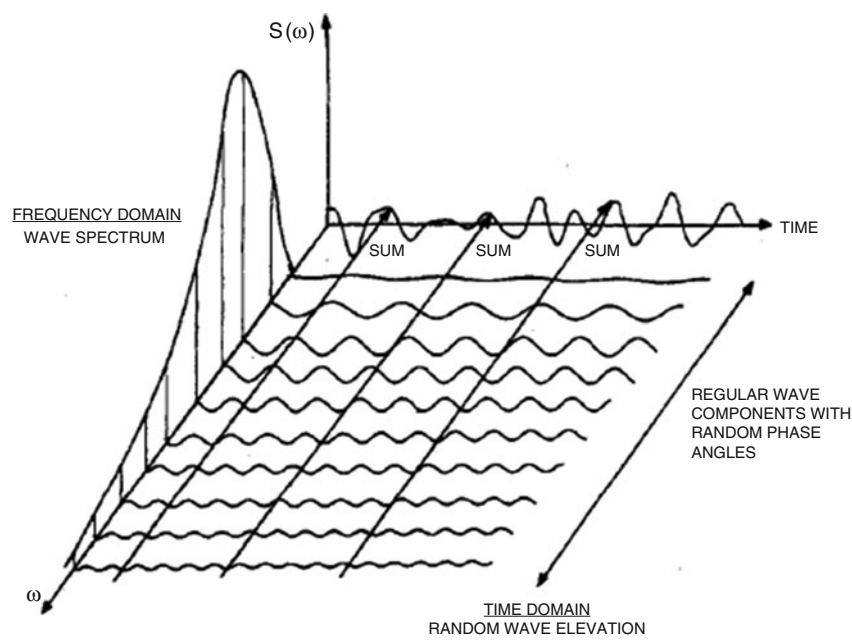
(18)

$\Delta\omega$: the constant difference between successive frequencies.

$$K_i = \frac{(\omega_i)^2}{g}$$

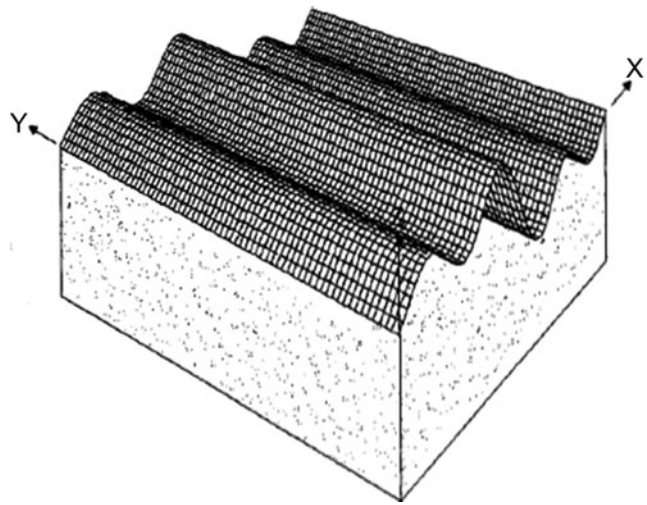
(19)

Furthermore, a random phase angle α_i , uniformly distributed between 0 and 2π , is assigned to each wave component. The wave kinematics is thus represented as a sum of linear components (Fig. 7). If “N” is the number of wave components, the sea state at a particular time and location can be represented by (Fig. 8):



Dynamic Behavior and Fatigue, Fig. 7 Connection between a frequency domain and a time domain representation of long-crested waves (Bai and Bai 2005)

Dynamic Behavior and Fatigue, Fig. 8 2D random long-crested waves (Bai and Bai 2005)



Surface elevation

$$\eta = \sum_{i=1}^N a_i \cdot \sin(\omega_i t - k_i x + \alpha_i) \quad (20)$$

Velocity component in the x-direction

$$v_x = g \cdot \sum_{i=1}^N \frac{a_i k_i}{\omega_i} \cdot \frac{\cosh(k_i(d+z))}{\cosh(k_i d)} \cdot \sin(-\omega_i t - k_i x + \alpha_i) \quad (21)$$

Velocity component in the z-direction

$$v_z = g \cdot \sum_{i=1}^N \frac{a_i k_i}{\omega_i} \cdot \frac{\sinh(k_i(d+z))}{\cosh(k_i d)} \cdot \cos(-\omega_i t - k_i x + \alpha_i) \quad (22)$$

Acceleration component in the x-direction

$$a_x = g \cdot \sum_{i=1}^N a_i k_i \cdot \frac{\cosh(k_i(d+z))}{\cosh(k_i d)} \cdot \cos(\omega_i t - k_i x + \alpha_i) \quad (23)$$

Acceleration component in the z-direction

$$a_z = -g \cdot \sum_{i=1}^N a_i k_i \cdot \frac{\sinh(k_i(d+z))}{\cosh(k_i d)} \cdot \sin(\omega_i t - k_i x + \alpha_i) \quad (24)$$

Dynamical pressure

$$P_{dyn} = \rho g \cdot \sum_{i=1}^N a_i \cdot \frac{\cosh(k_i(d+z))}{\cosh(k_i d)} \cdot \sin(\omega_i t - k_i x + \alpha_i) \quad (25)$$

Hydrodynamic Drag and Inertia Forces

Consider a section of pipeline exposed to a flow, it will experience hydrodynamic forces due to the combined effect of increased flow velocity above the pipe and flow separation from the pipe surface (Bai and Bai 2005). Figure 9 shows the velocity distribution around the pipe. The force vectors are:

Pipeline Exposed to Steady Fluid Flow

Fluid drag is associated with flow velocities due to steady currents superposed by any wave that may be present (Fig. 10). The following expression gives the transverse drag force component per unit length of the pipeline.

$$\text{Transverse drag force, } F_D = \frac{1}{2} \rho C_D D v_n |v_n| \quad (26)$$

where

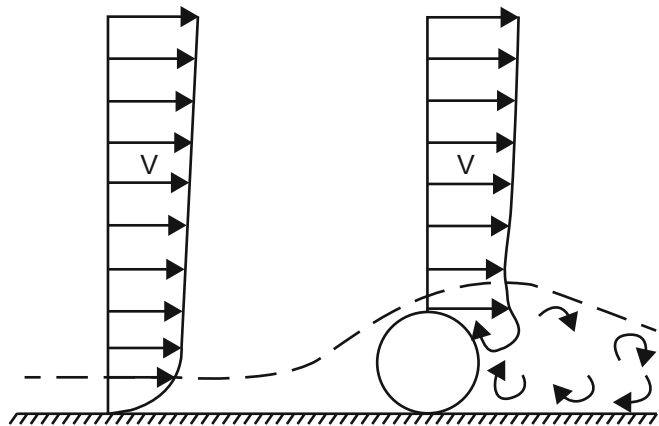
C_D : transverse drag coefficient

v_n : transverse water particle velocity

ρ : density of seawater

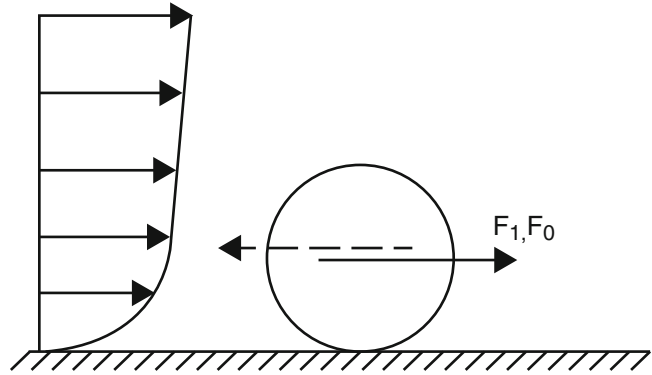
D : total external diameter of pipe

Dynamic Behavior and Fatigue, Fig. 9 Flow field around pipe (Bai and Bai 2005)



Dynamic Behavior and Fatigue, Fig. 10

Fluid drag and inertia forces acting on a pipe section (Bai and Bai 2005)



Pipeline Exposed to Accelerated Fluid Flow

A pipeline exposed to an accelerated fluid experiences a force proportional to the acceleration; this force is called the inertia force (Bai and Bai 2005). The following expression gives the transverse inertia force component per unit length of a pipeline:

$$\text{Transverse inertia force, } F_I = \frac{\pi}{4} \rho D^2 C_M a_n \quad (27)$$

where

C_M : $(C_a + 1)$, transverse inertia coefficient
 a_n : transverse water particle acceleration

Morison Equation

Morison et al. (1950) superimposed the linear inertia force (from potential theory and oscillating flows) and the adapted quadratic drag force (from real flows and constant currents) to get the following resultant force per unit length (Journée 2000):

Wave forces

$$F(t) = F_{inertia}(t) + F_{drag}(t) \quad (28)$$

or:

$$F(t) = \frac{\pi}{4} \rho C_M D^2 \cdot \dot{u}(t) + \frac{1}{2} \rho C_D D \cdot u(t) |u(t)| \quad (29)$$

where the first term is the inertia force and the second term the drag force. It is worthy to note that in Eq. 29, the drag and inertia force components are 90° out of phase with each other when seen as functions of time. This reflects a direct

consequence of the phase shift between velocity and acceleration in an oscillatory motion.

The formula presented in Eqs. 26 and 27 do not take into account that the pipe itself may have a velocity and acceleration. The inline force per unit length of a pipe is determined using the complete Morison's equation (Bai and Bai 2005).

Inline force per unit length of a pipe

$$\begin{aligned} F_{IL}(t) = & \frac{1}{2} \rho C_D \left(U - \frac{\partial y}{\partial t} \right) \left| U - \frac{\partial y}{\partial t} \right| \\ & + C_M \frac{\pi}{4} \rho D^2 \frac{\partial U}{\partial t} - (C_M - 1) \\ & \times \frac{\pi}{4} \rho D^2 \frac{\partial^2 y}{\partial t^2} \end{aligned} \quad (30)$$

where

U : instantaneous (time dependant) flow velocity

y : inline displacement of the pipe

$\delta/\delta t$: differentiation with respect to time

Drag and inertia coefficient parameter dependency

In general, the values of the dimensionless drag (C_D) and inertia (C_M) coefficients can be determined experimentally; however, their dependency is given by

$$C_D = C_D(R_e, KC, \alpha, (e/D), (k/D), (A_Z/D)) \quad (31)$$

$$C_M = C_M(R_e, KC, \alpha, (e/D), (A_Z/D)) \quad (32)$$

The following dimensionless quantities are presented:

Reynolds number

Reynolds number indicates the present flow regime (i.e., laminar or turbulent) and is given as

$$R_e = \frac{UL}{\nu} \quad (33)$$

where

R_e : Reynolds number (—)

U : flow velocity (m/s)

L : characteristic length/diameter (D) for pipelines (m)

ν : kinematic viscosity (m^2/s)

Froude number

The Froude number is associated primarily with free surface effects, while wave forces can be exerted on cylinder elements which are so far below the sea surface that no surface disturbance is generated (Journée 2000).

$$F = \frac{v}{\sqrt{g \cdot D}} \quad (34)$$

where

F : Froude number (—)

g : acceleration of gravity (m/s^2)

D : diameter for pipelines (m)

ν : kinematic viscosity (m^2/s)

Keulegan-Carpenter number (KC)

Keulegan and Carpenter (1958) determined C_D and C_M values for various cylinders in an oscillating flow. They discovered that their data could be plotted reasonably as a function of the dimensionless Keulegan-Carpenter number (Journée 2000). However, Keulegan-Carpenter number gives information on how the flow separation around cylinders will be for ambient oscillatory planar flow ($U = U_M \sin((2\pi/T)t + \varepsilon)$) (Bai and Bai 2005) and is given as

$$KC = \frac{U_M T}{D} \quad (35)$$

where

KC : Keulegan – Carpenter number (—)

U_M : flow velocity amplitude (m/s)

T : oscillating flow period (s)

D : pipe diameter (m)

ε : phase angle (radians)

t : time (s)

$$KC = \pi \cdot \frac{H}{D} = 2\pi \frac{\zeta_a}{D} \text{ (deepwater only)} \quad (36)$$

The current flow ratio may be related to classify the flow regimes:

$$\alpha = \frac{U_c}{U_c + U_w} \quad (37)$$

where

U_c : typical current velocity normal to the pipe

U_w : significant wave velocity normal to pipe given for each sea state ($H_S T_P \theta_W$)

Note that $\alpha = 0$ corresponds to pure oscillatory flow due to waves, while $\alpha = 1$ corresponds to pure (steady) current flow.

Sarpkaya beta

Sarpkaya and Isaacson (1981) conducted several experiments using a U-tube to generate an oscillating flow. They found that the ratio

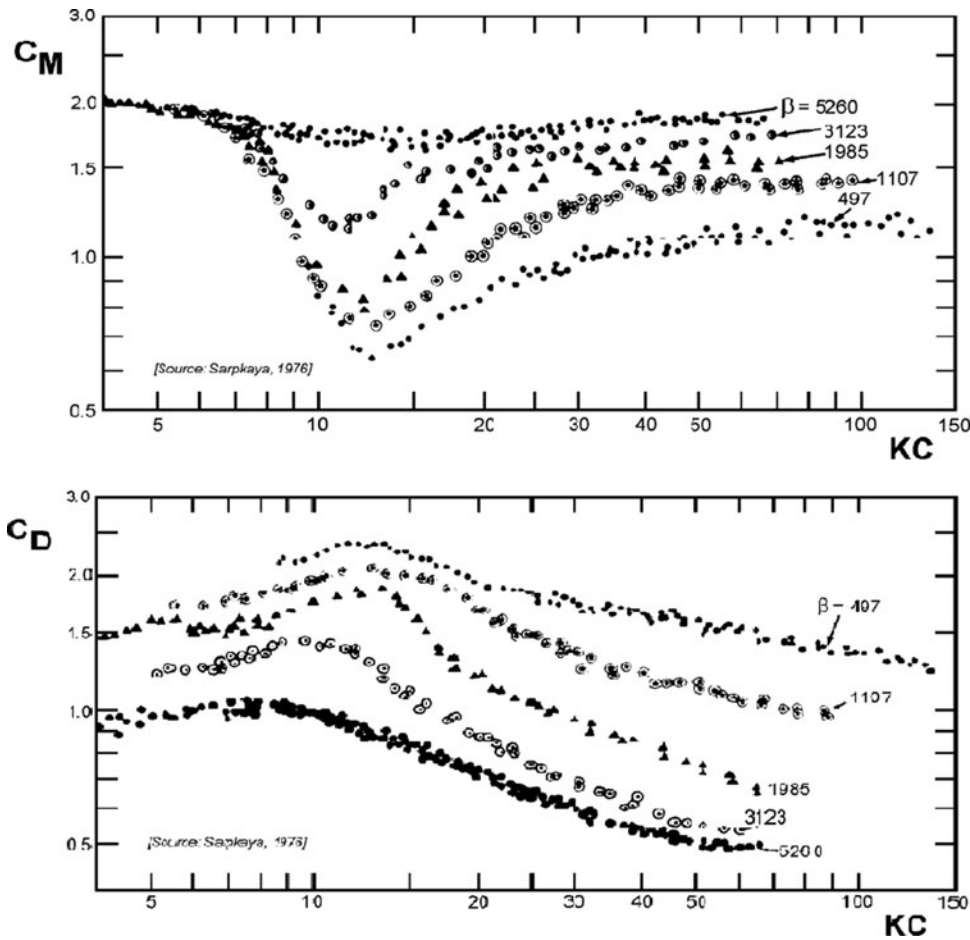
$$\beta = \frac{D^2}{V \cdot T} \quad (38)$$

was convenient for plotting their data which further procession like that of Iversen's modulus gives (Journée 2000):

$$\beta = \frac{R_e}{KC} \quad (39)$$

Dimensionless roughness

Cylinder roughness is generally made dimensionless by dividing it with diameter (Journée 2000):



Dynamic Behavior and Fatigue, Fig. 11 Typical laboratory measurement results from Sarpkaya (Journée 2000)

$$\frac{\varepsilon}{D} \text{ roughness height } / \text{ cylinder diameter} \tag{40}$$

The Keulegan-Carpenter number has stood as the most realistic and useful primary independent parameter for plotting C_D and C_M . This is sometimes augmented by using Re , β , or $\frac{\varepsilon}{D}$ to label-specific curves, thus introducing additional independent information (Journée 2000).

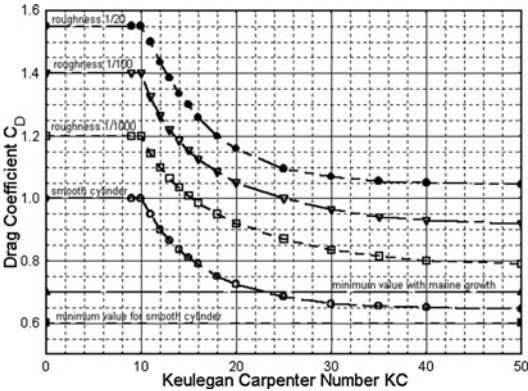
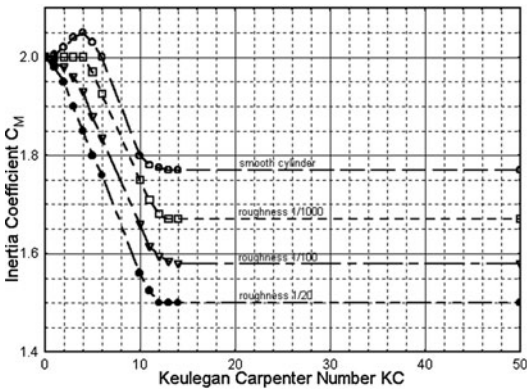
The results of Sarpkaya’s experiments with smooth cylinders in U-tubes are presented as graphs of the coefficients C_D and C_M as functions of β and KC (Journée 2000). Note that in Fig. 11, the horizontal (KC) axis is

Dynamic Behavior and Fatigue, Table 1 Suggested Morison coefficients (Claus 1992)

KC	$Re < 10^5$		$Re > 10^5$	
	C_D	C_M	C_D	C_M
<10	1.2	2.0	0.6	2.0
≥10	1.2	1.5	0.6	1.5

logarithmic. Respective curves on each graph are labeled with appropriate values of β . Claus (1992), for example, suggested the drag and inertia coefficient values as presented in the following Table 1.

Various design codes or rules also specify (or suggest) appropriate values for C_D and C_M .

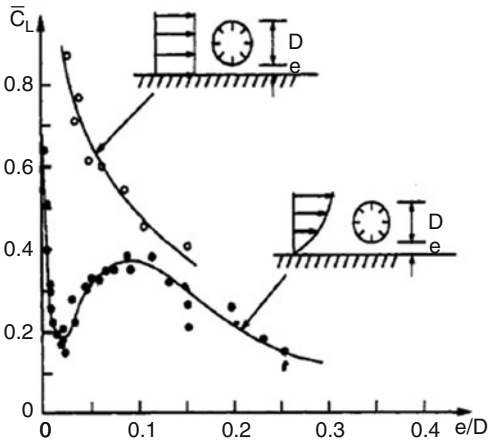


Dynamic Behavior and Fatigue, Fig. 12 Suggested drag and inertia coefficient values from DNV (Journée 2000)

Dynamic Behavior and Fatigue, Table 2 Morison coefficients by API and SNAME (Journée 2000)

	Smooth		Rough	
	C_D	C_M	C_D	C_M
API	0.65	1.6	1.05	1.2
SNAME	0.65	2.0	1.0	1.8

Those published by Det Norske Veritas (DNV) 1989 or by the American Petroleum Institute (API) are the most widely accepted; the DNV suggestions for design purposes are shown in Fig. 12. The API and SNAME have the simplest approach as shown in Table 2 (Journée 2000).



Dynamic Behavior and Fatigue, Fig. 13 C_L in shear and shear-free flow for $10^3 < Re < 30 \times 10^4$ (Bai and Bai 2005)

Hydrodynamic Lift Forces

Lift Force Using Constant Lift Coefficients

The lift force per unit length of a pipeline can be computed according to (Bai and Bai 2005)

$$\text{Vertical lift force, } F_L = \frac{1}{2} \rho D C_L v_n^2 \quad (41)$$

where

C_L : lift coefficient for pipe on a surface

v_n : transverse water particle velocity (perpendicular to the direction of the lift force)

ρ : density of seawater

D : total external diameter of pipe

Lift Force Using Variable Lift Coefficients

As can be imagined, the hydrodynamic lift coefficient (C_L) will vary as a function of the gap that might exist between the pipeline and the seabed. It can be seen from Fig. 13 that a significant drop in the lift coefficient is present even for very small ratios of e/D . This is true both for the shear and the shear-free flow (Bai and Bai 2005). The lift coefficients according to Fredsøe and Sumer (1997) are given in Fig. 13.

Dynamic Analysis of Offshore Structures

Basic Formulation of Equations of Motion

Before the motion of an offshore structure can be calculated, an analytical representation (mathematical model) is needed for the structure, together with the loadings and restraints (Wilson and Orgill 1984). Based on the free body sketch, the equations of motion are formulated in a straightforward manner, using either Newton's second law or Lagrange's energy methods.

Single Degree of Freedom Analysis

Consider the surge motion for the spread moored ship shown in Fig. 14. Let the motion of m be restricted to its horizontal displacement coordinate $v = v(t)$ of its mass center at G (Fig. 15). Sway, pitch, heave, and yaw motions are neglected. Figure 15 shows the FBD in the vertical plane which depicts the four types of externally applied loads: self-weight W , which is balanced by its buoyant force equal in magnitude to W ; the equivalent environmental forces $p_1(t)$; the equivalent of mooring line restraint force $q(v)$; and the damping force $f(\dot{v})$.

Let $\sum F_v$ denotes the sum of the four types of external force components on m in line with v and

positive in the positive v direction. In these terms, Newton's second law states:

$$\sum F_v = m\ddot{v} \quad (42)$$

where $\ddot{v}$ is the absolute acceleration of G . This equation is particularly useful in single degree of freedom models involving rigid body translation, in which the flexible supports restraining the motion of m are in line with v . When Eq. 42 is applied in the v direction to the FBD, the resulting equation for surge motion becomes (Wilson and Orgill 1984)

$$m\ddot{v} + f(\dot{v}) + q(v) = p_1(t) \quad (43)$$

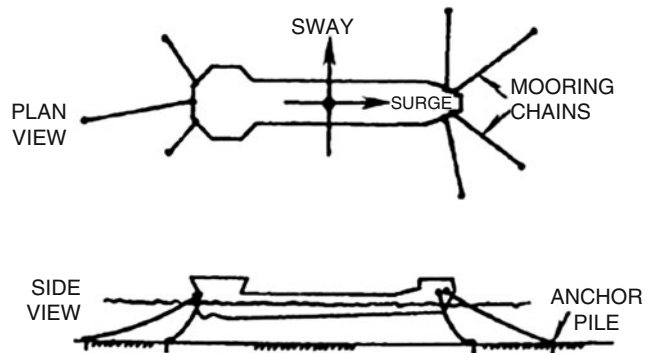
Multi-Degree of Freedom Analysis

The general set of equations representing structural motion is defined by the following matrix form:

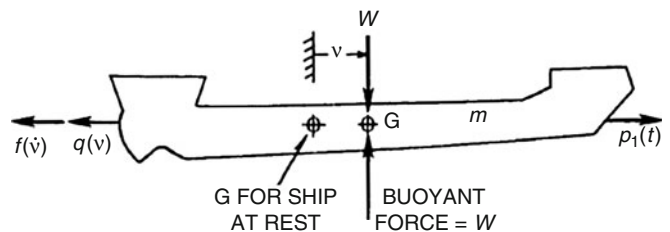
$$M\ddot{\xi} + C\dot{\xi} + K\xi = p \quad (44)$$

Equation 44 represents a finite set of N ordinary, linear differential equations in N -independent coordinates ($\xi_1, \xi_2, \dots, \xi_b, \dots, \xi_N$), in which the

Dynamic Behavior and Fatigue, Fig. 14 A spread mooring configuration (Wilson and Orgill 1984)



Dynamic Behavior and Fatigue, Fig. 15 Free body sketch for the spread moored ship of Fig. 14 (Wilson and Orgill 1984)



coefficient matrices M , K , and C are constant. When $N = 1$, Eq. 44 reduces to a single degree of freedom structural model. The respective terms of ξ, p, M, K , and C are the coordinate vector, the loading vector, and the mass, stiffness, and damping matrices.

Free, Undamped Motion

The governing equation for free and undamped motion of a structural system, i.e., when $C = p = 0$, is given as

$$M\ddot{\xi} + K\xi = 0 \quad (45)$$

Dynamic Analysis of Beams (Pipes) and Cables

Transverse Vibration of Beams and Cables

The transverse dynamic displacement of the beam (applicable for pipes) or cable model shown in Fig. 16 is deduced from Newton's second law as

$$\frac{\partial^2}{\partial x^2} \left(EI \frac{\partial^2 v}{\partial x^2} \right) - \frac{\partial}{\partial x} \left(P \frac{\partial v}{\partial x} \right) + \bar{c} \frac{\partial v}{\partial t} + \bar{m} \frac{\partial^2 v}{\partial t^2} = \bar{q}(x, t) \quad (46)$$

This is a general form of the linear Bernoulli-Euler dynamic beam-cable equation where EI , P , and $\bar{m}$ are arbitrary functions of x and the

excitation load $\bar{q}(x, t)$ is arbitrary in both x and t . For submerged beams and cables, a more realistic form of damping than the linear approximation $\bar{c}(\partial v / \partial t)$ in Eq. 46 is velocity squared damping, defined by

$$c' \frac{\partial v}{\partial t} \cdot \left| \frac{\partial v}{\partial t} \right| \quad (47)$$

Here c' is an experimental constant which generally depends on the frequency of oscillation of the structural components. Initial and boundary conditions must be specified for solutions to Eq. 46 to be unique. For a cable ($EI = 0$) and for a beam ($EI > 0$).

The following initial conditions are always required:

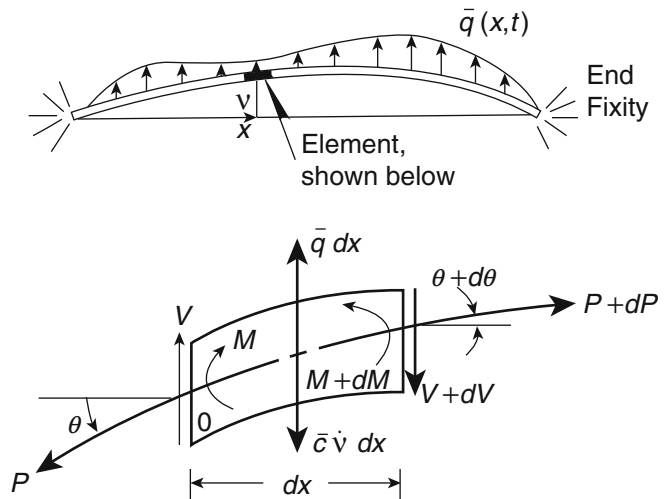
$$v(x, 0) \text{ and } \frac{\partial v}{\partial t}(x, 0), \text{ for } 0 \leq x \leq \ell \quad (48)$$

In addition, solutions to cable problems require that the end displacements $v(0, t)$ and $v(\ell, t)$ be specified, and solutions to beam problems require four boundary conditions, two at $x = 0$ and two at $x = \ell$.

Two special cases of Eq. 46 that are of practical importance in offshore structural systems are modeled as follows:

1. For an undamped, flexible cable ($EI = 0$) subjected to a tension load P which is independent of x , Eq. 46 becomes

Dynamic Behavior and Fatigue, Fig. 16 Dynamic model of a line element (Wilson and Orgill 1984)



$$-P \frac{\partial^2 v}{\partial x^2} + \bar{m} \frac{\partial^2 v}{\partial t^2} = \bar{q}(x, t) \quad (49)$$

Here m can vary with the longitudinal dimension, but it is constant for most applications.

2. For an undamped beam of constant stiffness EI , a constant $\bar{m}$, and a tension P which is independent of x , Eq. 46 becomes

$$EI \frac{\partial^4 v}{\partial x^4} - P \frac{\partial^2 v}{\partial x^2} + \bar{m} \frac{\partial^2 v}{\partial t^2} = \bar{q}(x, t) \quad (50)$$

Frequencies and Mode Shapes of Beams and Cables

Frequency parameter is given by

$$\omega = Y \sqrt{\frac{P}{\bar{m}}} \quad (51)$$

where

$$Y = \frac{n\pi}{\ell}, n = 1, 2, \dots \quad (52)$$

$$\omega_n = \frac{n\pi}{\ell} \sqrt{\frac{P}{\bar{m}}}, n = 1, 2, \dots \quad (53)$$

For each characteristic value Y , there is a frequency ω_n and a corresponding mode shape $X = X_n$.

$$X_n = C_n \sin \frac{n\pi x}{\ell}, n = 1, 2, \dots \quad (54)$$

in which the coefficients C_n are arbitrary. For fixed ends of the beam or cable

$$\begin{aligned} v(0, t) = 0 \text{ or } (0) = 0; v(\ell, t) = 0 \text{ or } X(\ell) \\ = 0 \end{aligned} \quad (55)$$

The first two mode shapes are shown in Fig. 17. It is observed that the fundamental frequency ω_n corresponds to the half sine wave and the next highest frequency ω_n corresponds to a full sine wave. This mathematical model predicts that ever-increasing frequencies are possible as n becomes larger and larger (Wilson and Orgill 1984).

Slug-Induced Vibration of Deepwater Riser Structure

Onuoha et al. (2018a) (A & B) studied the stress impact of severe slug on riser base and has successfully modeled the dynamic response of deepwater riser (Fig. 18) induced by severe slug buildup with the following equation system:

$$\begin{aligned} EI \frac{\partial^4 w}{\partial y^4} - T_{eff}(y) \frac{\partial^2 w}{\partial y^2} + \rho_f A_i u_{LS}^2 \frac{\partial^2 w}{\partial y^2} \\ + 2\rho_f u_{LS} A_i \frac{\partial^2 w}{\partial y \partial t} - (A_e P_e(y) - A_i P_i(Z, t)) \frac{\partial^2 w}{\partial y^2} \\ + (\rho_r A_r + \rho_f A_i) \frac{\partial^2 w}{\partial t^2} = 0 \end{aligned} \quad (56)$$

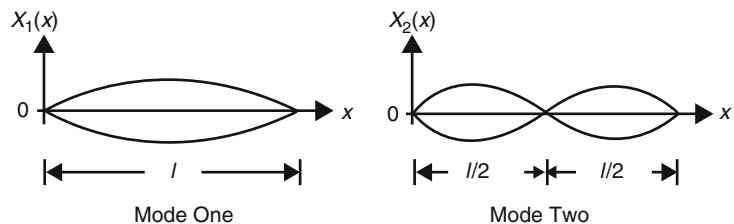
$$T_{eff}(y) = W_a(-y + fL) \quad (57)$$

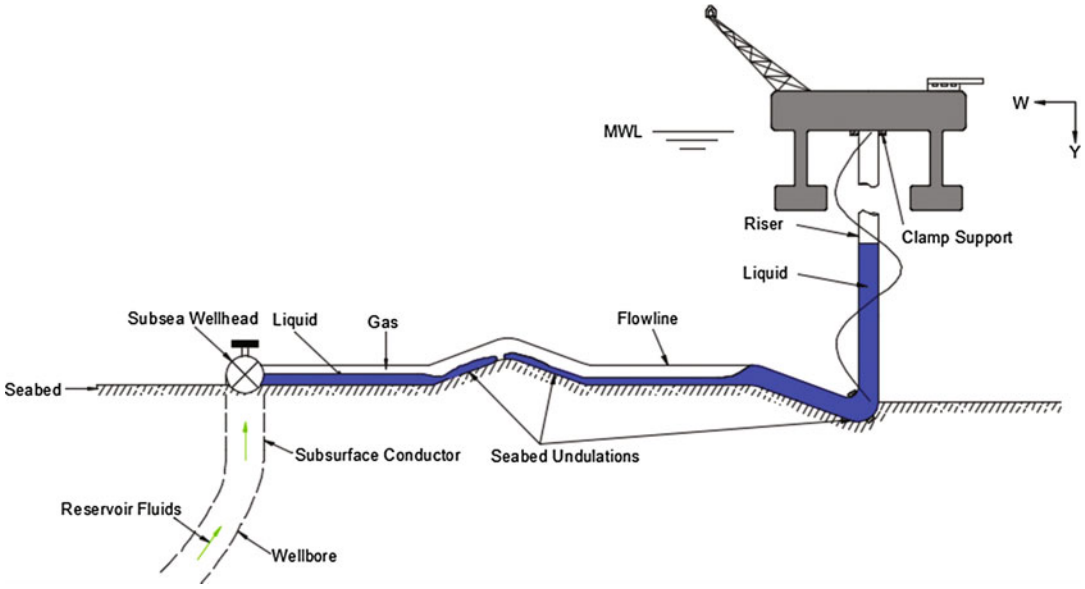
$$Z(t) = F_b(L, u_{GS}) - \alpha(x - x_i) + u_{LS}t \quad (58)$$

$$z_* = \alpha(x - x_i) \quad (59)$$

$$\begin{aligned} F_b(L, u_{SG}) = \left[-7.71 + 5.8 \left(u_{SG} + \frac{1}{u_{SG}} \right) \right] \\ \times \frac{L(y)}{50} \end{aligned} \quad (60)$$

Dynamic Behavior and Fatigue, Fig. 17 The first two mode shapes for both a fixed end cable and simply supported beam (Wilson and Orgill 1984)





Dynamic Behavior and Fatigue, Fig. 18 A flowline-riser subsea production system in undulated seabed terrain

For $z_* \cong 0$, i.e., when $\alpha(x - x_i)$ is very small

(61)

$$Z(t) = F_b(L, u_{SG}) + u_{LS}t \quad (62)$$

$$P_i(Z, t) = \rho_L g Z(t) \quad (63)$$

Boundary conditions

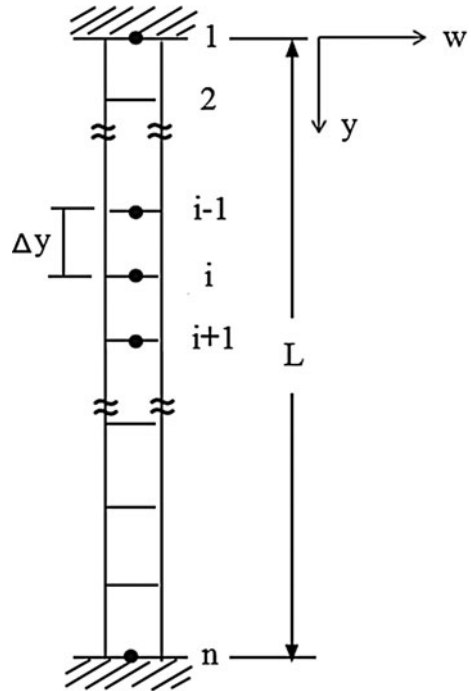
$$w(y, t)|_{y=0} = 0; \frac{\partial w(y, t)}{\partial y} \bigg|_{y=0} = 0 \quad (64)$$

$$w(y, t)|_{y=L} = 0; \frac{\partial w(y, t)}{\partial y} \bigg|_{y=L} = 0 \quad (65)$$

Initial conditions

$$w(y, t) = A \sin \left(\frac{\pi y}{L} \right) \bigg|_{t=0} \quad (66)$$

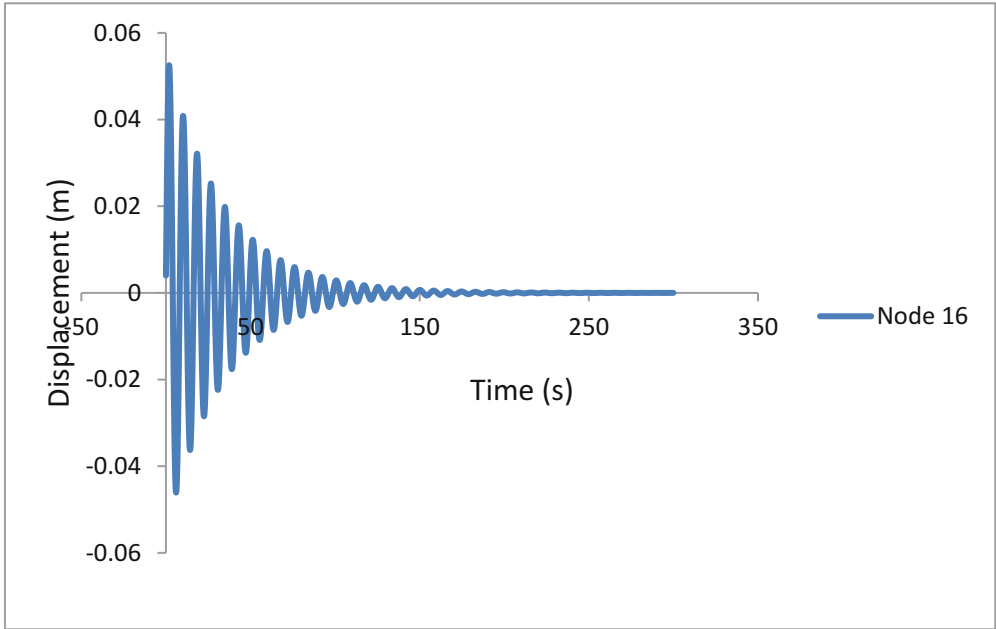
$$\frac{\partial w(y, t)}{\partial t} \bigg|_{t=0} = 0 \quad (67)$$



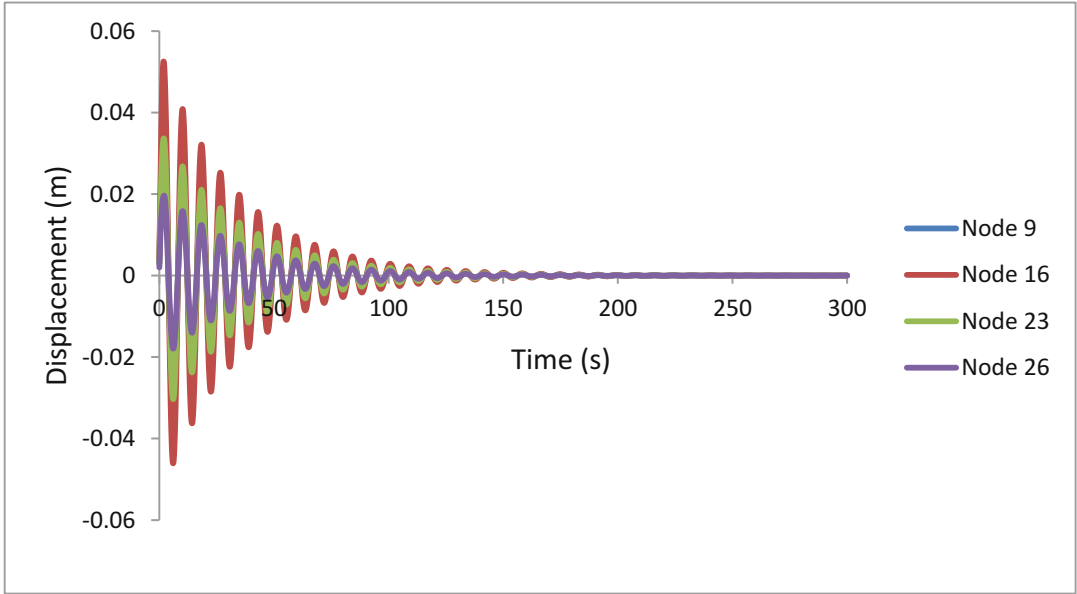
Dynamic Behavior and Fatigue, Fig. 19 Space discretization scheme for the top tensioned riser model

where $(w/OD < 0.05/\pi)$ is the small displacement of the riser, i.e., the amplitude, A (Kuiper et al. 2008). Numerical solutions of the discretized elements of

Fig. 19 give the transverse vibrations of the riser structure as shown in Figs. 20, 21, and 22.



Dynamic Behavior and Fatigue, Fig. 20 Transverse vibration of the mid-node (76.2 m) of the riser for $u_{LS} = 0.095 \text{ m/s}$

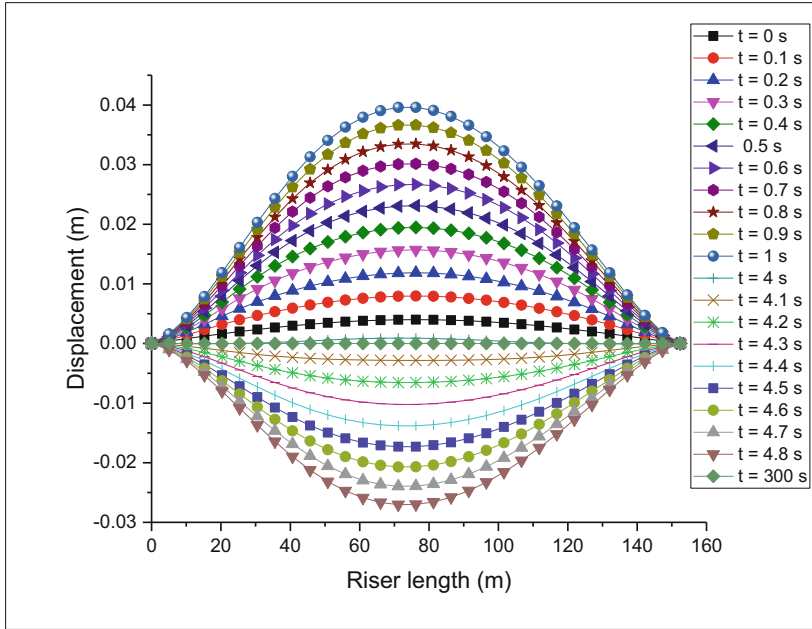


Dynamic Behavior and Fatigue, Fig. 21 Transverse vibration of different nodes of the riser for $u_{LS} = 0.095 \text{ m/s}$

Vortex-Induced Vibrations (VIV) of Pipelines

The interaction of the vortices with a circular cylinder is the source of the effect that causes vortex-induced vibration. Vortex shedding from

a smooth, circular cylinder in a steady subsonic flow is a function of Reynolds number. The major Reynolds number regimes of vortex shedding from a smooth circular cylinder summarized by Lienhard (1966) are shown in Fig. 23.



Dynamic Behavior and Fatigue, Fig. 22 Transverse displacement of the riser length at different time for the velocity $u_{LS} = 0.095 \text{ m/s}$

Reynolds Number

Vortex shedding from a smooth, circular cylinder in a steady subsonic flow is a function of Reynolds number. The Reynolds number is based on free stream velocity U and cylinder diameter D

$$R_e = \frac{UD}{\nu} \quad (68)$$

Strouhal Number

Strouhal number (S) is the dimensionless proportionality constant between the predominant frequency of vortex shedding and the free stream velocity divided by the cylinder width.

$$f_s = \frac{SU}{D} \quad (69)$$

As the flow velocity is increased or decreased such that the vortex shedding frequency (f_s) approaches the natural frequency (f_n) of an elastic structure, so that

$$f_n = f_s = \frac{SU}{D}, \text{ or } \frac{U}{f_n D} = \frac{U}{f_s D} = \frac{1}{S} = 5 \quad (70)$$

where f_s is the vortex shedding frequency in units of hertz, U is the free stream flow velocity approaching the cylinder, and D is the cylinder diameter. U and D must have consistent units (Blevins 1994). The Strouhal number of a stationary circular cylinder in a subsonic flow is a function of Reynolds number and to a lesser extent surface roughness and free stream turbulence as shown in Fig. 24.

Harmonic Model for VIV

Because vortex shedding is a sinusoidal process, it is reasonable to model the vortex shedding transverse force imposed on a circular cylinder as harmonic in time at the shedding frequency:

$$F_L = \frac{1}{2} \rho U^2 D C_L \sin(\omega_s t) \quad (71)$$

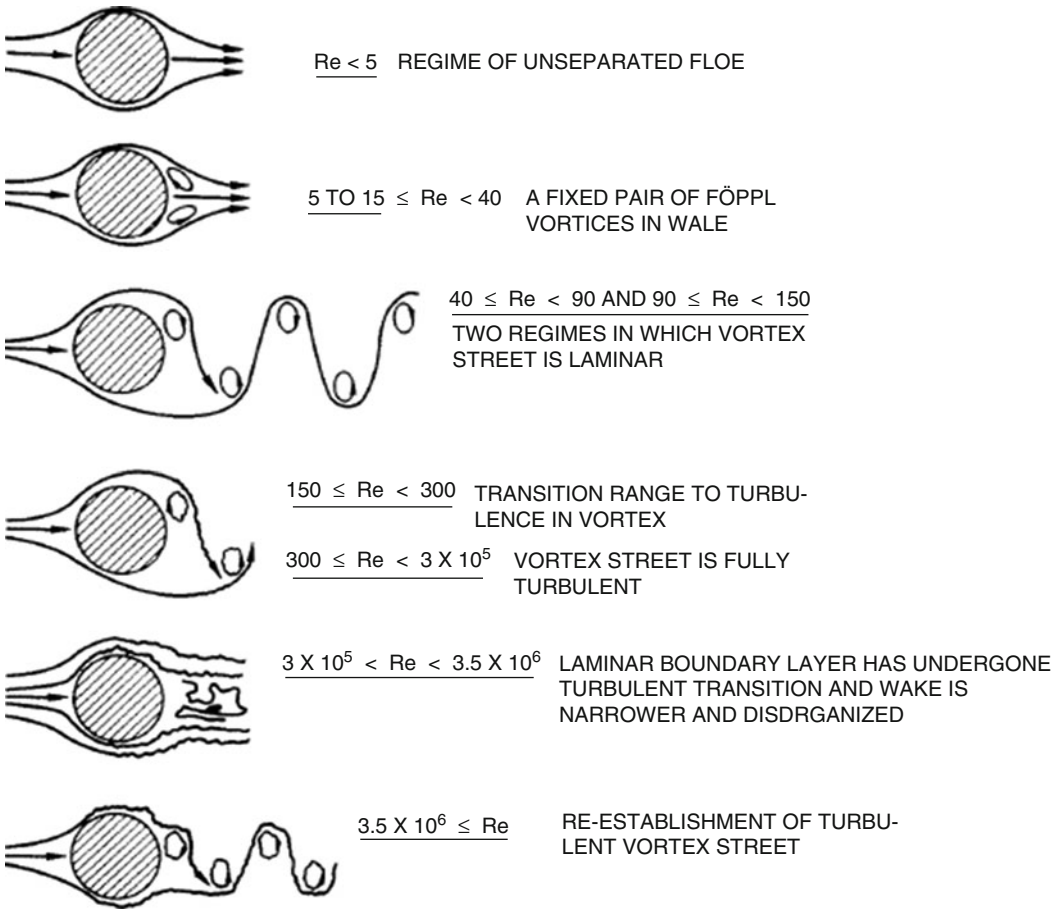
where

ρ : fluid density (must be in mass units)

U : free stream velocity

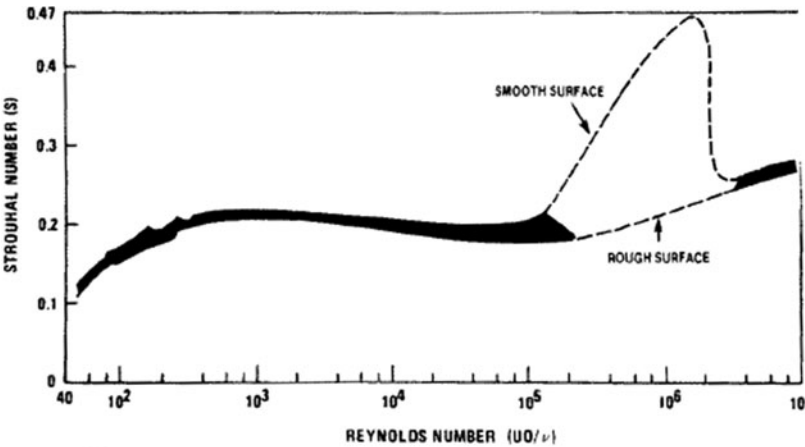
D : cylinder diameter

C_L : lift coefficient, dimensionless

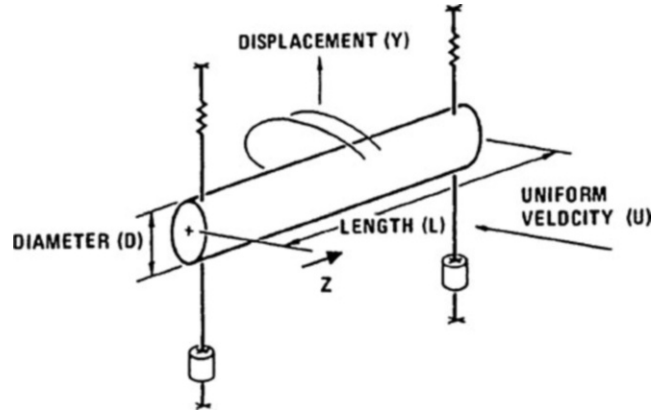


Dynamic Behavior and Fatigue, Fig. 23 Regimes of fluid flow across smooth circular cylinders (Lienhard 1966)

Dynamic Behavior and Fatigue, Fig. 24 Strouhal number-Reynolds number relationship for circular cylinders. $S = 0.21(1 - 21/R_e)$ for $40 < R_e < 200$ (Lienhard 1966)



Dynamic Behavior and Fatigue, Fig. 25 Cylinder and coordinate system with rigid cylinder mode shown (Blevins 1994)



D

ω_s : $2\pi f_s$, circular vortex shedding frequency, radians/s, where the vortex shedding frequency f_s is given by Eq. 69

t : time, s

F_L : lift force (force perpendicular to the mean flow) per unit length of cylinder

This force is applied to a spring-mounted, damped, rigid cylinder, shown in Fig. 25. The cylinder is restrained to move perpendicular to the flow. The equation of motion of the cylinder is

$$m\ddot{y} + 2m\xi\omega_y\dot{y} + ky = \frac{1}{2}\rho U^2 D C_L \sin \omega_s t \quad (72)$$

where

y : displacement of the cylinder in the vertical plane from its equilibrium position

m : mass per unit length of the cylinder, including added mass

ξ : structural damping factor

k : spring constant, force/unit displacement

$\omega_y = (k/m)^{1/2} = 2\pi f_y$, circular natural frequency of cylinder

Solutions to this linear equation are found by postulating a sinusoidal steady-state response with amplitude A_y , frequency ω_y , and phase φ (Blevins 1994)

$$y = A_y \sin(\omega_s t + \phi) \quad (73)$$

and substituting this equation into Eq. 72. The result is (Thomson 1988)

$$\frac{y}{D} = \frac{\frac{1}{2}\rho U^2 C_L \sin(\omega_s t + \phi)}{k \sqrt{\left[1 - (\omega_s/\omega_y)^2\right]^2 + (2\xi\omega_s/\omega_y)^2}} \quad (74)$$

where the phase angle is defined by

$$\tan \phi = (2\xi\omega_s\omega_y)/(\omega_s^2 - \omega_y^2) \quad (75)$$

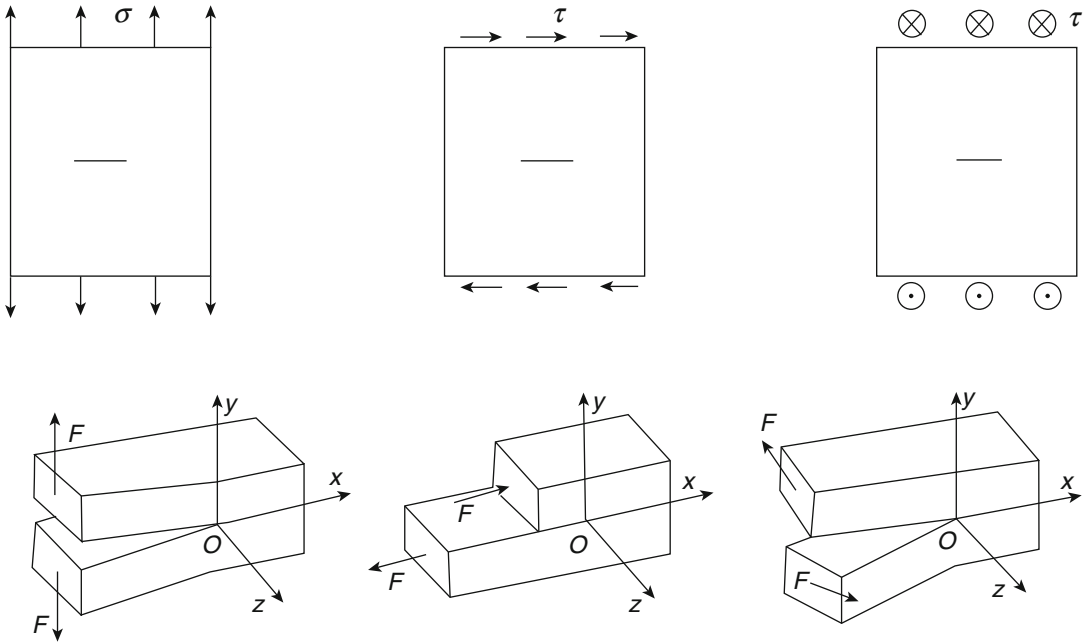
The phase angle shifts by 180 degrees as the cylinder passes through $f_s = f_y$. The response is largest when the shedding frequency approximately equals the cylinder natural frequency, $f_s = f_y$, a condition called resonance (Blevins 1994). Using Eqs. 73 and 74, the resonant vibration amplitude is

$$\left. \frac{A_y}{D} \right|_{f_y=f_s} = \frac{\rho U^2 C_L}{4k\xi} = \frac{C_L}{4\pi S^2 \delta_r} \quad (76)$$

Fatigue and Fracture

Definition

Fatigue is the process of progressive localized permanent structural change occurring in a material subjected to conditions, which produce fluctuating stresses and strains at a point or some points and which may culminate in cracks or complete fracture after a sufficient number of fluctuations (ASTM 1972). Fracture can be



Dynamic Behavior and Fatigue, Fig. 26 Three different crack modes

categorized under three categories of opening, shearing, and tearing modes (Fig. 26), which are termed as the modes I, II, and III, respectively (Yu et al. 2011).

1. Under tensile stress σ perpendicular to the crack plane surface, the crack is opened; this is termed as the opening mode, mode I (a).
2. Under shear stress τ parallel to the crack plane surface and perpendicular to the crack tip, the crack is staggered along the crack plane surface; this is termed as the shearing mode, mode II (b).
3. Under shear stress τ parallel to both the crack plane surface and the crack tip, the crack tears the material; this is termed as the tearing mode, mode III (c).

Mechanism

Fatigue is one of the primary reasons for the failure of structural components. The life of a fatigue crack has two parts, initiation and growth (propagation). Crack initiation is a process that forms cracks on the surface of a material. Fatigue leads to progressive and localized structural

damage when any material experiences cyclic loading. Due to cyclic loading, the material experiences continuous and repeated loads or forces at various points on the material. When such loads are high enough, they lead to crack initiation, growth of cracks, and ultimately a fracture. Crack growth is defined as the widening, lengthening, or increase in the number of cracks on a particular surface. The growth of a crack can be attributed to one or more factors including the application of additional loads, thermal stresses, stress concentrations, and repetitive shrinkage/expansion cycles. Crack growth can significantly reduce the load-carrying capacity of a structure and can occur in a variety of materials including metal, wood, and concrete. It is also known as crack propagation (<https://www.corrosionpedia.com/dictionary>).

Fatigue Assessment

Analysis Based on S-N Curves

Subsea production systems and its components such as pipelines, risers, unsupported free spans, welds, J-lay collars, buckle arrestors, riser

touchdown points, and flex joints should be assessed for fatigue. Potential cyclic loads that can cause fatigue damage include vortex-induced vibrations (VIV), wave-induced hydrodynamic loads, platform movements, and cyclic pressure and thermal expansion loads. The fatigue life of the component is defined as the time it takes to develop a through-wall-thickness crack of the component.

For high cycle fatigue assessment, fatigue strength is to be calculated based on laboratory tests (S-N curves) or fracture mechanics (Bai and Bai 2005). Low cycle fatigue of girth welds may be checked based on $\Delta\epsilon$ -N curves. The S-N curves to be used for fatigue life calculation are defined by the following formula:

$$\log N = \log a - m \cdot \log \Delta\sigma \quad (77)$$

where

N : the allowable stress cycle numbers

a & m : parameters defining the curves which are dependent on the material and structural detail

$\Delta\sigma$: stress range including the effect of stress concentration

For the pipe wall thickness in excess of 22 mm, the S-N curve is to take the following form:

$$\log N = \log a - \frac{m}{4} \cdot \log \frac{t}{22} - m \cdot \log \Delta\sigma \quad (78)$$

where t is the nominal wall thickness of the pipe. The fatigue damage can be computed based on the accumulation law by Palmgren-Miner:

$$D_{fat} = \sum_{i=1}^{M_c} \frac{n_i}{N_i} \leq \eta \quad (79)$$

where

D_{fat} : the accumulated fatigue damage

η : allowable damage ratio, to be taken as 0.1

N_i : the number of cycles to failure at the i th stress range defined by S-N curve

n_i : the number of stress cycles with stress range in block i

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Dynamic Ice Loads and Structural Responses

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Introduction

Importance of the Subject for Engineering

For the designer of a structure deployed in ice-covered waters, the extreme ice load associated with a return period (typically 100 years) is not the only problem to solve. Observation and measurement on full-scale structures have revealed that ice load is usually time-varying, and in some cases severe structure vibration may arise due to ice loading. The same as other dynamic excitations (e.g., by wave, wind, earthquake), dynamic ice load will lead to different problems than that of extreme ice load, which are illustrated in Fig. 1.

- For a structure itself, if the cyclic frequency of dynamic ice load is close to one of the natural frequencies (typically the first-order natural frequency) of structure, resonance might occur, and structure response is likely to be amplified significantly. On the other

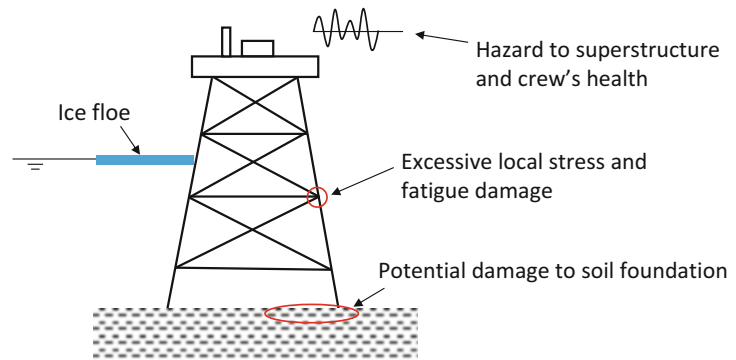
hand, dynamic ice loading will cause fluctuating response (stress). Engineering practice has proved that cyclic stress may cause fatigue damage in long term and a local fatigue crack at local spot might propagate rapidly and lead to collapse of structure.

- For the structures with fixed foundation at seabed (e.g., with pile, gravity), strong vibration may reduce the resistance of soil and cause instability or even overturning of structure.
- Besides the structure directly undertaking dynamic ice loading, strong vibration of superstructure (e.g., topside of offshore platform) may cause other hazards. For example, some type of equipment (e.g., long-span pipeline system) may be very vulnerable to vibration, which shall be considered during the design stage.
- Severe structure vibration may lead to other “nonstructural” problems. For example, vibration may have harmful effect to the crew working onboard offshore platform. According to research, people’s body is vulnerable to a certain range of vibration frequencies, and the relative direction (when the person is standing, sitting, or lying down) of vibration also influences the effect.

Difficulty of the Subject

The difficulty of understanding ice load is mainly due to the complex behavior of natural ice (Sanderson 1988), and it can be very difficult to predict the ice load on a given structure. The dynamic ice load is triggered by “cyclic” failure of ice, which is an even more complex physical

Dynamic Ice Loads and Structural Responses,
Fig. 1 Potential problems caused by dynamic ice load



process. The research on dynamic ice load has been performed for decades, but there are still a number of subjects which have not been well understood yet.

The Scope of the Entry

This entry intends to demonstrate the latest achievement of research and engineering practice on dynamic ice load.

History of Dynamic Ice Load Research

Dynamic ice load was first discovered on steel jacket platforms in Cook Inlet, Alaska (Peyton 1968). The measurement on the platforms shows that structure may vibrate when ice floe pass the structure, indicating that ice load must have significant dynamic feature.

Since then, a number of tests have been done (Fig. 2). From the 1970s to 1980s, full-scale measurements were performed on the light-houses and channel markers in the Baltic Sea, in order to collect data of ice-induced structure vibration (Engelbrektson 1977, 1989). From 1999 to 2001, an extensive measurement campaign was conducted on the lighthouse called “Norströmsgrund” which is located offshore Sweden; the measured data includes ice load time series, structure accelerations, ice parameters, etc. (Jochmann and Schwarz 2001; Kärnä and Qu 2009).

In the Bohai Sea, China, another measurement campaign was conducted from 1999 to 2003. The initial purpose was to monitor the ice-induced structure vibration and then guide the icebreaking

vessel operation to mitigate the vibration. The measurement system was used for several ice seasons, and a large amount of data has been collected (Yue et al. 2001).

In addition to full-scale measurements, many model scale tests have been done to study the dynamic ice load (e.g., Toyama et al. 1983).

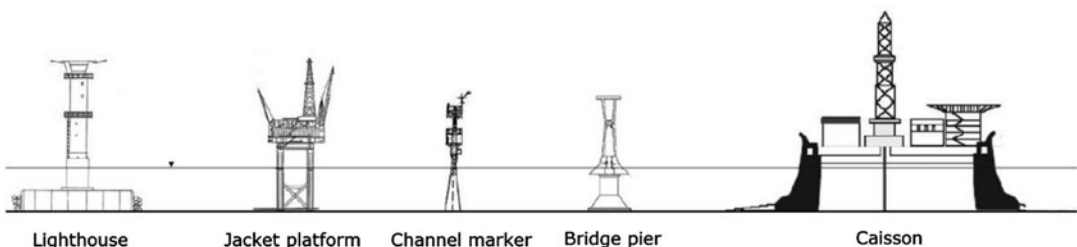
ISO 19906 (Arctic offshore structures) collects most of the latest engineering practices on dynamic ice load, and they are demonstrated in sections below.

Dynamic Ice Load on Cylindrical Structures

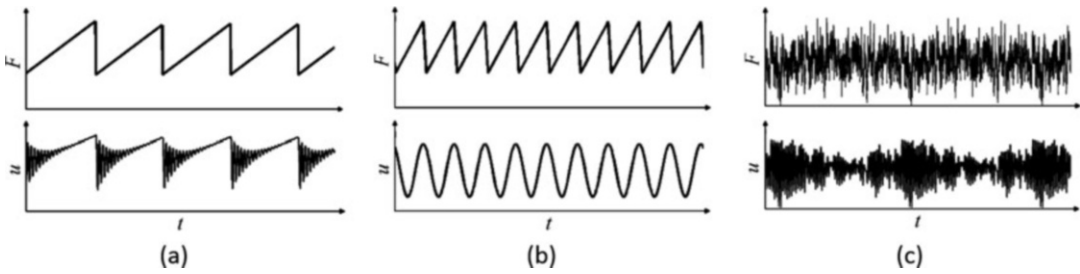
Most offshore structures have cylindrical legs or vertical surface at waterline. The measured ice load data indicates that dynamic ice load on cylinder structure has different characteristics. Depending on different ice drift velocities, three different ice load “modes” may occur (more detailed description may be found in ISO 19906) (Fig. 3):

Intermittent crushing or quasi-static ice load.

This mode is defined as intermittent crushing in ISO 19906 or quasi-static ice load in some other publications. This mode occurs when ice drift velocity is very slow, so that the period of loading-unloading cycle is much longer than the natural period of structure. Normally this mode may be considered as static load case, unless it occurs so frequent that the fluctuating stress has significant contribution to structure fatigue damage.



Dynamic Ice Loads and Structural Responses, Fig. 2 Different types of structure used to measure dynamic ice load (Bjerkås 2006)



Dynamic Ice Loads and Structural Responses, Fig. 3 Three dynamic ice load modes on cylinder structures (simulated time series, similar with that in ISO 19906)

Frequency lock-in. This mode is defined as “frequency lock-in” in ISO 19906. There have been two theories on the frequency of dynamic ice load: one theory believes that the cyclic failure process of ice sheet has its own characteristic frequency, and if this frequency happens to be close to the natural frequency of structure, resonance will occur (Sodhi 1988). Another theory believes that the frequency of ice load is controlled or “tuned” by the structure’s vibration, so the ice-induced vibration is a self-excited process (Blenkarn 1970; Määttänen 1978).

Even though the theoretical disagreement has not been resolved, ISO 19906 adopts the definition of “frequency lock-in” to describe the process when the dynamic ice load becomes very regular (almost harmonic), and ice load’s dominant frequency is close to natural frequency of structure. Due to dynamic amplification, this phenomenon is most likely to cause severe structure vibration.

Continuous crushing or stochastic ice load. When the ice drift velocity is relatively high, the fluctuating ice load becomes more irregular, and the resulting structure response becomes irregular as well. Normally the vibration magnitude by stochastic ice load is lower than that caused by frequency lock-in, but this mode can be important when contributing to the long-term structure fatigue damage (e.g., Liu et al. 2010).

Dynamic Ice Load on Conical Structure

Compared with cylinder structure, there has been less research work on dynamic ice load on

conical structures. One important data resource of this subject is from the measurement of conical structures in the Bohai Sea, China, where ice load and structure response were recorded for several ice seasons, and dynamic ice load function has been developed based on these data (Qu et al. 2006). ISO 19906 adopts the relevant methods (Fig. 4).

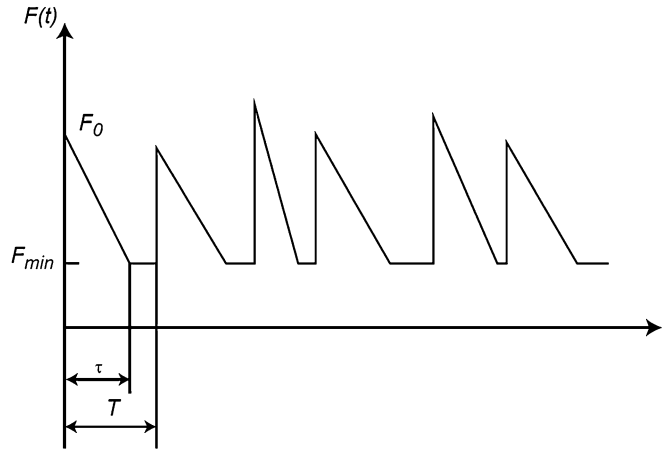
Structure Damage Due to Dynamic Ice Load

Back in the 1960s and 1970s, several structures were reported as damaged by excessive ice loading. Investigation afterwards found out that some of the structure damage incident might be (at least partly) due to dynamic ice load and associated dynamic amplification effect. For example, the winter from 1967 to 1968 was an extremely heavy ice season, and a steel jacket platform in China Bohai Bay was completely damaged by massive ice floe. The subsequent investigation concluded that the jacket structure may have experienced strong vibration before damage (Duan et al. 1994). From the 1960s to 1970s, several lighthouses in the Baltic Sea were damaged by ice (Bergdahl 1972), and investigation in recent years (Lehmann 2010) found that severe ice-induced vibration (especially frequency lock-in phenomenon) and dynamic amplification was perhaps the main reason (Figs. 5 and 6).

Apart from the consequence caused by relatively “short-term” (typically in a few minutes or even seconds) excessive ice-induced

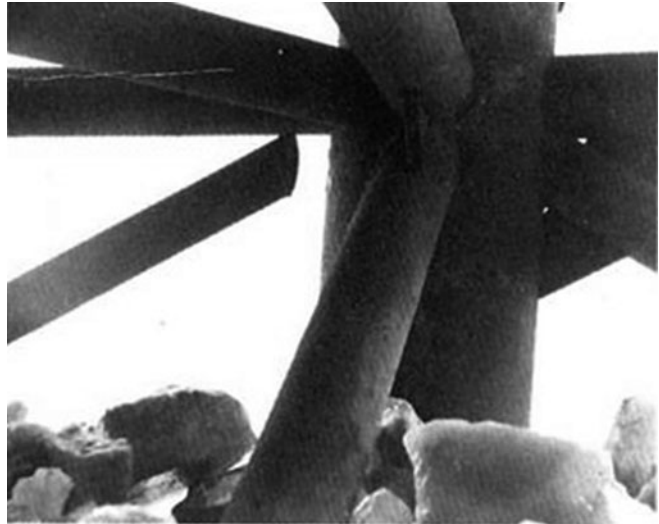
Dynamic Ice Loads and Structural Responses,

Fig. 4 Ice load function for conical structures (IceStruct JIP Guideline 2013)



Dynamic Ice Loads and Structural Responses,

Fig. 5 Jacket platform damaged by ice in the Bohai Sea (Duan et al. 1994)



structure vibration (e.g., frequency lock-in for vertical structure), long-term dynamic ice load may also lead to significant structural damage even though the magnitude is low. Structure dynamics theory and practice have concluded that although the stress magnitude is much lower than the yield limit of metal material, cyclic stress over a long period (typically a few years) may lead to fatigue damage, which is believed as the main contributor to most material failure in engineering structure.

Dynamic ice load will result in fluctuating stress in a structure, so that potentially fatigue damage may accumulate over months and years. So far, by engineering practice there was

no clear evidence of structure failure caused by ice-induced fatigue damage; thus the research work on ice-induced structure fatigue is quite limited, and ISO 19906 only includes discussion according to generic technical approach of fatigue assessment. The research by Dalian University of Technology applied the full-scale data and developed a spectral model to evaluate the ice-induced fatigue for steel jacket platforms in the Bohai Sea (Liu et al. 2010), which may be applicable for similar ice condition and structure type. During the joint industry project “IceStruct,” efforts were made to propose more guidance for designers to assess ice-induced fatigue damage (IceStruct Guideline 2013).

Dynamic Ice Loads and Structural Responses,

Fig. 6 Nygrån lighthouse damaged in Baltic Sea. (From Bergdahl 1972, photo taken by Swedish Board of Shipping and Navigation)



Dynamic Ice Loads and Structural Responses,

Fig. 7 Molikpaq during the incident on 12 May 1986 (Timco et al. 2005)



Foundation Damage Due to Dynamic Ice Load

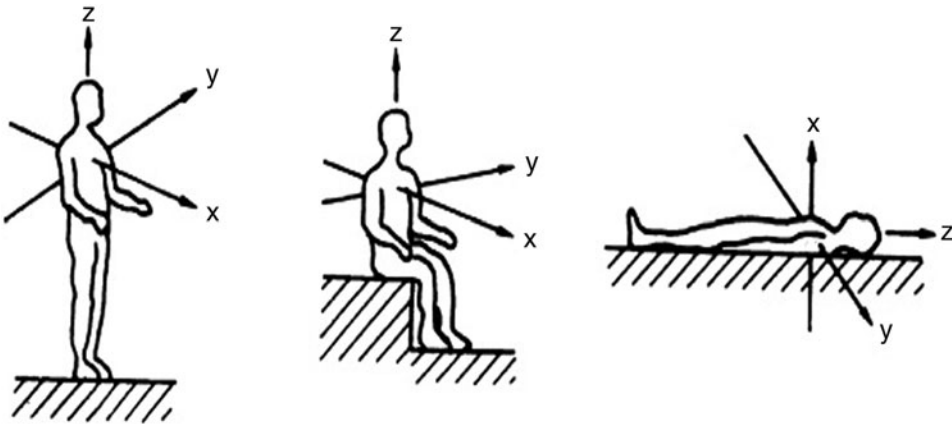
As demonstrated in Introduction, for a fixed foundation structure (e.g., pile foundation or gravity-based), dynamic ice load and vibration at foundation may reduce the resistance capacity of soil. A famous incident which is often mentioned was Molikpaq caisson deployed in the Beaufort Sea offshore Canada during the 1980s: during the winter in 1986, the Molikpaq caisson experienced

enormous amount of ice loading, and especially on 12 May 1986, a large multi-year ice floe hit Molikpaq and led to excessive ice loading and strong vibration of structure. It was believed that the structure foundation was moved and damaged due to liquefaction of soil beneath the structure.

Unfortunately, not much research work has been done on this subject, but the generic methods for geotechnical engineering are applicable provided the foundation response triggered by ice loading (Fig. 7).

Dynamic Ice Loads and Structural Responses,

Fig. 8 Pipeline broken by strong ice-induced vibration on jacket platform (Yue et al. 2001)



Dynamic Ice Loads and Structural Responses, Fig. 9 Illustration of human body reaction on vibration in different positions (ISO 2631 standard, to be replaced by new version)

Superstructure Damage Due to Dynamic Ice Load

As demonstrated in introduction, dynamic ice load may lead to strong vibration of superstructure. For example, a jacket platform for oil production normally has many types of equipment on the topside, which will be affected by the deck's vibration under dynamic ice loading.

In the winter 1999, a jacket platform in China Bohai Sea was hit by a large ice floe, and strong frequency lock in mode vibration lasted for almost 10 min. The survey afterwards found out that one pipeline was broken by the vibration (Fig. 8), fortunately the damaged pipeline segment was for fire extinguishing purpose and contains no

dangerous gas or liquid. However, the incident indicates that if ice-induced vibration is likely to occur on a structure, the design of equipment/pipeline system, etc. shall consider the effect of vibration, and the mechanism is similar with earthquake design (base vibration).

Crew Health Due to Ice-Induced Vibration

Similar with the effect on superstructure (equipment/pipeline, etc.), strong ice-induced vibration might also have negative effect to the crew working on the offshore installations. From the 1980s to 1990s, several jacket platforms were

installed in China Bohai Sea, but ice-induced vibration was not properly taken into account during the design process.

In the first ice seasons, many crew members reported that it was very uncomfortable during ice-induced vibration process, and the feeling was similar with sea sickness. Furthermore, some crew members even reported that it was difficult to fall in sleep during strong vibration, which might affect their work the next day. To systematically evaluate the human body reaction by ice-induced vibration, extensive study was performed by comparing the measured vibration time series and industry standards (e.g., ISO 2631, Fig. 9).

Cross-References

- [Offshore Structure Design Under Ice Loads](#)
- [Offshore Wind Turbine-Ice Interactions](#)
- [Sloping Structure–Level Ice Interactions](#)

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Dynamic Positioning (DP)

- [Maneuverability of Polar Vessel](#)
- [ROV Dynamic Positioning](#)
- [Thruster-Assisted Mooring](#)

Dynamic Positioning Control

- [ROV Dynamic Positioning](#)

Dynamic Positioning Control System

► ROV Dynamic Positioning

Dynamic Positioning in Ice

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Synonyms

DP, Dynamic positioning; DTF, Detection, tracking and forecasting; DYPIC, Dynamic positioning in ice; HSVA, Hamburg Ship Model Basin; POSMOOR, Position-mooring

Definition

Dynamic positioning refers to a system to automatically maintain position and heading of a ship or floating marine structure by using its propellers and thrusters. This allows operations at sea where mooring or anchoring is not feasible due to deep water and congestion on the sea bottom. It is much used in the offshore industry.

Introduction

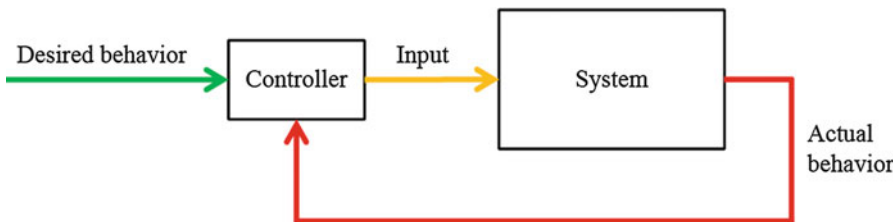
The main components of typical dynamic positioning (DP) system are introduced briefly for

marine operations. The basic principle and functions of DP system are presented. In the ice-covered area, there may exist a lot of challenges regarding DP operations in ice. The development of DP technology applied to marine operations in the ice-covered sea is presented. Finally, experiences regarding realistic marine operation is collected to give some lessons which are useful to improve marine operability in the ice region.

Dynamic Positioning System

Marine cybernetics is a multidiscipline that combines marine technology, control theory, and industrial information technology to address problems of monitoring and control on marine systems. The core of automatic controls is to deal with how you can get a system to behave the way you like. The objective is to find a technically feasible way to act on a given system so that it adheres, as closely as possible, to some desired behavior. A simplified control loop in concept is shown in Fig. 1.

Marine control system is composed of guidance system, navigation system, and control system. The guidance system generates the desired trajectories (such as position, velocity, and acceleration). Waypoint generator establishes the desired way points. Waypoint management system updates the active waypoint based on current position of the ship. Reference computing algorithms generate a smooth feasible trajectory. The navigation system generates the appropriate feedback signals with assistance of sensors (GPS, radar, inertial measurement units, accelerometers, gyro-compass, etc.). The signal will be processed by quality checking before transforming the



Dynamic Positioning in Ice, Fig. 1 Control loop in concept

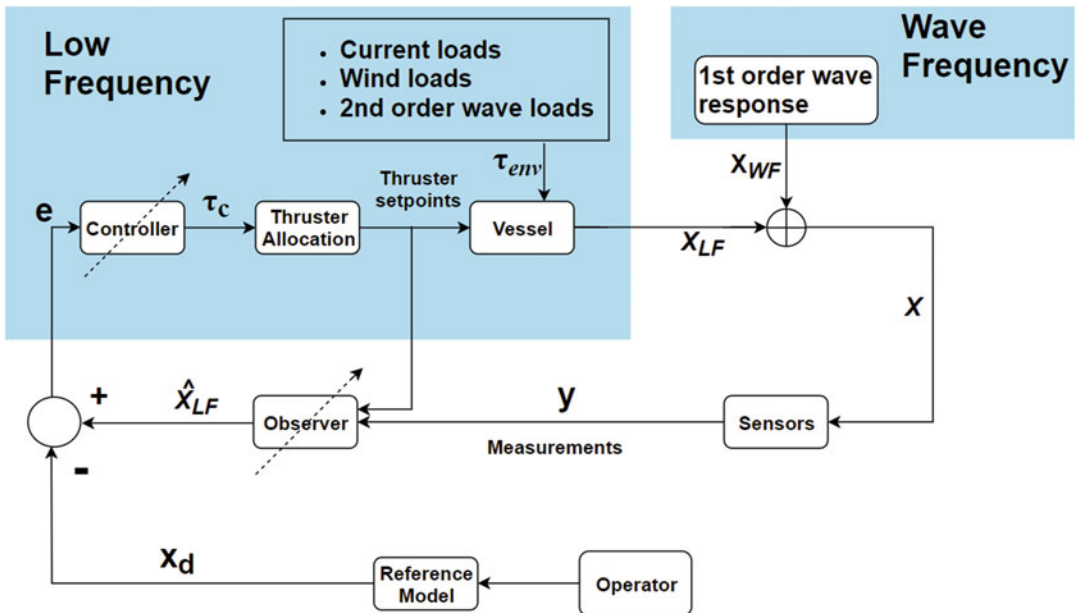
measured motion to that of the origin of the relevant reference frame. The control system generates appropriate commands. In the control system, wave filter/observer, controller, and control allocation are necessary. The wave filter (observer) is to fill out high-frequency signals from the measurements. The controller is to generate command based on the difference between the desired position and actual position. Control allocation is used to distribute the command into individual actuator commands.

Based on different operational requirements, there are three control objectives in general. The first one is to control the low-frequency motion including autopilots, dynamic positioning (DP), and POSMOOR. The second is to control the wave-frequency motion (heave, roll, and pitch stabilization). The last one is to control both low-frequency and wave-frequency motion such as DP with roll and pitch stabilization in high seas, course keeping, and roll stabilization.

Dynamic positioning system is used to solve the problem of simultaneous control of three states, namely horizontal positions and heading (surge, sway, and yaw motion). A typical DP architecture is composed of process plant,

measurement system, signal processing, control system, reference model, thrust allocation, and power management system. The process plant is the object to be controlled. The measured signal is processed by filter or observer onboard. The control system gives command to the propulsion system based on certain mathematic algorithm. Reference model generates a feasible way for the object to follow. The general command should be distributed to each thruster with thrust allocation system. The power management system delivers the necessary power with certain limits to the propulsion system.

In most operational conditions, the desired trajectory is represented by slowly varying motion (low-frequency motion) compared to the oscillatory motion induced by the waves (wave-frequency motion). The wave-frequency motion is due to first-order wave-induced loads, while slowly varying motion is mainly due to second-order wave loads, current, and wind. From the control plant sense, the main components of DP system are presented in the Fig. 2. Operator gives a command to the reference model which generates the desired position. The observer is to estimate the actual position based on the measurements from sensors. The controller



Dynamic Positioning in Ice, Fig. 2 Marine control plant DP system

collects both the actual and desired position to calculate the required forces to overcome the disturbance from environment at low-frequency range.

Arctic DP Challenges

Challenges from Sea Ice Loads

One of the main challenges for the DP system is from ice. Ice exists in a lot of features due to complex environment in the Arctic area. The normal sea ice includes level/intact ice, ice ridges, ice floes, rafted ice, and rubble piles. The extreme hard glacial ice includes icebergs, ice islands, and growlers. The ice load due to different ice features are varying significantly. For an icebreaker with normal size operating in ice, it may be subject to ice load less than 2 MN in broken ice (ice floes, rafted ice, and rubble piles) and within 5 MN in level ice, larger than 10 MN in ice ridges and larger than 100 MN in the glacial ice. The broken ice is like melt ice cream. The level ice is like hard ice cream, while the glacial ice is like concrete. Due to the limited thrust of propulsion system, DP operation can only be applied in the broken ice field.

The ice force in the real world is changing rapidly and varying significantly, containing low-frequency, wave-frequency, and high-frequency components. The duration of high peaks may last long which is not expected. Therefore, it is very difficult to design a DP system to cope with ice load completely. At present, there is not any theoretical model which can be applied for DP design. We have to rely on both ice model test and numerical simulation though there are a lot of limitations to them.

Changes in Ice Drift Direction

In most ice covered area, ice management are required to reduce ice load impact on the stationkeeping vessels with assistance of icebreakers. As an effective tool, icebreaker is often used to break large ice sheet into small pieces according to requirements of ship owners. Sometimes it can also be used to tow icebergs away when the icebergs could be observed before arriving at the operation site.

The number of icebreakers needed depends on the in situ ice conditions. In general, the most powerful icebreaker should operate on the upstream which is far from the operation site. It could cut large ice sheet into small ones. The smaller icebreaker should operate near the operation site, which breaks the ice floes into pieces. In order to decrease ice load impact, DP vessel is required to bow into the incoming ice. However, ice drift direction is changing in the ice-covered area due to wind, current, wave, etc. Therefore, ice drifting direction has to be monitored from time to time. The area of ice field to be managed should also be adjusted based on the ice drift.

Difficulty with Conventional DP Control

When a vessel operates in sea ice, the environmental forces are substantially different from conventional open-water. Conventional DP systems are not efficient to be used any more according to Gürtner et al. (2012), Hals and Jenssen (2012), Jenssen et al. (2009), and Kerkeni et al. (2013a).

It is difficult or even impossible for a polar vessel to move sideways due to limited deliverable thrust compared to high transverse ice loads. The thrust command using conventional DP systems could not be well implemented for side motion. The vessels should be orientated always against the drifting ice with the bow or the aft end (Kuehnlein 2009). Moreover, it has to turn with a small attack angle and could not rotate on the spot in ice which is significantly different from that in open waters. In addition, when DP system is used in the ice-covered sea directly, the filter or observer will not give good estimation of vessel status and lag behind dramatically. In order to improve the filter's tracking capability, the filter has to filter out less head/position, but this will bring in more aggressive utilization of thruster (Jenssen et al. 2009).

Arctic DP Control Strategy

As we mentioned, conventional DP has insufficient ability to keep the position of a ship in ice. New control strategies have been proposed. Considering the priority of keeping the vessel heading

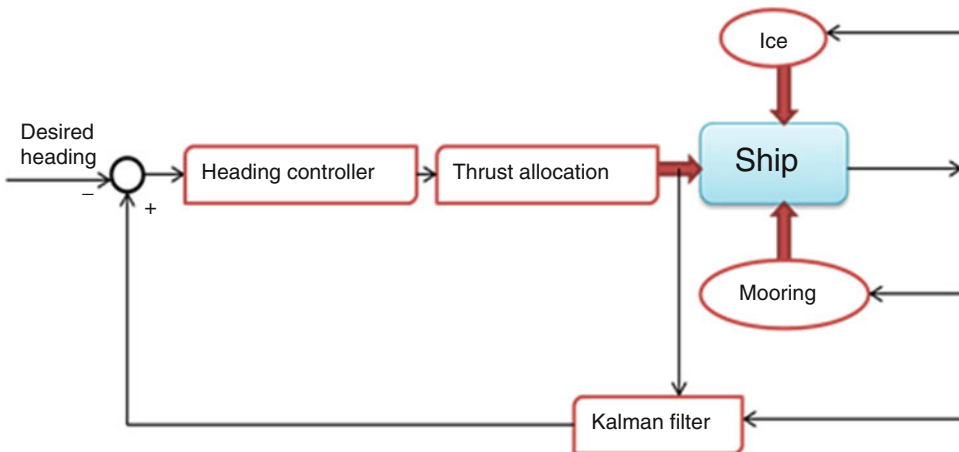
toward ice drift direction, Zhou et al. (2013) tried a heading controller combined with a mooring system to keep the position of a polar ship in level ice field. The heading controller was used to keep the vessel align with the drifting ice, while a mooring system was used to provide a reactive force to compensate for the mean drift loads of the environment due to ice. A Kalman filter was also applied to filter out the high-frequency components of measured heading signals. The overall schematic of control strategy is shown in Fig. 3.

Kjerstad et al. (2011) introduced a disturbance rejection design with acceleration feedforward to augment performance of conventional control system. The design could be used for a wide range of applications where disturbances can be suppressed to give a resulting linear system characteristics. The disturbance rejection could offer compensation for environmental forces and model uncertainties based on acceleration measurement. Another control algorithm called automatic icevaning was proposed by Kim et al. (2018). The idea is that DP vessel controls its position in a predefined position while heading up toward the ice drifting direction. DP model tests in ice were carried out to demonstrate the performance of the icevaning control method in the ice tank of Korea Research Institute of Ships and Ocean Engineering.

DP Capability in Ice

The capability plot, often presented as a polar diagram with a number of envelopes, is used to establish the vessel's capability to keep position in a certain environment with a certain combination of thrusters. In basic DP design, the capability assessment of a vessel is often based on static environmental and propulsion forces.

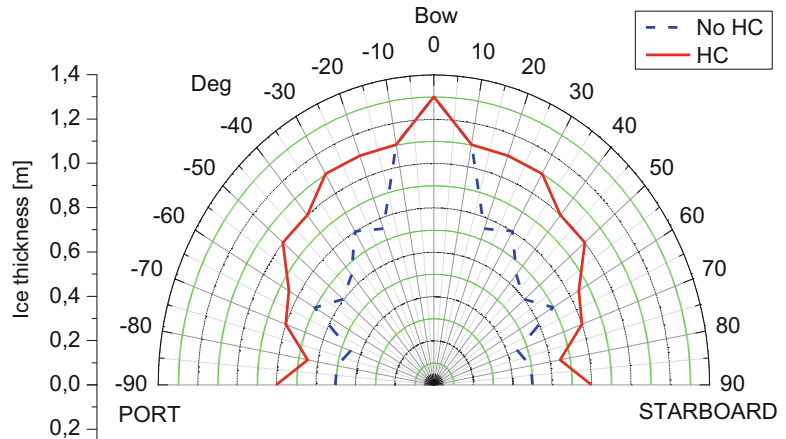
Zhou et al. (2012) simulated heading control of a moored icebreaking tanker under level ice regime where a dynamic ice simulator was developed to consider interaction between ice and hull. For the first time, the method of deriving capability plots of moored and DP vessel in ice based on dynamic ice-hull interaction was proposed in terms of maximum ice thickness that the vessel could resist under different ice drift angles. The result is shown in Fig. 4, where "HC" means the cases with heading controller, while "No HC" means the cases without heading controller. It is easy to find that heading controller is helpful to improve the capability of moored vessel in level ice. Later, Kerkeni et al. (2013b) proposed a method for establishing capability plots in broken ice field. The method is also based on dynamic numerical simulations by considering ice concentration, ice drift velocity, ice drift angle, ice thickness, and so on. Some experimental tests were used to validate the proposed method.



Dynamic Positioning in Ice, Fig. 3 Block diagram of control strategy

Dynamic Positioning

in Ice, Fig. 4 Capability plot for moored tanker with and without HC (Zhou et al. 2012)



Model Test of DP in Ice

From 2010, the HSVA (Hamburg Ship Model Basin) started to perform a European research and development program named DYPIC (dynamic positioning in ice), which will last 3 years. The project is jointly supported by the national research agencies of Germany, France, and Norway. The aim is to improve the testing capability for dynamic positioning in the ice basin conditions. A toolbox for the numerical prediction of stationkeeping capabilities of a polar ship in ice is desired as the scope of the project (Kerkeni et al. 2014).

Many DP model tests under intact ice conditions were carried out in the HSVA in 2012, where a conceptual model-scale Arctic drillship with dynamic positioning system was tested. The vessel model was equipped with six azimuth thrusters. Three of them are located in the stern and others in the bow region. A scale factor of 1:30 with Froude scaling was applied throughout the model testing campaign. The model ice was manufactured according to the standard method within HSVA. The flexural bending is the main failure mode. Therefore, the bending strength of ice should be made as expected based on experience.

The main goal of DP testing was to ensure the heading, surge, and sway of the vessel model to stay as close as constant desired values, which are preset by operators. Any offsets from original position donates deviations from the target. The

displacements of the model, general command by the DP control system, and the delivered thrust of the actuators were measured and then sent back to the DP controller as feedback. Furthermore, all these measurements were recorded and stored in the data acquisition system for further analysis and processing (Baardson 2012; Jenssen et al. 2012; Van der Werff 2012; Dal Santo and Jochmann 2012; Haase et al. 2012; Haase and Jochmann 2013). The movement of drifting ice was achieved by towing the vessel through the ice field rather than pushing ice toward the ship model. The vessel model navigated with its own power in front of the main carriage. The thrusters were controlled with a model-scale DP system through electric engines by adjusting the RPS and the azimuth angles. When the main carriage moved along the ice tank, the vessel model was towed through the ice. The distance from the main carriage to the vessel and the orientation of the vessel were continuously measured by the qualisys motion tracking system during the tests.

Arctic DP Operations

A system called ice detection, tracking, and forecasting (DTF) is often used to ensure a safe and reliable arctic operation. The function of DTF is to forecast weather, different types of ice, tide, and currents. It could be also applied to collect information related to ice conditions and handle data, which is helpful to find most severe ice condition

for alarm system. Some devices are used in the DTF process, such as satellites, airborne plane and helicopters, unmanned aerial vehicles, sonars, autonomous underwater vehicles (AUV), and radar.

The Aurora Borealis, a dynamically positioned vessel for the European Polar Research Icebreaker Consortium was designed as a combination of a heavy icebreaker, a deep-sea drilling ship, and a multipurpose research vessel (Deter et al. 2009). This multipurpose research vessel is a heavy icebreaker with the highest ice class for worldwide operation. It is powered to continuously break in more than 2.5 m of multi-year ice and enables to manage ridges up to 15 m. The ship shall perform research tasks including scientific drilling year-round in the Arctic and Antarctic without any support vessels. The capacity of performing stationkeeping operations in drifting solid ice of more than 2.0 m thickness during drilling and other research tasks is mandatory requirement. A powerful propulsion system with a capacity up to 108 MW is installed for the various tasks including transit at 16 knots, icebreaking, and stationkeeping in ice. The ice model tests for stationkeeping in drifting solid ice of up to 2.0 m thickness, i.e., icebreaking in a practically stationary mode, were carried out in two ice tanks in Helsinki and Hamburg. Based on the results of the Aurora Borealis icebreaker design effort, it was concluded that stationkeeping operation in solid drift ice is feasible.

Keinonen et al. (2000) reported that station keeping with dynamic positioning systems was employed for drilling and diving operations in heavy ice conditions in the offshore Sakhalin. The water depth is 30 m. As the first ever such operation, the DP operation was supported by two icebreakers: Smit Sakhalin and Magadan. The operation lasted 6 weeks in varying ice conditions including ten tenths of ice cover and ice pressure for a level ice thickness ranging from 0.7 to 1.5 m. The operation was carried out successfully with the total ice down time 22%. The limitations on overall performance were believed to be caused by the limited ability of the icebreakers to manage the ice.

The related drilling operations using dynamic positioning in ice in the Arctic Ocean to recover deeply buried sediments was also reported by Moran et al. (2006). In 2004, a convoy of three icebreakers headed north to begin the Arctic Coring Expedition, IODP Expedition 302. The site water depth ranges from 1100 to 1300 m. Ice floes with thickness of 1–3 m covered 90% of the ocean surface, and ice ridges as high as several meters were encountered where floes converged. The ice drifted at speeds of up to 0.15 m/s and changed direction over short time periods, sometimes within 1 h.

A detailed ice management plan had been made before the operation including radar, helicopter, onboard weather observation, and ice monitoring system for DTF. The drilling vessel, the Vidar Viking, remained at a fixed location. Two icebreakers were used to escort drillship. The Sovetskiy Soyuz icebreaker is nuclear-powered which is believed to be the most powerful icebreaker. It broke against the first oncoming heavy ice floes on the upstream. The Oden is powered with diesel-electric which is the last defense to the drillship Vidar Viking by circling on the upstream of the drifting ice. The three ships coordinated their efforts through a central Fleet Manager, at times on a minute-to-minute basis.

The fleet and ice management teams successfully ensured the smooth drilling operations to recover cores from three sites despite severe ice conditions. The drillship Vidar Viking were able to maintain within a watch circle of 50–75 m for drilling up to 9 days. However, manual positioning rather than automatic control mode was used since automatic control lost its function. There were a lot of ice floes around the hull without open water. The vessel cannot move laterally due to high ice load. The operator had to steer bow into the changing ice drift directions continuously.

Conclusion

So far there is no much experience in arctic DP operating in severe ice conditions. To make Arctic

DP operation safe and effective, a well-prepared ice management system is crucial especially in heavy ice conditions. Provided efficient and effective ice management is available, DP operations should be feasible with a DP control system that is designed and capable of compensating for the quickly changing ice forces. However, this will be at the cost of increased usage of thrusters and thus bring in additional wear and maintenance. Moreover, heading control should be prioritized to the position control in order to reduce ice impact. Therefore, ice drift direction should be monitored from time to time.

Cross-References

- [Ice Management in Offshore Operations](#)
- [Numerical Simulation of Ice-Going Ships](#)
- [Offshore Structure Design under Ice Loads](#)
- [ROV Dynamic Positioning](#)

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Dynamic Positioning of Remotely Operated Vehicles

- [ROV Dynamic Positioning](#)

Dynamic Positioning of Underwater Robotic Vehicles

- ▶ [ROV Dynamic Positioning](#)

Dynamic Segment-Based Predominant Learning Swarm Optimizer (DSPLSO)

- ▶ [Obstacle Avoidance Technology for Underwater Vehicle](#)

Dynamic Positioning System

- ▶ [Mooring System](#)
- ▶ [ROV Dynamic Positioning](#)
- ▶ [Station-Keeping System for VLFS](#)

DYPIC, Dynamic Positioning in Ice

- ▶ [Dynamic Positioning in Ice](#)

ECMWF - European Centre for Medium-Range Weather Forecasts

► [Resource Assessment](#)

Ecological Sensitive Areas

► [Marine Ecological Red Line](#)

Ecological Vulnerable Areas

► [Marine Ecological Red Line](#)

Economic Assessment

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Definition

Economic assessment for a farm of offshore renewable energy devices refers to estimation of the levelized cost of energy including both capital

expenditures and operation and maintenance expenditures as well as the lifetime electricity generation.

LCOE (Levelized Cost of Energy)

There exist many direct and indirect uses of offshore renewable energy. But, most of the devices developed today, offshore wind turbines, wave energy converters, and ocean and tidal current turbines, are designed to generate electricity for commercial use. Cost of energy (or electricity) is the most important criterion for developing such technologies.

The levelized cost of energy (LCOE) is defined as the average total cost to build and operate a power-generating device over its lifetime divided by the total energy output of the device over that lifetime (Wikipedia 2018). In this section, energy means electricity, and typical lifetime of offshore renewable energy devices is 25 years. Using LCOE, we can compare different technologies of electricity generation on a consistent basis.

Formally, LCOE is given by

$$\begin{aligned} \text{LCOE} &= \frac{\text{sum of costs over lifetime}}{\text{sum of electricity produced over lifetime}} \\ &= \frac{\sum_{t=1}^n \frac{I_t + M_t}{(1+r)^t}}{\sum_{t=1}^n \frac{E_t}{(1+r)^t}} \end{aligned}$$

I_t : capital expenditures (CAPEX) in the year t

M_t : operation and maintenance expenditures (OPEX) in the year t

E_t : electricity energy generated in the year t

r : discount rate

n : expected lifetime

The LCOE refers to the lifetime cost and electricity production, and it has a unit of, for example, Euro per kWh. From the cost point of view, it includes both CAPEX and OPEX. CAPEX refers to the initial costs to acquire and install the physical assets, provision of services, and rental of equipment/means for construction work. For example, for an offshore wind farm, it includes design, fabrication, transport, and installation of wind turbine, foundation, electrical equipment like cables and substation, rent for transport vessels, crane vessels and cable laying vessels, marine service such as geotechnical survey, environmental impact study, and project financing and management. OPEX refers to the cost associated with operation and maintenance of the producing assets. For example, it includes continuous costs for personnel, spare parts, rents, supply vessels, and consuming materials that are used for operation, maintenance, repair, and replacement. CAPEX occurs once at the beginning of the project, while OPEX is continuous when the project is running.

Another important part of the LCOE is the lifetime electricity production in terms of kWh. Estimation of lifetime production of offshore renewable energy devices needs to consider the long-term variation of environmental conditions as well as the availability of the devices. For offshore wind turbines, typically the wind turbine power curve is used to estimate the electricity output, taking into account the probability distribution of mean wind speed. This means offshore wind turbines do not always operate at the rated value of the power output, and they operate at a capacity factor (which is defined as the average generated power divided by the rated power) of 40–60%. When an offshore wind farm is considered, potential wake effect on the power production of the whole wind farm should be properly estimated. The electricity output from a tidal turbine farm can be estimated in a similar way. The lifetime

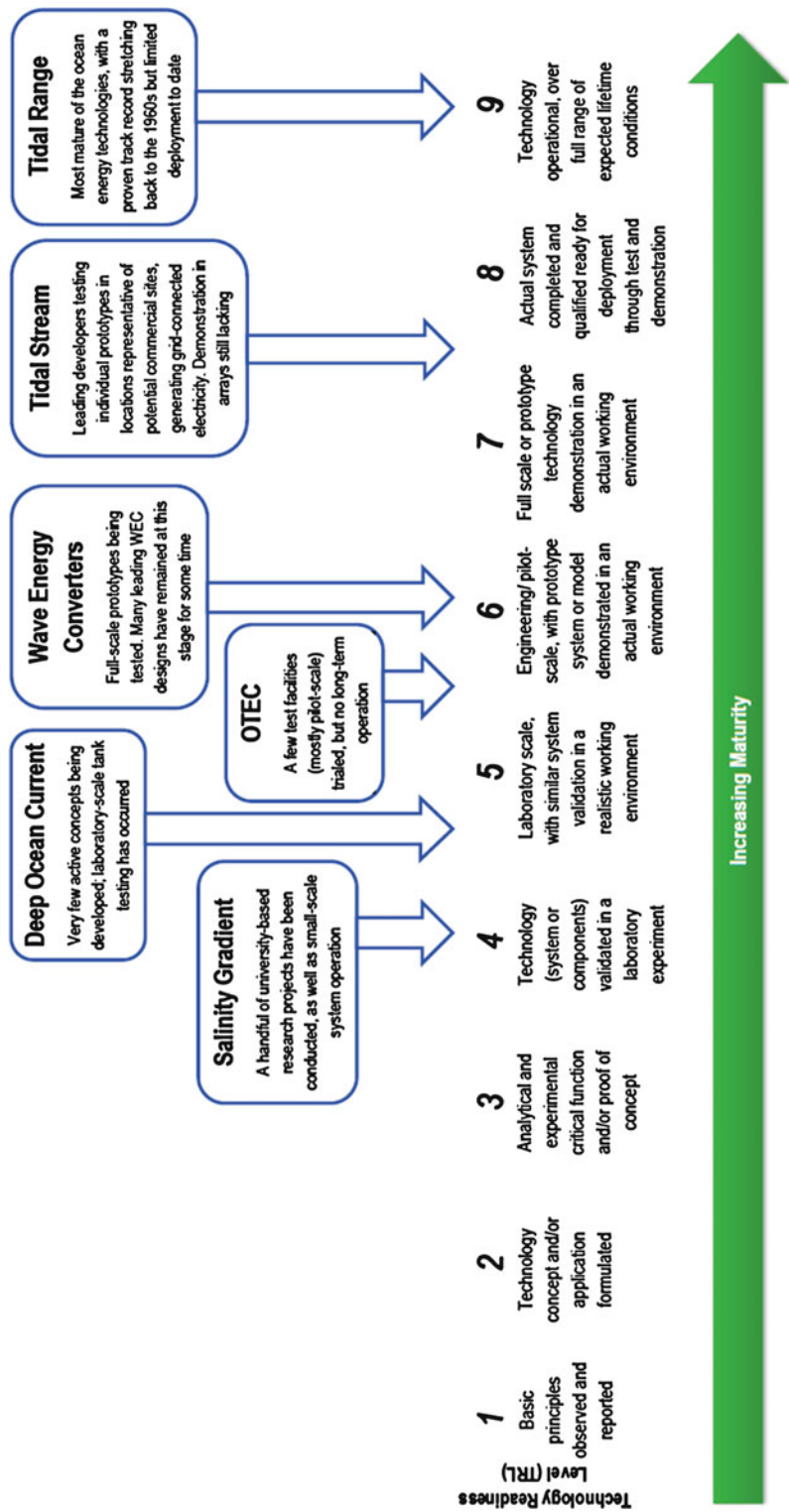
power output from wave energy converters is more complicated and depends on both wave height and period, as well as their joint distribution.

The LCOE for offshore renewable energy devices strongly depends on the Technology Readiness Level (TRL) for different electricity generation technologies. It should be noted that both wave energy and tidal turbine technologies are in the early stage of development, while offshore wind turbine technology, in particular with bottom-fixed foundations, is relatively mature and has already been commercialized. That means there is a significant uncertainty about the LCOE estimation for wave energy converters and tidal turbines. While, there still exists but less uncertainty for the LCOE estimation of offshore wind farms, although it is not easy to get these values from the developers. On the other hand, offshore wind technology evolves, and the LCOE is continuously decreasing.

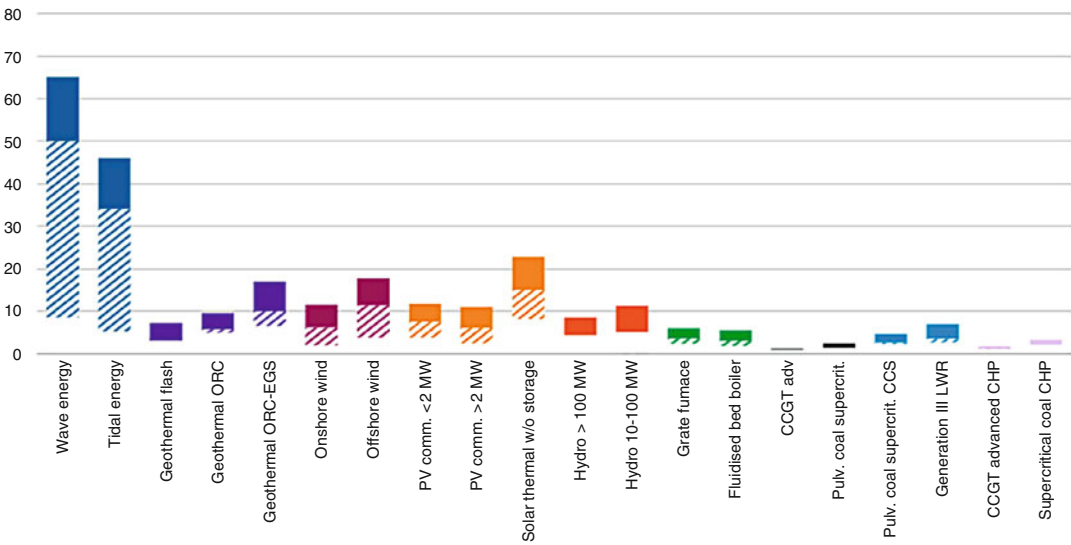
As shown in Fig. 1, large-scale tidal turbines have been developed and tested at sea, leading to a TRL of 7, but the array economics needs to be demonstrated (Mofor et al. 2014). As for wave energy, most of the concepts are still in the development phase with a typical TRL of 5, meaning that this technology is partially validated, but not fully demonstrated in relevant environment. Some leading WEC concepts have researched TRL 6 and 7, with demonstrations and prototypes at the sea. But in general, wave energy converters are less mature than tidal turbines.

Current Status and Future Prediction

LCOE is a universally adopted parameter to compare the economics of different electricity generation technologies. According to Magagna and Uihlein (2015), a comparison of the LCOE for alternative renewable energy and conventional energy technologies, including wave energy, tidal energy, onshore wind, offshore wind, solar PV, hydropower, and power plant with coal, is shown in Fig. 2. The solid bars indicate the cost range as per 2015, while the shaded bars indicate the expected future cost reductions in 2050. The LCOE of onshore wind is 7–11 Euro cent per kWh, which is similar to that of small-scale

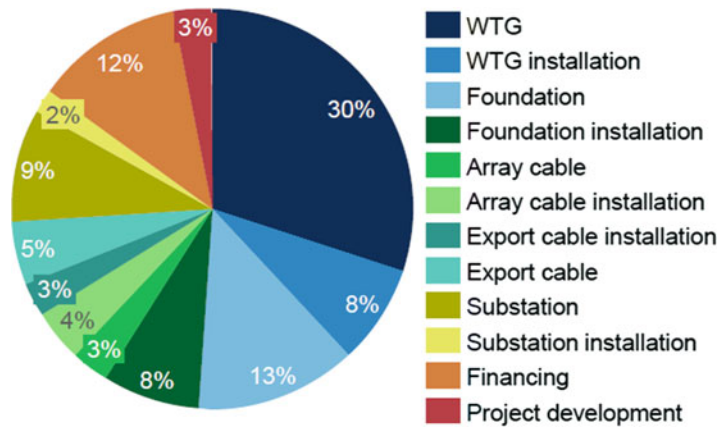


Economic Assessment, Fig. 1 Technology Readiness Level (TRL) for ocean energy (Mofor et al. 2014)



Economic Assessment, Fig. 2 LCOE (Euro cent per kWh) for alternative renewable and conventional energy technologies (Magagna and Uihlein 2015)

Economic Assessment, Fig. 3 Cost breakdown for typical offshore bottom-fixed wind farms in Europe (MAKE Consulting 2016)



hydropower stations and 50% higher than that of large-scale hydropower plants. The LCOE of offshore wind is 12–18 Euro cent per kWh, which is about twice of the onshore wind today, but potentially can be reduced to the same level by 2050. The LCOE of wave and tidal energy devices today is 50–65 and 35–45 Euro cent per kWh, respectively, which is much higher than other technologies and also shows a larger scatter among the different devices. But there is a big potential for cost reduction if both technologies are commercially developed in large-scale farms.

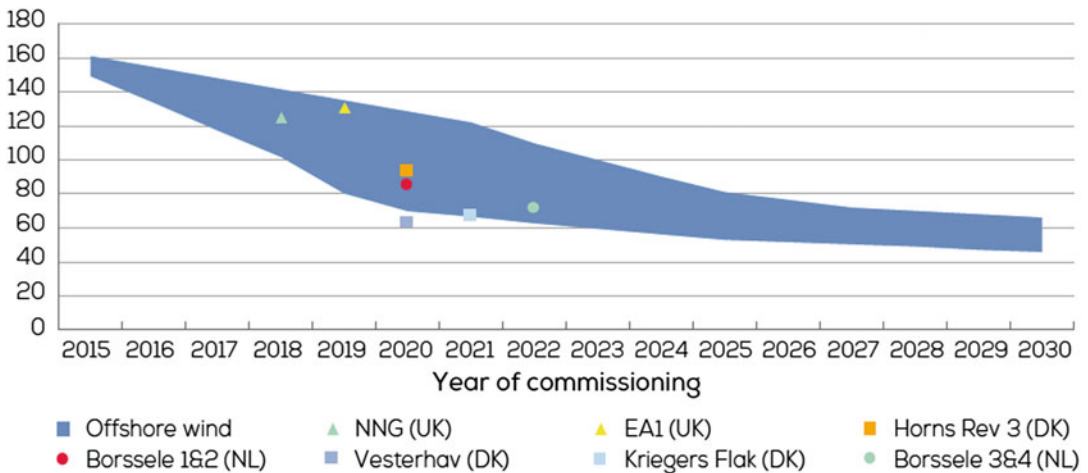
In order to reduce the LCOE for a certain technology, it will be beneficial to look at in detail the cost breakdown in terms of CAPEX and OPEX. In today's offshore wind farms, OPEX is about 10–20% of the total cost, while CAPEX accounts for 80–90%. Figure 3 shows the CAPEX breakdown for different components and their installation for selected European offshore wind farms (MAKE Consulting 2016). Wind turbine itself (including rotor, nacelle, gearbox, and generator) accounts for 30% and its installation 8%, followed by foundation 13%

and its installation 8%, and then power cable and substation. It should be noted that the total cost for the same technology can be very different from project to project. In particular, water depth and distance from shore play a very important role on the cost of individual projects since the cost of foundations and power cables is very much dependent on these two parameters. Moreover, turbine size is continuously increasing, which reduces the installation cost per MW and will also potentially reduce the maintenance cost per MW. On the other hand, the total installation cost, including wind turbine, foundation, and power cable installation, is about 25%, which is expected to reduce since purpose-built installation vessels are being more used.

Unlike the offshore oil and gas platform which is normally one of its kind, offshore wind turbines need to be developed into farms so that the advantage of mass production should be fully used to reduce the cost. Moreover, good logistics for transport and installation operations and good project and finance management will also contribute to cost reduction. Similarly, purpose-built supply vessels are being developed to provide access to offshore wind turbines at higher sea states for maintenance, repair, and replacement to reduce the downtime and the OPEX (Willow and Valpy 2015). Since 2015, there is a positive general trend of cost reduction for offshore wind turbines due to

the improvement and maturation of the offshore wind technology and project management as well as the introduction of large-scale (6–8 MW) wind turbines. In particular, as shown in Fig. 4, the bidders for several offshore wind farms in Europe in 2016, including Borssele 3 and 4 in the Netherlands, Kriegers Flak, and Vesterhav in Denmark, offered quite low bids, from 72 to 60 Euro/MWh. In 2017, Dong Energy and EnBW won the bids to build first subsidy-free offshore wind farms in the North Sea and the Baltic Sea in Germany, respectively (Offshore Wind Industry 2017). A further cost reduction in offshore wind industry is expected, and the LCOE would reach 40–60 Euro/MWh by 2030.

As mentioned, most of the offshore wind turbines today are based on bottom-fixed foundations, and the LCOE discussed above is purely for bottom-fixed (mainly monopile) wind turbines. Floating wind turbines are still in the early stage of development. In addition to the prototypes in Norway, Portugal, Japan, and the USA, Equinor started to operate the world's first floating wind farms of five 6 MW turbines in Scotland since October 2017 based on their Hywind spar technology. The LCOE for this phase has already been reduced significantly by Equinor as compared to that of their 2.3 MW Hywind Demo which was installed in 2009. They also aims for even lower LCOE at 40–60 Euro/MWh by 2030 for large-



Economic Assessment, Fig. 4 Offshore wind LCOE range and trajectory from 2015 to 2030 (Hundleby and Freeman 2017)

scale wind farm development (MontelNews 2018). If this becomes true, the LCOE for floating wind turbines will be comparable to that for bottom-fixed wind turbines.

Cross-References

- [Marine Operations](#)
- [Offshore Wind Turbines](#)
- [Tidal and Ocean Current Turbines](#)
- [Wave Energy Converters](#)

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Economic Barriers

- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

EEDI (Energy Efficiency Design Index)

- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

EIP – Equipment Installation Platform

- [Buffer Station](#)

Elastic Connectors

- [Connectors of VLFS](#)

Electric Cable

- [Umbilical Cable](#)

Empirical Design

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Synonyms

[Concept design](#); [Contract design](#); [Design spiral](#); [Detailed design](#); [Optimal design](#); [Preliminary design](#); [Sequential engineering](#); [Technical design](#)

Definition

Empirical design is a design method generally characterized by the collection of a large amount of data before much speculation as to their significance, or without much idea of what to expect, and is to be contrasted with more theoretical methods in which the collection of empirical data is guided largely by preliminary theoretical exploration of what to expect. The empirical method is necessary in entering hitherto completely unexplored fields and becomes less purely empirical as the acquired mastery of the

field increases. Successful use of an exclusively empirical method demands a higher degree of intuitive ability in the practitioner (from Access Science @McGraw-Hill).

Scientific Fundamentals

What Is Empirical Method?

The word “empirical” means “based only on observation or experiment.” So “empirical evidence” is evidence that is based on observation. And an “empirical question” is one that can be answered through scientific experimentation.

Empirical method means: based on observation or experiment, not on theory. Also implies, results achieved by the trial and error method (www.answers.com).

Strict empiricists are those who derive their rules of practice entirely from experience, to the exclusion of philosophical theory.

This approach is far more successful with simple problems and in games and is often resorted to when no apparent rule applies. This does not mean that the approach need be careless, for an individual can be methodical in manipulating the variables in an attempt to sort through possibilities that may result in success. Nevertheless, this method is often used by people who have little knowledge in the problem area. The trial-and-error approach has been studied from its natural computational point of view (Bei et al. 2013).

What Is Empirical Design Method?

Empirical design method refers to the design method that the researchers determine the appropriate research method and parameters according to the target requirements, conduct repeated tests and comparisons, and find out the targets that meet the requirements. Then the targets will be tested and compared again, until the researcher finds out the most satisfying targets.

The Process of Empirical Design

Through constant testing and error elimination, the designer can explore the system with black box properties by the method. This method is applied unconsciously in the behavior of animals

and consciously in the behavior of humans. Empirical design method is a learning method purely based on experience. By the response of a test black box due to changing the parameters of the black box system intermittently or continuously, the subjects of the applied empirical design seek a way to reach their goal. The success or failure of the subject’s behavior is evaluated by its degree of approaching to the goal or the process of reaching the intermediate goal. When information approaching the goal is given to the subject, the subject will continue to take successful behavior. When information that deviates from the goal is fed back to the subject, the subject will avoid to take unsuccessful behavior. Through this constant trial and evaluation, the subject can gradually achieve the goal to be pursued.

Because the designer does not know where the satisfactory solution lies, before finding the solution or a more satisfactory solution, they have to make a lot of mistakes. The number of mistakes often depends on the designer’s knowledge level and experience. The so-called innovation is the work of a few talents, which is exactly the experience of trial and error.

This kind of method to solve the problem is more likely to be successful when applied to simple problems or games and is often used when there is no other obvious rule that can be used to solve the problem. Trial and error does not mean that the user must try the method unintentionally. The user can systematically control the variables in order to sort out the solution with the best chance to solve the problem successfully.

The Features of Empirical Design

Empirical design has a number of features:

- Solution-oriented: Makes no attempt to discover why a solution works, merely that it is a solution.
- Problem-specific: Makes no attempt to generalize a solution to other problems.
- Nonoptimal: Generally an attempt to find a solution, not all solutions, and not the best solution.
- Needs little knowledge: Can proceed where there is little or no knowledge of the subject.

It is possible to use empirical design (the trial-and-error method) to find all solutions or the best solution, when a testable finite number of possible solutions exist. To find all solutions, one simply makes a note and continues, rather than ending the process, when a solution is found, until all solutions have been tried. To find the best solution, one finds all solutions by the method just described and then comparatively evaluates them based upon some predefined set of criteria, the existence of which is a condition for the possibility of finding a best solution. (Also, when only one solution can exist, as in assembling a jigsaw puzzle, then any solution found is the only solution and so is necessarily the best.)

The Application Example of Empirical Design

Empirical design methods are widely used in various industries, especially in traditional industries such as Navy Architecture and Ocean Engineering, Machinery Manufacturing, etc. In the process of ship design, empirical design methods are needed in many cases, such as in the calculation of hull friction resistance, in the concept design of new ship according to the previous design experience, and so on.

The Calculation of Hull Friction Resistance

Based on the previous calculation experience of hull friction resistance, the researchers have summarized the formula of hull friction resistance coefficient as shown below.

The Schoenheer Formula

In 1932, Schoenheer applied the law of logarithmic velocity distribution and gave the following empirical formula according to the results of the plate drag test.

$$\frac{0.242}{\sqrt{C_f}} = \lg(ReC_f)$$

This is the Schoenheer formula, which is most commonly used in the USA. In 1947, the American Ship Model Test Tank Conference

decided to use this formula as the standard formula to calculate the ship friction resistance, so this formula is also known as the formula 1947 ATTC. Because the formula is very difficult in the actual calculation, therefore, when Re within 10^6 – 10^9 , the formula can be turned into simple formula with the same results.

$$C_f = \frac{0.4631}{(\lg Re)^{2.6}}$$

The Prandtl–Schlichting Formula

By applying the same principle above, Prandtl and Schlichting have obtained the Prandtl–Schlichting formula which is very similar to the Schoenheer formula.

$$C_f = \frac{0.455}{(\lg Re)^{2.58}}$$

This formula is most commonly used in continental Europe and has been widely used in China in the past.

The Hughes Formula

Because the above Schoenheer formula and the Prandtl–Schlichting formula were obtained based on the results of the plate test, geometric similarity was not considered in the analysis. According to the Reynolds theorem in the similarity theory, the theorem that the coefficient of friction resistance is a function of Reynolds number is only applicable to the geometrically similar plates. If the geometric shape of the plate is not similar, that is, the aspect ratio (that is, the width and length ratio) is not equal, then the coefficient of friction resistance should be a function of both Reynolds number and B/L , that is to say,

$$C_f = f\left(Re, \frac{B}{L}\right)$$

The influence of different aspect ratio on the coefficient of friction resistance is called the edge effect.

In 1952, Hughes analyzed a number of previously published plate data and confirmed that the coefficient of friction resistance was related to

the aspect ratio. In 1954, Hughes published his plate test data. In his test, the Reynolds number is between 2×10^4 and 3×10^9 and aspect ratio is between 0.0156 and 42. The formula for a smooth flat plane with an infinite aspect ratio is obtained.

$$C_f = \frac{0.066}{(\lg Re - 2.03)^2}$$

This formula is called the Hughes formula. The calculation result of this formula is smaller than that of the Schoenheer formula, because the Schoenheer formula is derived from the interpolation of the finite aspect ratio (three-dimensional flow) plate data.

The Formula of 1957 ITTC

Although the calculated results according to the existing formulas of smooth plate's friction resistance are very close, there are some differences. In particular, when the ship model test results are converted to the real ship, the calculated resistance of the real ship varies with different formulas. Based on the analysis of the resistance test results of geometrically similar ship model, the Eighth International Ship Model Test Tank Conference (ITTC for short) held in Madrid, Spain, in 1957, believed that the value obtained by the Schoenheer formula at low Reynolds number was low. Finally, the following new formula was proposed. It is called "1957 international model test tank real ship - model conversion formula," abbreviated as "the formula of 1957 ITTC."

$$C_f = \frac{0.075}{(\lg Re - 2)^2}$$

It should be pointed out that the formula of 1957 ITTC is not exactly the formula for calculating the friction resistance coefficient of turbulent smooth plate, and it is used for the resistance conversion between ship model and real ship. China now uses the formula of 1957 ITTC.

According to the empirical calculation method of ship resistance, the frictional resistance coefficient curves calculated by different formulas can be obtained. It can be seen from the curves that, at a low Reynolds number, the slope of the formula

of 1957 ITTC is steeper than other formula's slope, and at high Reynolds number, the result of the formula of 1957 ITTC is not different from the Schoenheer formula's result. Besides, the formula of 1957 ITTC is very similar to the Hughes formula in form, but it is about 12.5% larger in numerical terms than the Hughes formula. While the Prandtl-Schlichting formula is very similar to the Schoenheer formula in form, and it is about 2.0–2.5% larger in numerical terms than the Schoenheer formula at most.

The Concept Design of New Ship

In addition, in order to reduce the wave resistance of ships, designers can adopt different design concepts to design new ships according to the previous design experience. Successful empirical design will lead to the emergence of a new type of ship, which will greatly improve the performance of a ship. Here are some typical empirical design concepts.

Catamaran and Multihull Ship Design Concept

Based on past design experience, researchers have found that catamarans have some unique properties. The catamaran has two pieces, and the friction resistance is larger than that of monohull ship of the same length and displacement. At the same time, the wave interference between two bodies will increase the resistance. However, the length drainage volume coefficient or length-width ratio of each piece of the catamaran is much larger than that of monohull ship, so the wave-making resistance can be significantly reduced. Therefore, in a certain range of speed, the total hull resistance of the catamaran is usually smaller than that of the monohull ship.

Multibody ships, such as three-body ships, not only increase the fineness coefficient or length-width ratio, which can make the wave-making resistance coefficient drop, but also can use the favorable interference between the before-and-aft bodies to further reduce the wave-making resistance.

The Design Concept of Lifting the Hull Out of the Water

Based on previous design experience, the researchers found that it is an effective measure to

reduce wave resistance to slide the hull above the water surface or to leave the water surface. Based on these experiences, the researchers designed planing boat, hydrofoil boat, hovercraft, and so on.

When the planing boat glides in the water, the bottom of the planing boat generates the dynamic lift force to lift most of the boat out of the water, and the drainage volume decreases so as to reduce the wave resistance.

When the hydrofoil boat is in the airfoil condition, the lift force generated by the hydrofoil causes the boat body to be lifted out of the water. Thus, only the hydrofoil and appendage generate resistance in the water. This resistance is relatively small, so the hydrofoil can reach a higher speed.

The hovercraft is a kind of boat that the rising pressure of the air cushion lifts the hull out of the water. The wave-making resistance of the hovercraft decreases as only the apron or side wall appendages produce resistance

The Design Concept of Hull Submergence

It is known from previous design experience and wave theory that the wave-making mainly occurs near free surface, and the wave-making amplitude will decrease exponentially with the increase of immersion depth. Therefore, the design concept of hull submergence is to move part or all of the displacement volume of the hull below the water surface, so as to reduce wave resistance. Based on their design experience, the researchers designed semi-submersibles and diving boats.

Selection and Design of Submarine Structure Materials

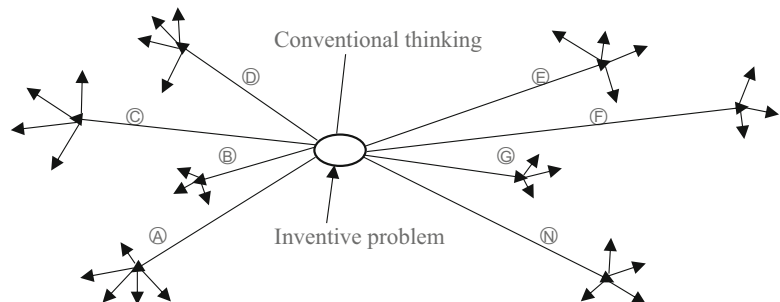
There are many kinds of materials on the submersible, which can be divided into pressure-resistant

structure materials and nonpressure-resistant structure materials. Generally, for the selection of pressure-resistant structure materials, the following methods should be followed.

Firstly, according to the function requirements of material design (usage requirements, shape requirements, pressure depth, etc.), design personnel usually select the suitable raw materials, determine the appropriate material processing methods and the process parameters, conduct repeated tests and comparisons, and find out the applicable material whose basic performance meets the requirements. Then the material will be tested and compared again, until the designers find out the most satisfying material. The process is a kind of typical experience design method.

Secondly, the trial-and-error method refers to the method that the designers put forward the working principle of new product according to the existing product or previous design experience and then make sample part through continuous modification and improvement. If the sample part does not meet the requirements, the designer will return to the plan design and start again. Until the sample design is proved to meet the requirements, the sample part can be manufactured in small batches and in batches. As shown in the following Fig. 1, the designers look for solutions along direction A according to experience or existing products. If there is no appropriate solution, they will adjust the direction and search along direction B. If still not, transfer direction to C, D, and so on. The designers did not complete the task until a satisfactory “solution” is encountered in one direction. This is the most primitive way of seeking new things and the first method of technological creation in history.

Empirical Design,
Fig. 1 The schematic diagram of the trial-and-error method



Cross-References

- Concept Design
- Design of Submersibles
- Design Spiral
- Detailed Design
- Hydrodynamic Design
- Multidisciplinary Design Optimization (MDO)
- Optimal Design
- Preliminary Design
- Reliability-Based Design (RBD)
- Structural Design
- Technical Design

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Energy Consumption Management

- Human Factors in the Role of Energy Efficiency

Energy Efficiency

- Energy Efficiency Regulations from IMO
- Human Factors in the Role of Energy Efficiency
- Impact of Maritime Transport
- Ship Construction and Operation Impact on Energy Efficiency
- Ship Propulsion System
- Underwater Acoustic Sensor Network

Energy Efficiency Design Index (EEDI)

- Energy Efficiency Regulations from IMO

Energy Efficiency Factor

- Energy Efficiency Regulations from IMO

Energy Efficiency Regulations from IMO

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Synonyms

Energy efficiency; Energy efficiency design index (EEDI); Energy efficiency factor; Energy saving technology; IMO (International Maritime Organization)

Introduction

The global warming is caused by greenhouse gases which are accumulating solar heat in the atmosphere by keeping the long wave radiation energy from the earth effectively. Since the latter half of the twentieth century, global warming is concerned by the whole world gradually. In 1992, the United Nations Framework Convention on Climate Change (UNFCCC) which was developed by the United Nations Climate Change Committee was endorsed by 154 countries in the United Nations Conference on Environment and Development in Rio de Janeiro, and came into force on March 21, 1994; this is the first international convention that withstands the global warming by controlling the emission of greenhouse gases overall. Meanwhile, in Tokyo, 1997, for the purpose of

propelling the emission of greenhouse gases, the Kyoto Protocol, which forced the emission of greenhouse gases in developed countries and set standards of emission, was adopted in the Third Conference of the Contracting Governments on Climate Framework Convention and came into force in February 2005. In the same year, the Solution for CO₂ Emission of Ships was adopted by the Contracting Governments based on the International Convention for the Prevention of Pollution from Ships 1973 (MARPOL73/78 Convention), and the Marine Environment Protection Committee (MEPC) was authorized to consider the workable reduction strategies of greenhouse gases.

The Considering Procedure of MEPC for EEDI

The considerations of MEPC for energy efficiency design index (EEDI) are summarized as three stages.

The Stage of Extensive Discussion and Argument, MEPC56–MEPC59

In July 2007, the draft resolution of setting the assessment panel of ship emission reduction was adopted at 56th session of MEPC, since then, the panel including 23 members was to carry out the research of ship emission reduction.

In March 2008, the issues of mandatory CO₂ design index of new ships, guidelines for voluntary and temporary trial of greenhouse gases emission index, and levying marine fuel oil tax worldwide, etc., were discussed at 57th session of MEPC. At the ship owner's request, the mandatory CO₂ Design Index of New Ships was decided to be developed to consider design standards and operational measures and market-based measures separately. During the session, a working group meeting on CO₂ Emission Index Mechanism was added, and the amendments to MARPOL Annex VI of the regulations on harmful gas reduction for ships were adopted.

In October 2008, a Draft of Interim Guidelines on the Method of Calculation of EEDI for New Ships was developed at 58th session of MEPC by changing CO₂ Design Index of New Ships into

EEDI, in consideration of that International Maritime Organization (IMO) were to put forward the standards of new ship design and construction from the point of improving energy efficiency level, other than emphasizing the requirement of greenhouse gases (GHG) reduction.

In March 2009, the EEDI calculation methods were adjusted at the second working group meeting during session.

In July 2009, the Interim Guidelines on the Method of Calculation of EEDI for New Ships were further modified at 59th session of MEPC, and the Interim Guidelines on the Method of Calculation of EEDI and the Interim Guidelines on Voluntary Verification of EEDI were published by Circulars. The guidelines, which were used voluntarily at that stage, encouraged ship owners and designers to design and construct ship with higher energy efficiency standards by technical innovation and energy-saving technology.

The Stage of Development of Operation Measures and Technical Measures

The Regulations on Energy Efficiency for Ships were adopted and added in MARPOL Annex VI at 60–62nd session of MEPC.

In December 2009, the Draft of Amendments for MARPOL Annex VI in which the EEDI was added at some countries' request was developed at 60th session of MEPC; meanwhile a further research on confirmation of base line based on the existing ship data was agreed.

In September 2010, the issue of GHG emission reduction was discussed at 61st session of MEPC; no agreement on how to enforce the energy efficiency management of ships as a part of MARPOL more reasonably was reached. The enforcement of EEDI was delayed.

In July 2011, the EEDI and the Ship Energy Efficiency Management Plan (SEEMP), as the lasting of MARPOL Annex VI, were adopted at 62nd session of MEPC, and came into force on January 1, 2013. In this session, EEDI calculation guidelines, survey and certification guidelines, SEEMP development guidelines, and interim guidelines for minimum installed power and ship speed in adverse conditions were completed. As a result, shipping industry became the first industry

that implemented the CO₂ emission reduction measures worldwide.

The Stage of Measures' Discussion and Preparation of Market Emission Reduction Mechanism

During the stage of issuance and implementation of energy efficiency rules since the 63rd session of MEPC, the implementation problems are being modified and improved.

In February 2012, a series of significant guidelines of enforcing energy efficiency rules were discussed and adopted at 63rd session of MEPC, included:

- Guidelines on the Method of Calculation of the Attained Energy Efficiency Design Index (Attained EEDI) for New Ships
- Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP)
- Guidelines on Survey and Certification of the Energy Efficiency Design Index (EEDI)
- Guidelines for Calculation of Reference Lines for Use with the Energy Efficiency Design Index (EEDI)

In October 2012, the relevant technologies and operation standards on energy efficiency management were developed at the 64th session of MEPC, and the following were adopted:

- A new Chapter 4 of MARPOL Annex VI will enter into force on January 1, 2013, including EEDI of new ships and SEEMP of all ships.
- Amendments to the 2012 Guidelines on the Method of Calculation of the Attained Energy Efficiency Design Index (EEDI) for New Ships (Resolution MEPC.212(63)), including the calculation of shaft generator power and shaft motor power.
- Amendments to the 2012 Guidelines on Survey and Certification of the Energy Efficiency.

In May 2013, the applicability issues of EEDI and SEEMP for different types of ships were discussed at the 65th session of MEPC, and the following resolutions were adopted.

- The session agreed that the ships without means of propulsion including platforms, drilling rigs and barges, etc., were not to meet the energy efficiency requirements.
- The application of EEDI for ships having ice-breaking capability.
- The EEDI requirements may be omitted for the cargo ships having ice-breaking capability based on the high installed power.
- The Draft 2013 Interim Guidelines for Determining Minimum Propulsion Power to Maintain the Maneuverability of Ships in Adverse Conditions.
- These interim guidelines are applied to bulk carriers, tankers, and combination carriers with the deadweight of equal or more than 20,000 DWT and conventional mode of propulsion during Phase 0. The minimum power guidelines were considered and approved.
- The 2013 Guidance on Treatment of Innovative Energy Efficiency Technologies were considered and approved.
- Guidelines for calculation of f_w , which is a nondimensional coefficient indicating the decrease of speed in representative sea conditions of wave height, wave frequency and wind speed (e.g., Beaufort Scale 6).
- The calculation methods and requirements of EEDI for ro-ro vehicle ships, ro-ro cargo ships, and ro-ro passenger ships.
- The development of relevant EEDI requirements for cruise passenger ships with non-conventional propulsion system.
- The calculation of EEDI for liquefied natural gas (LNG) carriers.
- The application of correction factors for general cargo ships.
- The application suggestion of assignments of three correction factors including f_j (correction factor to account for ship-specific elements)/ f_{IVSE} (structural enhancement correction factor)/ f_I (correction factor for ships equipped with cranes and other cargo-related gears) for general cargo ships, and the draft amendments to EEDI calculation guidelines.
- The sea trial methods and tank tests.
- The 2012 Guidelines on Survey and Certification of the Energy Efficiency Design Index

(EEDI) (Resolution MEPC.214(63)) were approved to be revised by specifying ITTC 7.5-04-01-01.1 part 1 and part 2 instead of ISO 15016:2002.

At the 66th session of MEPC in March 2014, the following resolutions were adopted.

1. Amendments to the 2012 Guidelines on the Method of Calculation of EEDI

- The use of Carbon Conversion Factor C_F for dual-fuel engines.
- The power weighting methods are not to apply for C_F but multiplying C_F -Gas and C_F -fuel oil by consumption of each fuel respectively at the EEDI load point.
- LNG carriers.
- The electrical efficiency should be taken as 91.3% for LNG carriers; alternatively, if the value more than 91.3% is to be applied, the method of ship measurement is to be adopted.
- The SFC of diesel-electrical propulsion systems should be taken as the value at 75% of MCR power.
- The methane leakage of LNG carriers is not to be taken into account in EEDI calculation, although the greenhouse gases emission caused.
- The application of correction factor f_j to account for ship-specific elements.
- The suggestion of correction factor f_j of EEDI calculation for refrigerated cargo carriers with ice class notations and that of editing revise of correction factor f_j of EEDI calculation for general cargo ships were added to the amendments of guidelines.
- The EEDI calculation of passenger ships with convention propulsion systems.
- The Attained EEDI of passenger ships with convention propulsion systems is to be calculated in accordance with the requirements in MARPOL Annex VI and the existing EEDI calculation guidelines.

The 2014 Guidelines on the Method of Calculation of EEDI were adopted at MEPC 66.

2. The amendments of Guidelines on Survey and Certification of EEDI

The 2014 Guidelines on Survey and Certification of EEDI were not adopted at this session, and would be considered at MEPC 67.

3. The amendments of MEPC.1/Circ.796 (Interim Guidelines on Calculation of f_w factor)

The progress of Interim Guidelines on Calculation of f_w factor developed by China and Japan was considered; further results were expected to be submitted and be added to guidelines on calculation of f_w .

4. The establishment of EEDI database

The proposal of establishing EEDI database was considered at this session; the goal and basic principles including security, publicity, and administrations of this database were confirmed; meanwhile, the minimum data parameters table for data collection, which based on the International Association of Classification Societies (IACS) version, was developed by the group.

At the 67th session of MEPC in October 2014, the relevant issues and resolutions of EEDI were as follows:

- The amendments suggestion of the Guidelines on Calculation of EEDI with dual-fuel engines.
- The calculation concept of EEDI with dual-fuel engines proposed by China was approved to a further consideration.
- The amendments of Guidelines on Survey and Certification of EEDI.
- The Amendments to Guidelines on Survey and Certification of EEDI including the main fuel definition of dual-fuel ships proposed by China were adopted.
- The guidelines for minimum power.
- The application duration of the 2013 Guidelines for Minimum Power in Adverse Conditions was approved to be extended to Phase 1.
- The communication group was approved to be set up for the technical consideration of EEDI Reference Line Value (RLV) during Phase 2 in MARPOL Annex VI.

- The basic content of revised data collection system was approved; the communication group was set up again to develop the full copy for data collection system of fuel consumption.

At the 68th session of MEPC in May 2015, the relevant issues and resolutions of EEDI were as follows:

- The data collection mechanism of fuel oil consumption for ships in operation.
- The consideration of EEDI data and EEDI in Article 21.6 of MARPOL Annex VI.
- The identified numbers of ships, which were only used by IMO secretariat, were approved to be added to EEDI database.
- The communication group was approved to be set up for further consideration of other data, such as the relevant information of innovative technology, main engine power, dimension parameters, ice class, etc. The results were to be taken into account for the setting and implement time of EEDI standards which were to be discussed at MEPC 69.
- The amendments of 2013 Interim Guidelines on the Minimum Installed Propulsion Power.
- The existing Interim Guidelines on the Minimum Installed Propulsion Power were amended, in which the requirements of level 1 for bulk carriers and oil tankers were enhanced.
- The amendments of 2014 Guidelines on Survey and Certification of EEDI (MEPC.254(67)) were adopted; the ISO 1506:2015 was added and approved to apply to the ships carrying out sea trial on or after September 1, 2015.
- The Amendments to 2014 Guidelines on Calculation of EEDI.
- Some specific ship types were approved to be further considered upon the suggestion of remaining the EEDI RLV during Phase 2 standards proposed by communication group.
- The consideration of EEDI technologies were to be continued based on the relevant data including innovative energy-saving technologies and dimension parameters, such as the names, working principles and performance of innovative energy-saving technologies, the length, breadth, and depth/draught of ships, or proportion of dimensions (L/B, L/D, etc.).
- The consideration of minimum propulsion power was approved to be separate from that of EEDI technologies.
- The consideration of recommendations to calculation method of the correction factor for ice-classed ships was delayed to MEPC 70,
- The method of calculation of ships with dual-fuel engines.
- The suggestion of EEDI calculation method of ships with dual-fuel engines proposed by China was approved generally, and China was invited to submit improved proposals to the next session by connecting with other interested countries and organizations.
- The Guidelines on Survey and Certification of EEDI concerning ISO 15016:2015.
- The regulations of water temperature measurement were contained in ISO 15016:2015, and the load variation factors for tank tests were considering by ITTC, there was no need to consider.
- The exemption conditions of SEEMP requirements.
- The MEPC circular, which contained the exemption suggestion of energy efficiency requirements for ships engaged on domestic voyages when engaged on international voyages once only, was approved.

The draft Amendments to Article 2.6 in 2014 Guidelines on Calculation of EEDI (MEPC.245(66)), which were proposed by organizations including IACS, were considered and approved. At the 69th session of MEPC in April 2016, the following resolutions were adopted:

At the 70th session of MEPC in October 2016, the Amendments to Guidelines of EEDI Calculation were developed.

- The consideration of EEDI RLV during Phase 2 standards.

- The requirements including reference lines, reduction factors, etc., of ship types other than ro-ro ships during Phase 2 were decided to maintain.
- The amendments to Guidelines on Calculation of EEDI.
- The amendments in which the EEDI was to be calculated with gas fuel as the nonessential fuel were adopted.
- The proposal of adding cubic capacity correction factors to EEDI calculation guidelines for wood cargo vessels was approved.
- The consideration of adding relevant information to IMO EEDI database.
- At this session, it was agreed that more IMO EEDI data were to be submitted including the main dimension, ship speed, installed power, and relevant summarization for application of energy-saving technologies, and this requirement was to come into force on April 1, 2017. For the data submitted, there was no need to supplement.
- The amendment of SEEMP Guidelines.

A Part II, as a guidance to develop data collection plans for fuel oil consumption, was added to the SEEMP Guidelines based on the Amendments to MARPOL Annex VI.

The resolution of Guidelines for the Development of SEEMP was issued.

At the 71st session of MEPC in July 2017, the following resolutions were adopted.

- The consideration of EEDI requirements for ro-ro passenger ships and ro-ro cargo ships.
- The amendments to Article 21 in MARPOL Annex VI were approved, in which the parameter value of reference lines for ro-ro passenger ships and ro-ro cargo ships was adjusted, in particular, the reference lines for ro-ro passenger ships with the deadweight more than 10,000 DWT and ro-ro cargo ships with that more than 17,000 DWT were set to be a constant value. The corresponding amendments to MARPOL Annex VI were to be adopted at MEPC 72.
- EEDI database: the suggestion of the definitions of relevant parameters, data rounding,

etc., proposed by secretariat was approved, so was the template of data submission. Meanwhile, in order to add EEDI data to MARPOL module of GISIS platform, it was agreed that the relevant information was to be reported to follow-up committee.

- The amendments to Interim Guidelines on the Minimum Installed Propulsion Power

Discussed by the Committee, the existing Guidelines on the Minimum Installed Propulsion Power, were approved to be extended to EEDI Phase 2. At the 72nd session of MEPC in April 2018, the following resolutions were adopted.

- The Amendments to EEDI requirements for ro-ro passenger and ro-ro cargo ships: The amendments to reference lines of ro-ro passenger ships and ro-ro cargo ships were approved and were to be entered into force on September 1, 2019. The segmented reference lines for ro-ro passenger ships with the deadweight more than 10,000 DWT and ro-ro cargo ships with that more than 17,000 DWT were to be developed for use from Phase 2 (2020.1.1). Meanwhile, these amendments could be put into effect earlier by Administrations of every country.
- The EEDI requirements for ice-classed ships: The amendments to Article 19.3 of MARPOL Annex VI were to be pointed to the Polar Code, the original clause “The EEDI requirements may be omitted for the ice-classed ships with an ice class of equal or higher than IACS PC5 (or equal ice class)” was to be revised as “The Article 20 and 21 are not applicable to the ships of A category in the Polar Code.” Meanwhile, the definition of cargo ships having ice-breaking capability in Annex VI was suggested to be deleted, and that of Polar Code to be added.

Calculation Method of EEDI

Calculation Formula

The Attained EEDI is calculated by the following formula:

$$EEDI_a = \left(\prod_{j=1}^n f_j \right) \left(\sum_{i=1}^{n_{ME}} P_{ME(i)} \cdot C_{FME(i)} \cdot SFC_{ME(i)} \right) + (P_{AE} \cdot C_{FAE} \cdot SFC_{AE})$$

$$+ \left(\left(\prod_{j=1}^n f_j \cdot \sum_{i=1}^{n_{PTI}} P_{PTI(i)} - \sum_{i=1}^{n_{eff}} f_{eff(i)} \cdot P_{AEeff(i)} \right) C_{FAE} \cdot SFC_{AE} \right) - \left(\sum_{i=1}^{n_{eff}} f_{eff(i)} \cdot P_{eff(i)} \cdot C_{FME} \cdot SFC_{ME} \right)$$

$$\frac{f_i \cdot f_c \cdot f_w \cdot W \cdot V_{ref}}{(1)}$$

where:

$EEDI_a$ is the attained EEDI value, measured in g/(t·nm).

n is the number of power correction factors.

f_j is the power correction factor.

n_{ME} is the number of main engines.

$P_{ME(i)}$ is the power of each main engine, measured in kW.

$C_{FME(i)}$ is the carbon conversion factor of fuel for each main engine, selected according to Table 1.

$SFC_{ME(i)}$ is the specific fuel consumption of each main engine, measured in g/kW h.

P_{AE} is the power of auxiliary engines, measured in kW.

C_{FAE} is the carbon conversion factor of fuel for auxiliary engines according to Table 1.

SFC_{AE} is the specific fuel consumption of auxiliary engines, measured in g/kW h.

n_{PTI} is the number of shaft motors.

$P_{PTI(i)}$ is the power of each shaft motor, measured in kW.

n_{eff} is the number of new energies and new technologies adopted.

$f_{eff(i)}$ is the energy efficiency factor of each innovative energy-efficient technology.

$P_{AE\ eff(i)}$ is the auxiliary power reduction due to innovative electrical energy-efficient technology measured at P_{ME} , measured in kW.

$P_{eff(i)}$ is the output of the innovative mechanical energy-efficient technology for propulsion at 75% main engine power, measured in kW.

C_{FME} is the carbon conversion factor of fuel for main engines.

SFC_{ME} is the specific fuel consumption of main engines, measured in g/kW h.

f_i is the capacity correction factor.

f_c is the cubic capacity correction factor.

f_w is the speed loss coefficient.

W is the capacity.

V_{ref} is the ship speed, measured in knot. Note 1: If part of the normal maximum sea load is provided by shaft generators, for that part of the power, SFC_{ME} and C_{FME} may be used instead of SFC_{AE} and C_{FAE} . Note 2: If $P_{PTI(i)} > 0$, the weighted average value of $(SFC_{ME} \cdot C_{FME})$ and $(SFC_{AE} \cdot C_{FAE})$ is to be used for calculation of P_{eff} .

Energy Efficiency Regulations from IMO, Table 1 Carbon conversion factor

No.	Type of fuel	Reference	Carbon content	C_F
1	Diesel/gas oil	ISO 8217 grades DMX through DMC	0.8744	3.206
2	Light fuel oil (LFO)	ISO 8217 grades RMA through RDM	0.8594	3.151
3	Heavy fuel oil (HFO)	ISO 8217 grades RME through RMK	0.8493	3.114
4	Liquefied petroleum gas (LPG)	Propane	0.8182	3.000
		Butane	0.8264	3.030
5	Liquefied natural gas (LNG)	—	0.7500	2.750
6	Methanol	—	0.3750	1.375
7	Ethanol	—	0.5217	1.913

The C_F value for different type of fuel is as follows:

Ship Speed (V_{ref})

V_{ref} is the ship speed on deep water in the condition corresponding to the capacity as defined in paragraph 2.2.3 (in case of passenger ships and cruise passenger ships, this condition is to be summer load draught as provided in paragraph 2.2.4) at the power of the main engine(s) as defined in paragraph 2.2.5.1 and assuming the weather is calm with no wind and no waves.

Capacity

Capacity is defined as follows for different ship types:

- For bulk carriers, tankers, gas carriers, LNG carriers, ro-ro cargo ships (vehicle carriers), ro-ro cargo ships, ro-ro passenger ships, refrigerated cargo carriers, general cargo ships, combination carriers and offshore supply vessels, deadweight (DWT) is to be used as capacity.
- For passenger ships and cruise passenger ships, gross tonnage (GT) in accordance with the International Convention of Tonnage Measurement of Ships 1969, annex I, regulation 3, is to be used as capacity.
- For containerships, 70% of the deadweight (DWT) is to be used as capacity. EEDI values for containerships are calculated as follows: Attained EEDI value is to be calculated using 70% DWT in accordance with EEDI formula; required EEDI value is to be calculated using 100% DWT in accordance with reference line formula.

Deadweight (DWT)

Deadweight means the difference in tonnes between the displacement of a ship in water of relative density of 1025 kg/m³ at the summer load draught and the lightweight of the ship.

Power of the Main Engines (P_{ME})

P_{ME} is 75% of the maximum continuous rating (MCR) for each main engine. The MCR value specified on the Engine International Air

Pollution Prevention Certificate (EIAPP) is to be used for calculation. If the main engines are not required to have an EIAPP certificate, the MCR value on the nameplate is to be used for calculation. For LNG carriers having diesel electric propulsion system, P_{ME} is to be calculated by the following formula:

$$P_{ME(i)} = 0.83 \times \frac{MPP_{Motor(i)}}{\eta_{(i)}} \quad (2)$$

where $MPP_{Motor(i)}$ is the rated output of motor specified in the certified document, $\eta_{(i)}$ is to be taken as the product of electrical efficiency of generator, transformer, converter, and motor, taking into consideration the weighted average as necessary.

In case where a shaft generator is installed, if the maximum allowable deduction for the calculation of $P_{ME(i)}$ is not more than P_{AE} as defined in paragraph 2.2.5.4, $P_{ME(i)}$ is to be calculated by the following formula, and if the power of engine is higher than the maximum design power of propulsion system, the value of $P_{ME(i)}$ is to be 75% of that maximum design power of propulsion system.

$$P_{ME(i)} = 0.75 \times (MCR_{ME(i)} - P_{PTO(i)}) \quad (3)$$

Fig. 1 gives guidance for determination of $P_{ME(i)}$.

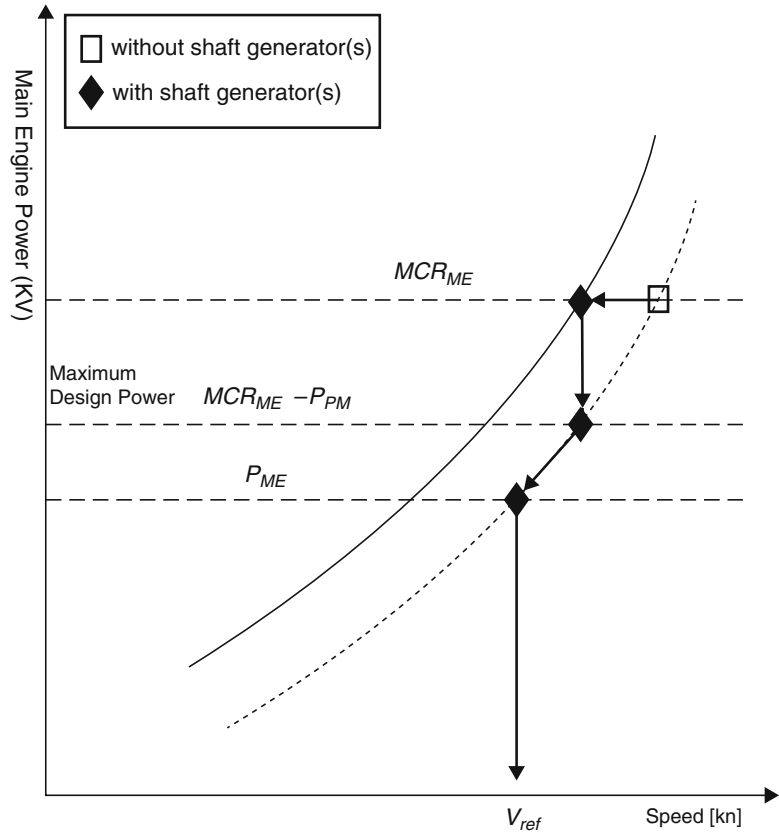
In case where shaft motor(s) are installed, P_{ME} is to be calculated by the following formula at which V_{ref} is measured:

$$P_{ME} = \sum P_{ME(i)} + \sum P_{PTI(i),shaft} \quad (4)$$

$$\sum P_{PTI(i),shaft} = \sum (P_{PTI(i)} \cdot \eta_{PTI(i)}) \cdot \eta_{gen}$$

where $P_{PTI(i),shaft}$ is the output of each shaft motor, measured in kW; $\eta_{PTI(i)}$ is the efficiency of each shaft motor; η_{gen} is the weighted average efficiency of the generators; P_{ME} is higher than 75% of the maximum design power of propulsion system, then 75% of the maximum design power of propulsion system is to be used as P_{ME} .

Energy Efficiency Regulations from IMO,
Fig. 1 Determination of the power P_{ME} of a main engine



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Shaft Generator Power P_{PTO}

In case where a shaft generator is installed, the shaft generator power $P_{PTO(i)}$ is 75% of the rated electrical power output for each shaft generator. In case where shaft motor(s) are installed, $P_{PTI(i)}$ is 75% of the rated power consumption of each shaft motor divided by the weighted average efficiency of the generator(s). In case of combined PTI/PTO , the normal operational mode at sea will determine which of these is to be used in the EEDI calculation. For example, if this combined system is used as a shaft generator for ships in normal operation at sea, the P_{PTO} parameter is to be used in the EEDI calculation formula and P_{PTI} equals 0. The shaft motor's chain efficiency may be taken into consideration instead of the weighted average efficiency of the generator to account for the energy losses in the equipment from the switchboard to the shaft motor, if the chain efficiency of the shaft motor is given in a verified document.

Auxiliary Engine Power P_{AE}

P_{AE} is the required auxiliary engine power to supply normal maximum sea load including necessary power for propulsion machinery/systems and accommodation, for example, main engine pumps, navigational systems and equipment, and living on board, but excluding the power not for propulsion machinery/systems, for example, thrusters, cargo pumps, cargo gear, ballast pumps, maintaining cargo, for example, reefers and cargo hold fans, in the condition where the ship engaged in voyage at the speed (V_{ref}) under the maximum design load condition (Capacity).

P_{AE} used for the calculation of Attained EEDI of ships is to be calculated by the following experience-based formulae:

- For ships with a main engine power of 10,000 kW or above, P_{AE} is defined as:

$$P_{AE} = \left[0.025 \times \left(\sum_{i=1}^{n_{ME}} MCR_{MEi} + \frac{\sum_{i=1}^{n_{PTI}} P_{PTI(i)}}{0.75} \right) \right] + 250 \quad (5)$$

(b) For ships with a main engine power below 10,000 kW, P_{AE} is defined as:

$$P_{AE} = 0.05 \times \left(\sum_{i=1}^{n_{ME}} MCR_{MEi} + \frac{\sum_{i=1}^{n_{PTI}} P_{PTI(i)}}{0.75} \right) \quad (6)$$

(c) For passenger ships and cruise passenger ships, where the P_{AE} value calculated by above formulae is significantly different from the total power used at the speed V_{ref} , the P_{AE} value is to be estimated by the consumed electric power (excluding propulsion) in conditions when the ship is engaged in a voyage at the reference speed (V_{ref}) as given in the electric power table, divided by the average efficiency of the generator(s) weighted by power.

Propulsion power reduced by using the innovative technology $P_{eff(i)}$, which is the output power of the innovative mechanical energy-efficient technology for propulsion at 75% main engine power. In case of a ship equipped with a dual-fuel engine or a number of engines, the C_{FME} and SFC_{ME} are to be the power weighted average value of all the main engines.

Auxiliary power reduced due to innovative technology $P_{AE\ eff(i)}$, which is the auxiliary power reduced due to the adoption of innovative electrical energy-efficient technology when the ship is in the $P_{ME(i)}$ state.

Specific Fuel Consumption (SFC)

The certified specific fuel consumption (SFC) of the engines is defined as follows:

- For engines certified to the E2 or E3 duty cycles, the engine SFC ($SFC_{ME(i)}$) is that recorded in the test report included in a NO_x technical file for the engine(s) at 75% of MCR power or its torque rating.
- For engines certified to the D2 or C1 duty cycles, the engine SFC ($SFC_{AE(i)}$) is that recorded in the test report included in a NO_x technical file at the engine(s) at 50% of MCR power or its torque rating.
- For conventional passenger ships and cruise passenger ships, where the P_{AE} value calculated by the above procedures is significantly different from the total power used at normal seagoing, the SFC (SFC_{AE}) of the auxiliary generators is that recorded in the test report included in a NO_x technical file for the engine (s) at 75% of MCR power or its torque rating. For those engines that do not have an EIAPP certificate because its power is below 130 kW, the SFC specified by the manufacturer and endorsed by the Administration or China Classification Society (CCS) is to be used.

Correction Factor

Power correction factor f_j : f_j is a correction factor to account for ship-specific design elements. For ice-classed ships, this factor is to be taken as the greater value of f_{j0} and $f_{j,min}$ as tabulated in Table 2, but not to be greater than 1.0.

The factor f_j for shuttle tankers with propulsion redundancy and having a deadweight of 80,000–160,000 tonnes is to be $f_j = 0.77$. The shuttle tankers with propulsion redundancy are tankers used for loading of crude oil from offshore installations and equipped with dual-engine and twin-propellers need to meet the requirements for dynamic positioning and redundancy propulsion class notation.

Capacity correction factor f_i : f_i is the capacity correction factor to account for any technical/regulatory limitation on capacity, and can be assumed 1.0 if no necessity of the factor is granted. For ice-classed ships, this factor is to be taken as the lesser value of f_{i0} and $f_{i,max}$ as tabulated in Table 3, but not to be less than 1.0.

For ships with voluntary structural enhancements, f_{VSE} is to be expressed as follows:

Energy Efficiency Regulations from IMO, Table 2 Correction factor f_j for ice-classed ships

Ship type	f_{j0}	$f_{j,min}$ depending on ice class			
		IC	IB	IA	IA Super
Tanker	$\frac{0.308L_{pp}^{1.920}}{\sum_{i=1}^{nME} P_{ME(i)}}$	$0.70L_{pp}^{0.06}$	$0.45L_{pp}^{0.13}$	$0.27L_{pp}^{0.21}$	$0.15L_{pp}^{0.30}$
Bulk carrier	$\frac{0.639L_{pp}^{1.754}}{\sum_{i=1}^{nME} P_{ME(i)}}$	$0.87L_{pp}^{0.02}$	$0.73L_{pp}^{0.04}$	$0.58L_{pp}^{0.07}$	$0.47L_{pp}^{0.09}$
General cargo ship	$\frac{0.0227L_{pp}^{2.48}}{\sum_{i=1}^{nME} P_{ME(i)}}$	$0.67L_{pp}^{0.07}$	$0.56L_{pp}^{0.09}$	$0.43L_{pp}^{0.12}$	$0.31L_{pp}^{0.16}$

Note 1: L_{pp} is the length between perpendiculars

Note 2: B1*, B1, B2 and B3 are ice class notations specified in CCS Rules for Classification of Sea-going Steel Ships, which correspond to IA Super, IA, IB, IC as specified in Finish and Swedish Ice Class Rules (FSICR)

E

Energy Efficiency Regulations from IMO, Table 3 Correction factor f_i for ice-classed ships

Ship type	f_{i0}	$f_{i,max}$ depending on ice class			
		IC	IB	IA	IA Super
Tanker	$\frac{0.00138L_{pp}^{3.331}}{capacity}$	$1.27L_{pp}^{-0.04}$	$1.47L_{pp}^{-0.06}$	$1.71L_{pp}^{-0.08}$	$2.10L_{pp}^{-0.11}$
Bulk carrier	$\frac{0.00403L_{pp}^{3.123}}{capacity}$	$1.31L_{pp}^{-0.05}$	$1.54L_{pp}^{-0.07}$	$1.80L_{pp}^{-0.09}$	$2.10L_{pp}^{-0.11}$
General cargo ship	$\frac{0.0377L_{pp}^{2.625}}{capacity}$	$1.28L_{pp}^{-0.04}$	$1.51L_{pp}^{-0.06}$	$1.77L_{pp}^{-0.08}$	$2.18L_{pp}^{-0.11}$
Container ship	$\frac{0.1033L_{pp}^{2.329}}{capacity}$	$1.27L_{pp}^{-0.04}$	$1.47L_{pp}^{-0.06}$	$1.71L_{pp}^{-0.08}$	$2.10L_{pp}^{-0.11}$
Gas carrier	$\frac{0.0474L_{pp}^{2.590}}{capacity}$	$1.25L_{pp}^{-0.04}$	$1.60L_{pp}^{-0.08}$	$2.10L_{pp}^{-0.12}$	1.25

$$f_{iVSE} = \frac{DWT_{reference\ design}}{DWT_{enhanced\ design}}$$

$$= \frac{\Delta_{ship} - lightweight_{reference\ design}}{\Delta_{ship} - lightweight_{enhanced\ design}} \quad (7)$$

where f_{iVSE} is the capacity correction factor of ships with voluntary structural enhancements; $DWT_{reference\ design}$ is the deadweight prior to application of the structural enhancements, measured in t; $DWT_{enhanced\ design}$ is the deadweight following the application of voluntary structural enhancements, measured in t; Δ_{ship} is the displacement of a ship, measured in t, for this calculation, the same displacement (Δ) is to be taken for reference and enhanced designs; $lightweight_{reference\ design}$ is the lightweight of a ship prior to application of the structural enhancements, measured in t; $lightweight_{enhanced\ design}$ is the lightweight of a ship following the application of voluntary structural enhancements, measured

in t. For this calculation, a change of material (e.g., from aluminum alloy to steel) or a change in grade of the same material (e.g., in steel types, grades, properties and conditions) between reference design and voluntarily enhanced design is not to be allowed for the f_{iVSE} calculation.

For bulk carriers and oil tankers that are constructed according to Common Structural Rules (CSR) and assigned the CSR notation, the following capacity correction factor f_{iCSR} is to be used:

$$f_{iCSR} = 1 + \left(0.08 \times \frac{LWT_{CSR}}{DWT_{CSR}} \right) \quad (8)$$

where f_{iCSR} is the correction factor to account for any technical/regulatory limitation on capacity for ships which are constructed according to CSR; LWT_{CSR} is the lightweight of ships which are constructed according to CSR, measured in t; DWT_{CSR} is the deadweight of ships which are

constructed according to CSR, measured in t. For other ship types not included in the Table 3, f_i is to be taken as 1.0.

Cubic Capacity Correction Factor f_c

f_c is to be taken as 1.0 generally. For chemical tankers, the cubic capacity correction factor f_c is to be taken as the following formula for $R < 0.98$, or $f_c = 1.00$ for $R \geq 0.98$. For gas carriers that are constructed or adapted to carry liquefied natural gas in bulk and with propulsion systems directly driven by diesel engines, the f_c is to be:

$$f_c = R^{(-0.7)} - 0.014, f_c = R^{-0.56} \quad (9)$$

where R is the ratio of the ship's DWT (in tonnes) to the total cubic capacity (in m^3) of its cargo tanks.

Speed Loss Coefficient f_w

f_w is to be taken as 1.0 generally. Where the Owner requests on a voluntary basis the application of f_w , the Attained EEDI value using f_w is to be referred to Attained EEDI_{weather} and confirmed by the Administration, and indicated in the related certificate. f_w is to be determined as follows:

- f_w can be determined by conducting the ship-specific simulation of its performance at representative sea conditions. The simulation methodology is to be that as prescribed in the Guidelines developed by IMO and the method and outcome for an individual ship is to be verified by the Administration or CCS.
- In case where the simulation is not conducted, f_w value is to be taken from the "Standard f_w " table/curve prescribed in the Guidelines developed by IMO.

Note: Refer to Interim Guidelines for the calculation of the coefficient f_w for decrease in ship speed in a representative sea condition for trial use, approved by the Organization and circulated by MEPC.1/Circ.796.

Energy Efficiency Factor f_{eff}

f_{eff} is the availability factor of each innovative energy efficiency technology. f_{eff} for waste energy recovery system is to be taken as 1.0.

Verification of EEDI

Verification Process

Verification of the EEDI is to be conducted on two stages: preliminary verification at the design stage, and final verification at the sea trial. The verification procedure is shown in Fig. 2.

Preliminary Verification at the Design Stage

For the preliminary verification at the design stage, an application for the preliminary verification and an EEDI technical file containing the necessary information for the verification and other relevant background documents are to be submitted.

Prior to submitting the application for the preliminary verification of EEDI, the ship designer and/or the shipbuilder is to complete basic design of the ship and carry out a tank test, the Attained EEDI is to be calculated in accordance with the calculation methods in section "The Considering Procedure of MEPC for EEDI" above using the obtained parameters.

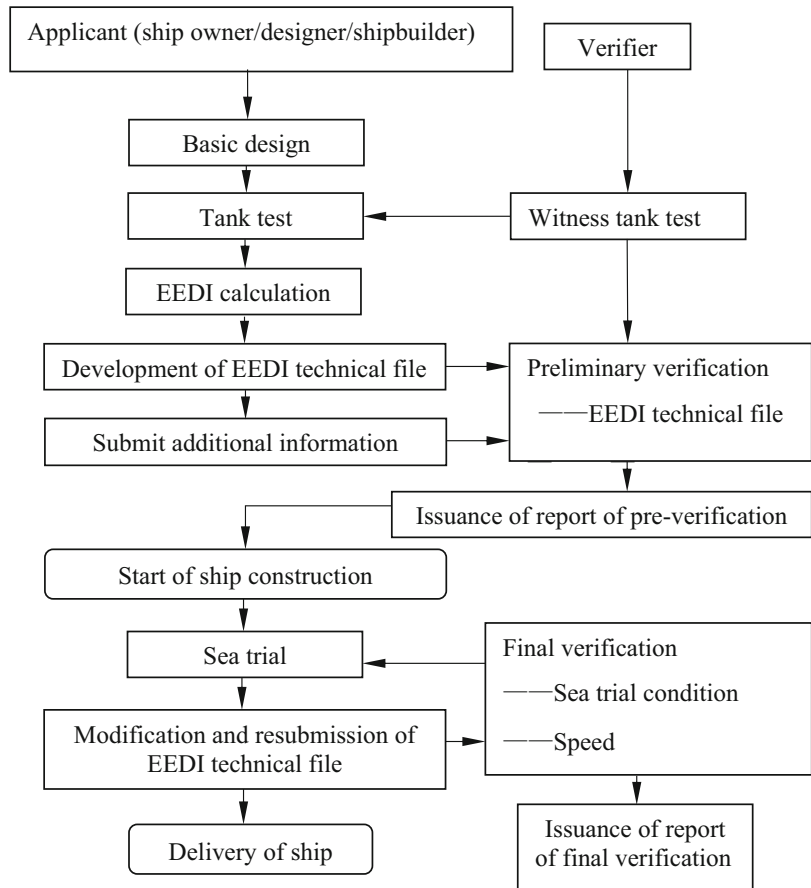
If dual fuel engines are installed, and gas fuel will primarily be used as fuel on the ship, the following information is to be provided:

- The use of boil-off gas or capacities of the gas fuel storage tanks, and the capacities of fuel storage tanks
- Arrangements of bunkering facilities for gas fuel in the intended operational area of the ship

The specific fuel consumption (SFC) of the main and auxiliary engines is to be quoted from the approved NO_x technical file; in case the NO_x technical file has not been approved at the time of the application for the initial survey, the test reports provided by manufacturers are to be used. In this case, at the time of the sea trial verification, a copy of the approved NO_x technical file and documented summary of the correction calculations are to be submitted.

The power curves used for the preliminary verification at the design stage are to be based on reliable results of the tank test. A tank test for an individual ship may be omitted based on technical justifications such as availability of the results of tank tests for ships of the same type. In addition, omission of tank tests is acceptable for a ship for

Energy Efficiency Regulations from IMO,
Fig. 2 Verification procedure



which sea trials will be carried out under the condition of specified draught corresponding to the capacity, upon agreement of the ship owner and shipbuilder and with approval of the verifier. For ensuring the quality of tank tests, the quality system of the International Towing Tank Conference (ITTC) is to be taken into account.

Final Verification of the Attained EEDI at Sea Trial

Sea Trial Conditions

At sea trial, the draught is to be set according to the section “[The Considering Procedure of MEPC for EEDI](#)” above at the speed V_{ref} .

Preparation Prior to Sea Trial

Prior to the sea trial, the following documents are to be submitted by the applicant:

- The procedure of speed testing, including all items to be measured and corresponding measurement methods to be used for developing power curves under sea trial conditions
- The final displacement table (or hydrostatic table) and the measured lightweight at inclining test, or a copy of the deadweight survey report
- A copy of NO_x technical file, if not submitted for the preliminary verification

Sea Trial Verification Requirements

Prior to the sea trial, the following conditions are to be confirmed:

- Prior to the sea trial, the ship is to be under seaworthy condition, and the measured power equipment are to be under good or normal condition.

- Draft and trim are to be confirmed by the draft measurements taken prior to the speed testing. The draft and trim are to be as close as practical to those at the assumed conditions used for estimating the power curves.
- Sea conditions are to be measured in accordance with ISO 15016:2002 or the equivalent.
- Ship speed is to be measured in accordance with ISO 15016:2002 or the equivalent and at more than two points of which range includes the main engine at 75% of MCR power.
- Output power measurement.
- The shipbuilder is to develop power curves based on the measured ship speed and the measured output of the main engine at sea trial.
- The shipbuilder is to compare the power curves obtained as a result of the sea trial and the estimated power curves at the design stage; in case differences are observed, the Attained EEDI is to be recalculated in accordance with the power curves at the sea trial.
- In case where the finally determined deadweight/gross tonnage differs from the designed deadweight/gross tonnage used in the EEDI calculation at the design stage, the ship owner or shipbuilder is to recalculate the Attained EEDI using the finally determined deadweight/gross tonnage. The finally determined gross tonnage is to be confirmed in the Tonnage Certificate of the ship.
- In case where the Attained EEDI is calculated at the preliminary verification by using SFC based on the manufacturer's test report due to the nonavailability at that time of the approved NO_x Technical file, the Attained EEDI is to be recalculated by using SFC in the approved NO_x technical file.
- The EEDI technical file is to be revised, as necessary, by the shipbuilder or the diesel engine manufacturer, taking into account the results of sea trial. Such revision is to include, as applicable, the adjusted power curve based on the results of sea trial, the final determined deadweight/gross tonnage and SFC described in the approved NO_x technical file, and the recalculated Attained EEDI based on these modifications.
- The EEDI technical file, if revised, is to be submitted to the verifier for the confirmation

that the revised Attained EEDI is calculated in accordance with the calculation methods in section [“The Considering Procedure of MEPC for EEDI”](#) above.

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- [Christmas Tree](#)

Environment Detection

- [Environmental Perception for Underwater Vehicles](#)

Environment Modeling

- [Environmental Perception for Underwater Vehicles](#)

Environmental Characterization Optics (ECO)

- [Underwater Information Sensing Technology](#)

Environmental Impact

- [Human Factors in the Role of Energy Efficiency](#)
- [Impact of Maritime Transport](#)
- [Ship Propulsion System](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

Environmental Impact Assessment (EIA)

- [Environmental Impact Assessment System](#)

Environmental Impact Assessment System

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Synonyms

[Environmental impact assessment \(EIA\)](#)

Definition

Development and environmental protection are often in conflict, and the purpose of an **environmental impact assessment (EIA)** is to contribute toward the decision-making process by resolving and preventing conflicts between the two (Environmental Impact Assessment Ordinance 1998). EIA is a system for forecasting future effects

of a new project on the environment, aimed to minimize negative effects. EIA is also a tool to assess ecological implications of projects, policies, and physical plans and prevent future environmental problems in an action in advanced manner.

Scientific Fundamentals

Sustainable development is development targeted to meet the needs and interests of the present generation without compromising those of the future generations. **EIA** system is a decision-making tool to prompt sustainable development by informing decision-makers to strike a balance between development and protection, and it requires legal compliance enforcement support to govern the issuing of **environmental permits (EP)**. The EIA system helps to achieve better environmental outcomes through identifying and evaluating potential impact of development on the environment alongside the necessary prevention and mitigation measures at the early stages of project planning and design, by applying the cardinal principle of EIA, i.e., **avoidance-minimization-mitigation** (Fig. 1). After screening and scoping, an EIA report identifies and predicts likely environmental and socioeconomic impacts and evaluates their significance. It should then develop measures to avoid, reduce, or compensate for impacts and attempt to offset any environmental damage. An EIA report usually includes an **Environmental Monitoring and Auditing (EM&A) Manual** (Environmental Impact Assessment Ordinance 1998), as EM&A is the critical implementation and follow-up stage after the project (Morrison-Saunders et al. 2001). Using the EIA system in Hong Kong as an example (Wood and Coppell 1999), the processes involved in conducting an EIA for a proposed project are described in the following sections.

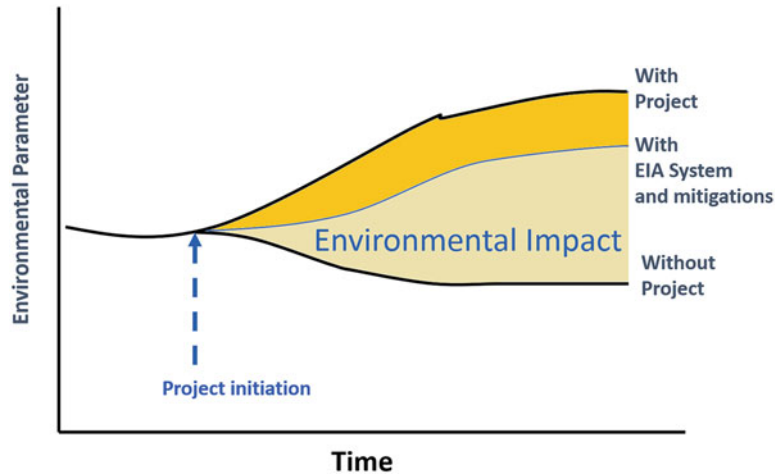
EIA Process

Screening and Scoping of a Project

While the exact process of an EIA and corresponding legal frameworks may differ

Environmental Impact Assessment System,

Fig. 1 An illustration that depicts the environmental impact of a developmental project, with residual impact after mitigations



between countries around the world, the role and purpose of an EIA is similar at its core. EIAs are intended to identify the potential impacts of a project in the early planning stages, as well as alternatives or measures to minimize environmental impact, which serve to inform decision-makers before proceeding with the project (Fig. 1).

The EIA system in Hong Kong is regulated by the **Environmental Impact Assessment Ordinance (EIAO)**. Under the Ordinance, the first stages of an EIA involve **screening and scoping**. Screening is the process of determining whether an EIA is required for a proposed project. To address this, project proponents are required to provide a Project Profile to enable the Director of **Environmental Protection Department (EPD)**, who enforces the Ordinance, to make informed decisions. The Project Profile should provide clear analysis on the scope of environmental issues of the project, issues and anticipated outcomes of the project, and technical and procedural requirements to be addressed in EIA. If there is no adverse impact anticipated in carrying out the proposed project, such project may be exempted from an **EIA Study/Report** and project proponents may directly apply for an environmental permit, which indicates approval by authorities to proceed with the project further (Fig. 2).

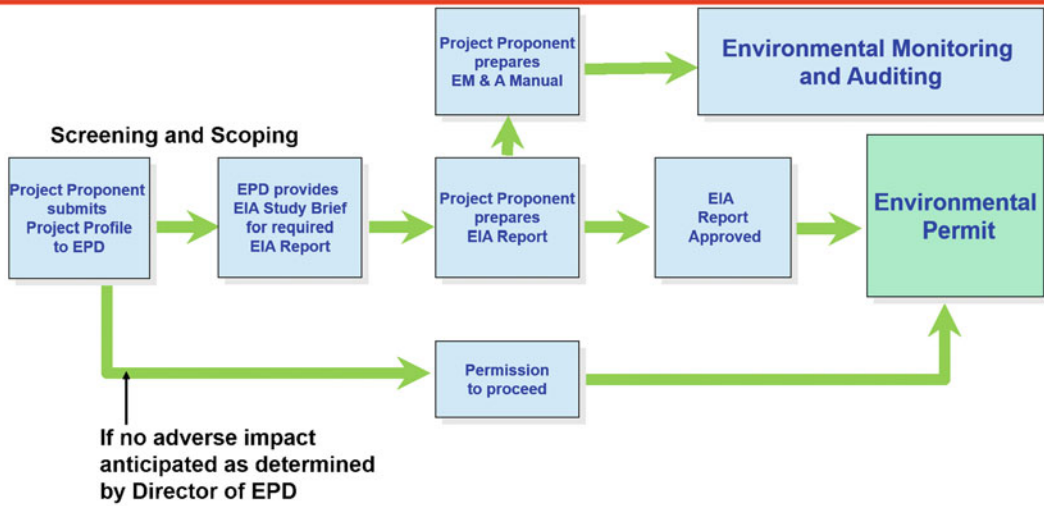
In most cases, a development project is likely to result in significant irreversible impact that needs to be addressed by an EIA Study before

proceeding (Fig. 2). Authorities including developers and experts will usually define key issues with the proposed project and possible impact in greater detail, known as scoping, in the form of an **EIA Study Brief**. The Study Brief should define the purpose and objectives of the EIA Study/Report, specifying the scope of environmental issues that need to be addressed including the type, scale, time frame, location, emissions, discharges, waste generation, destruction, alteration, or environmental changes that may result from the project. It should also highlight other requirements such as methodologies to follow, relevant experience and complaints from previous EIAs and environmental studies, time schedule, public participation, and budget issues. Depending on the nature of proposed project, the Study Brief should also stipulate the geographical and temporal boundaries of the project, define issues related to the combined impacts of projects and cumulative impacts of existing and planned developments in the vicinity of the project, and specify the duration of validity of the Study Brief itself as well as the number of EIA Reports and Executive Summary required. The Study Brief serves to inform the Project Proponent the issues to be addressed in the subsequent EIA Study/Report (Fig. 2).

EIA Study and Report

The next stage of EIA involves conducting the relevant impact assessments to generate a full EIA

Overview of the EIAO Process



E

Environmental Impact Assessment System, Fig. 2 An overview of the EIA process. Revised from training manual for the EIA mechanism by Hong Kong EPD (Wood and Coppel 1999)

Study/Report, following the scopes of EIA (as specified in the Technical Memorandum of the Ordinance). The objective of an EIA Study is to describe the proposed project and the associated works and their impact, identifying the elements of the community and sensitive receivers that are likely to be affected due to emission sources across potentially affected areas. The EIA Study should also identify and quantify the significance of the impact resulting from the project in terms of the potential losses and propose the provision of infrastructure or mitigation to minimize the pollution and environmental disturbance during construction and operation of the project. Residual effects after mitigation should also be considered. Finally, the EIA Study should look to provide appropriate monitoring and auditing design as a follow-up to the project. As a result, the EIA Study is often prepared by consultants and academics within a multidisciplinary team due to the need for expertise in determining whether standards and criteria in specific technical requirements, as defined by previous studies and enforced by legislation of the region, have been met.

Environmental Baseline Studies

Environmental baseline studies refer to the collection of baseline information on biophysical, social, and economic aspects of a project area. Project area refers to the area where environmental effects and impacts are felt (the site of the project, its access, and any other areas likely to be impacted) during construction or operational stages or decommissioning of a project. The purpose of environmental baseline studies is to reinforce current project's impact assessment by providing a description of the status and trends of environmental factors, to which identified and predicted impact and changes resulting from the project can be compared to and evaluated in terms of importance. Through monitoring, environmental baseline studies also provide a means of detecting actual changes after the project has been initiated. A good environment baseline enables potential environmental impacts from a project to be predicted and evaluated and is the foundation of an EIA, resembling the piling of a building. Baseline information plays a critical role in the EIA process as it is necessary to determine the existing environmental conditions on the site

and identify existing receivers and communities affected, habitats, and resources which are vulnerable to anticipated changes before any further project commences.

Major data sources and methods of collection are two critical issues in the environmental baseline studies. Impact assessment is fundamentally the prediction of the effect of a project based on established baseline information. To achieve this, a combination of accurate primary data sources and reliable secondary data sources is required. Primary data are results from the field or laboratory data collected and analyzed directly. In contrast, secondary data are data collected indirectly from previous published records or documents such as project documents, village profile, maps, photos, and Internet sources. Common methods of data collection include literature review, map interpretation, checklists (e.g., scaling and questionnaire checklists, matrices, etc.), and resource-based methods, such as scientific instruments and techniques (inventory, species area curve, sampling techniques, etc.). A good baseline information should consider multiple data sources to form an overview on the environment. The impact on a project area due to the proposed project can then be modeled using primary data and compared with the baseline information. For example, meteorological condition data can be used to model air quality impacts in assessments.

Impact Assessment and Mitigations

The impacts which may result from a proposed project are assessed comprehensively under different criteria. In most EIA reports, possible impact on the environment include aspects of air quality, noise, water quality, waste management implications, terrestrial ecology, marine ecology, fisheries impact assessment, hazard assessment, socioeconomic impact, cultural heritage and anthropological impact, and landscape and visual impact. For each of the identified impact, the EIA Report should clearly indicate the relevant environmental legislation, policies, plans, standards, and criteria that referred to in the assessment. The Report should then provide a description of the environment of the project area and identify

the environmental impact as well as the sensitive receivers of the impact through construction and operation phases of the project. The assessment methodologies, baseline information, evaluation and mitigation of impact, and residual impact for both construction and operation phases should then be clearly described. Finally, environmental monitoring and auditing design should be proposed for each of the assessed impact, often standalone in an **EM&A Manual** which supplements the EIA Study/Report.

Project Monitoring and Auditing

As a follow-up of the EIA, project monitoring and auditing covers both the construction stage and operational stage of the development project, and it spans a much longer time period than the completion of the EIA Report. Project monitoring and auditing can help to verify that the EIA predictions and assumptions are valid, monitor the actual environmental performance to preempt impacts before breaching established criteria, ensure that mitigation measures are properly and timely implemented, and improve future EIA predictions.

In general, EM&A shall be required if the project has the potential of causing environmental impacts which are or are likely to be prejudicial to the health or well-being of people, the flora, fauna, or ecosystem if the recommended measures are not properly implemented. EM&A should also be required if it involves mitigation measures of which the effectiveness may require a long period to establish, e.g., compensatory planting of trees, an unproven technology, or mitigation measure, or is based on a new technique or model, or there is other uncertainty about design assumptions and/or the conclusions, or project scheduling is subject to change such that significant environmental impacts could result.

In Hong Kong, the issuing of EP may impose requirements for monitoring to verify predictions or the effectiveness of measures as conditions. The specifications are derived from the EIA Study carried out by project proponents. In an approved EM&A Manual, EM&A programs required will be clearly defined including the monitoring parameters, frequency, duration, and

monitoring station locations. Similar to the EIA Study, EM&A work should also be carried out by qualified personnel.

In the implementation of EM&A, the proponent is responsible for implementing the necessary mitigation measures, ensuring the expected performance is achieved, and taking necessary remedial measures to address unanticipated or unacceptable impacts. The steps in developing a monitoring program should be methodical. The scope and objectives of the monitoring program should be identified to decide what and how data are to be collected and analyzed. The area to be monitored should be clearly defined as well as the follow-up actions to be carried out. In a multi-disciplinary team, responsibilities of each party involved should be clearly defined.

Key Elements of an EM&A Program

The key elements of an EM&A program include baseline monitoring data, impact/compliance monitoring program, action/limit levels and event/action plan, reporting procedures, and implementation mechanism. Environmental monitoring refers to the systematic collection of environmental data through a series of repetitive measurements. Two different monitoring activities are involved in the process: baseline monitoring and impact/compliance monitoring.

Baseline monitoring refers to the measurement of environmental parameters during a representative pre-project period for the purpose of determining the nature and ranges of natural variation. Baseline monitoring can also be applied to establish, where appropriate, the nature of change.

Impact/compliance monitoring involves the measurement of environmental parameters during project construction and implementation to detect changes in these parameters which can be attributed to the project. This can serve to ensure that regulatory requirements are in place, and standards are met.

Aside from monitoring, environmental auditing is an equally important part of the EM&A program. Audit is a term taken from financial accounting to infer the notion of verification of practice and certification of data. **Environmental audit** is the process of verification that all or selected parameters

measured by an environmental monitoring program are in compliance with regulatory requirements, internal policies and standards, and established environmental quality performance limits. Environmental audit involves the process of comparing project impact predictions with actual impacts for the purpose of assessing the accuracy of predictions, the effectiveness of the environmental management system, practices and procedures, and determining the degree and scope of any necessary remedial measures in case of exceedances of compliance.

To supplement the auditing process, **action levels** are the levels of parameters which if exceeded are indicative of a deteriorating ambient environmental quality. Appropriate remedial actions may be necessary to prevent the environmental quality from going beyond the limit levels, which would be unacceptable. **Limit levels** are the levels stipulated in relevant pollution control ordinances, or the Hong Kong Planning Standards and Guidelines, or other appropriate criteria established by EPD for a particular project, beyond which the works should not proceed. Details in legislation may differ between regions. In an EM&A program, the **event/action plan** specifies the responsibilities of each key player, and “what” action to be undertaken by “whom” in case of exceedance in action/limit levels. The normal action for single exceedance of action level is to increase the monitoring frequency in order to identify the cause of problems and to be subsequently addressed.

Key Applications

The difference between an EIA and Strategic Environmental Assessment lies in application. EIA is used to identify the environmental and social impacts of a proposed project, usually by individuals and companies, prior to decision-making in order to predict environmental impacts at an early stage in project planning and design. In contrast, Strategic Environmental Assessment is used at the policy, planning, and programming levels. Strategic Environmental Assessment is a systematic process for evaluating the

environmental implications of a proposed policy, plan, or program and provides means for looking at cumulative effects and appropriately addresses them at the earliest stage of decision-making alongside economic and social considerations. In any case, early consultation and involvement of EIA professionals in a project's conceptual development, planning, and design is the recommended approach, as it can often lead to a better project design with the consideration of sensitive receivers and avoidance of impacts. Having to revise a poorly designed project plan at a later stage is inefficient and can be a major setback for any project and its stakeholders.

Since the publication of *Silent Spring* by Rachel Carson back in 1963, the issue of environmental protection has been brought up and slowly developed into a global topic which demanded action (Environmental Impact Assessment Ordinance 1998). There is a desperate need for sustainable development which meet present needs without compromising the future generations. As a result, environmental impact assessment is a well-developed, organized system of evaluating the nature and extent of environmental impact arising from construction and operation of any proposed development project. It provides a comprehensive overview of potential impact on not only the environment but also at the socio-economic level, concerning human health and cultural impact, to aid the decision-making process and optimize the balance between development and protection with mitigation measures. The philosophy of preventing impact from happening instead of reacting to environmental damage which already occurred proved to be a more effective approach in addressing environmental problems around the world (LIM 1988).

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Environmental Perception

► [Environmental Perception for Underwater Vehicles](#)

Environmental Perception for Underwater Vehicles

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Synonyms

[Autonomous underwater vehicles \(AUVs\)](#); [Environment detection](#); [Environment modeling](#); [Environmental perception](#); [LIDAR \(Light Detection and Ranging\)](#); [Synthetic aperture sonar \(SAS\)](#); [Unmanned underwater vehicles \(UUVs\)](#)

Definition

Environmental Perception is one of the most basic functions of underwater vehicles, and it is the method for underwater vehicles to recognize their environment. Through environmental perception technology, underwater vehicles can determine their own location and identify the distribution of surrounding objects, targets, or obstacles. It is the basis for realizing the navigation and path planning of underwater vehicles.

Scientific Fundamentals

The development of oceans and the associated competition have become the strategic focus of many countries, and these goals have become pursued with increasing fierceness. Compared with other underwater vehicles that can be

explored, underwater vehicles have numerous advantages such as underwater payload capacity, maneuverability, and depth of activity. In a variety of marine technologies, autonomous underwater vehicles (AUVs) can be used for comprehensive surveys and studies in areas where the depth cannot be reached by general diving technology; their ability to accomplish various missions have brought marine development into a new era (Hall 2006; Bonin-Font et al. 2017).

As human activity space gradually expands into the marine sector, underwater vehicles will have an important role in ocean exploration. Underwater vehicles development involves high technologies, such as mechanics, fluid mechanics, hydroacoustics, optics, electronic communication, navigation, automatic control, computer science, sensor technology, bionics, artificial intelligence, and many other contemporary achievements. The important application value of AUVs has received an increasing amount of attention from scientists and technologists in the civil and military fields. In the field of marine engineering, AUVs are employed for the structural inspection of dams, installation/disassembly of underwater bases, observation of underwater targets, and search and rescue; they also aid divers. In the field of marine scientific research, AUVs can be deployed to conduct data collection in a marine environment, investigate underwater shipwrecks, perform geological and geomorphological exploration of the seabed, and explore petroleum and other resources. In the military field, AUVs are used in mine countermeasures, target detection, intelligence gathering, surveillance and reconnaissance, environmental data collection, and anti-submarine warfare. To achieve environmental perception and autonomous navigation of underwater vehicles, a variety of techniques is required to collaborate on an AUV (Dagleish et al. 2007; Xu et al. 2007).

General System Structure of Underwater Vehicles

Marine scientists are among the first to capitalize on the use of underwater vehicles for environmental perception. Oceanographers, for instance,

started using unmanned underwater vehicles (UUVs) to study deep marine environments and the seafloor. However, although the majority of such early applications were devoted to monitoring marine habitats, nowadays there are a significant and increasing number of underwater vehicles that contribute to other environmental science domains.

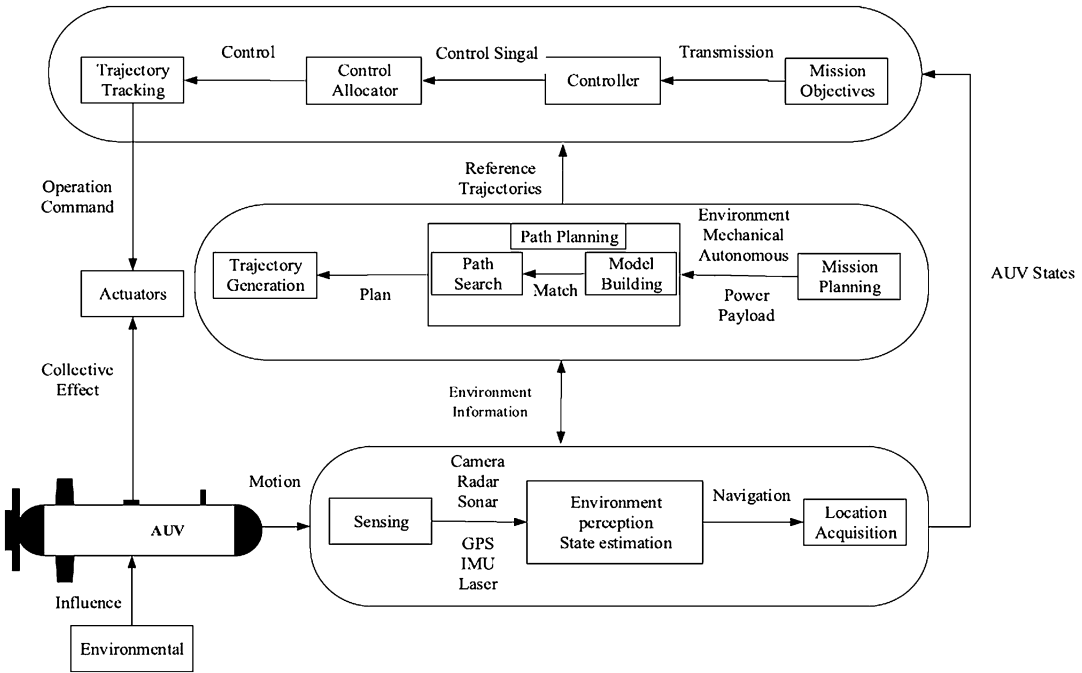
With the increased selection of sensors now included in the AUV allowing the performance of underwater explorations, the system composition and structure of underwater vehicles have become much more complex, as shown in Fig. 1, and at the same time, the functions of the vehicles have become more powerful. The underwater 3D mapping now relies on acoustic multibeam or side-scan sonars to produce elevations maps (Hurtos et al. 2010; Pathak et al. 2010). This ability to accurately map underwater environments yields high added value to any survey, as such results convey immense information easily interpretable by humans.

Environmental Perception Technologies

Sonar Perception

Reliable, sonar-based perception is crucial if robots are to perform useful tasks in the ocean. Undersea sonar data, however, is not easy to interpret. The dominant approach for undersea sonar processing is based on “imaging.” Undersea sonars are often designed to create as large an aperture as possible, resulting in narrow beam patterns. These beams can then be steered, either mechanically or electronically, to produce a visual representation of the scene – an image. Most interpretation is either performed manually by human operators or automatically by image processing algorithms such as shadow detection. Common examples of imaging undersea acoustic sensors include side-scan sonars, forward-look sonars, and mechanically scanned sonars.

In most work in synthetic aperture sonar (SAS) the implicit goal is to produce an “image” of the scattered field (Rolt 1991). One of the key issues in a geometrical approach to sonar data interpretation is the correspondence problem. The goal is to associate measurements obtained from multiple



Environmental Perception for Underwater Vehicles, Fig. 1 The relationship of AUV autonomous control system structure. (Li et al. 2018)

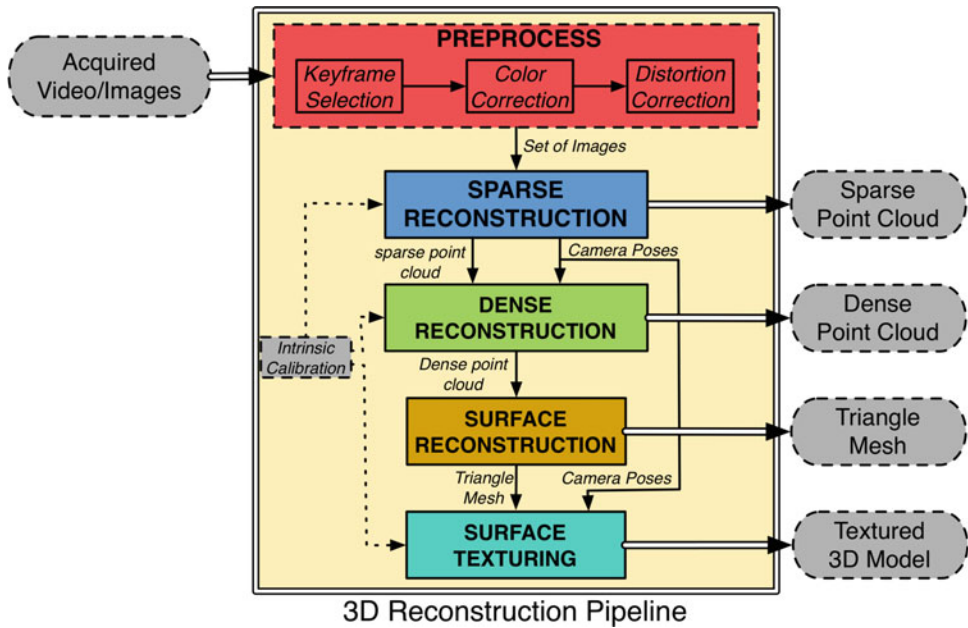
vantage points with the same feature/object in a scene. Previous approaches have considered the correspondence problem in conjunction with the object classification and mapping problems (Moran et al. 1997). Freedman's method predicts that when a sonar pulse is an incident on a generally smooth but possibly faceted object, the returned waveform will be comprised of a discrete number of image pulses, which are time-delayed and attenuated replicas of the incident pulse. This simplified physical model of the sonar sensing process is based on the geometrical acoustics high-frequency approximation of acoustic scattering (Rikoski and Leonard 2003).

Optical Perception

Underwater environments with their particular phenomena (i.e., light attenuation, blurring, and low contrast) can be regarded as hostile places for optical surveying. Acquisitions have to be performed at close range, significantly limiting the area viewed in a single image, thus enormous amounts of data have to be collected and

processed to generate a wide-area view enabling the extraction of valuable information on a broader space scale. In scenarios with significant 3D structures, the aforementioned prerequisites cannot be met, resulting in obvious distortions. However, redundant information from multiple images can be used to provide visually rich 3D reconstructions of the underwater terrain. Furthermore, as the camera poses do not have to be orthogonal to the seafloor, images can convey more meaningful information regarding the global shape of the object, especially in the case of intricate structures (e.g., underwater hydrothermal vents), which are impossible to capture accurately using downward-looking cameras.

Hernandez et al. (2016) have presented an optical-based 3D reconstruction pipeline. Through a series of sequential modules, as shown in Fig. 2, the pipeline is able to reconstruct a 3D model based on the optical imagery acquired from an arbitrary number of cameras in various poses. Cameras do not have to be synchronized and can be freely repositioned between missions, based on the



Environmental Perception for Underwater Vehicles, Fig. 2 3D reconstruction pipeline. (Hernandez et al. 2016)

mission’s goals and the expectations of the terrain in the surveyed area. However, in order to obtain a metric-scaled reconstruction, the cameras have to be connected with the other sensors in the AUV or known measurements of the environment have to be introduced (Garcia et al. 2011). Each of the reconstruction steps results in a different representation of the reconstruction. While the preferable representation for visualization and dissemination is a textured triangle mesh, some applications may require solely dense or even a sparse point cloud of points describing the observed area. It is also worth noting that an important preprocessing step of color correction is required to reduce the image degradation effects of the water medium.

Laser Perception

Blue and green laser systems provide underwater target detection with a higher range, more precise positioning, and better imaging than traditional electromagnetic and acoustic detection. These systems are often utilized for underwater laser short-range target detection technology research. Underwater vehicles equipped with laser detectors can complement sonar detection technology obtain high-precision underwater images, enhance the

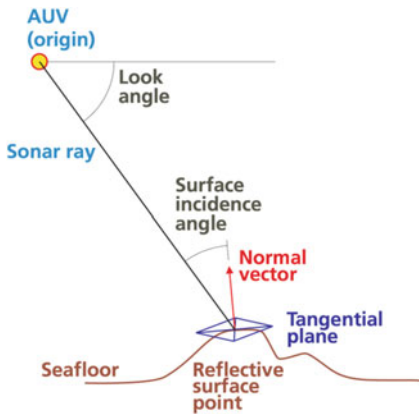
detection capability of underwater targets – especially underwater high-speed moving targets – and assist in completing various underwater exploration tasks (Zha and Zhang 2014).

When blue and green laser pulses are transmitted underwater, the beam energy attenuates due to the strong absorption and scattering of the seawater. The strong scattering effect causes the beam to diffuse in space, and the multipath effect of the photon leads to optical pulse broadening and transmission delay which altogether degrade the accuracy of underwater pulse laser ranging. At present, research on the underwater laser detection system mainly focuses on the transmission characteristics of light in seawater and the elimination of backscattering noise. Signal intensity, noise estimation, pulse distortion, and other factors are mainly studied, and the signal processing of the receiving circuit is simplified (Lee et al. 2013; Zotta et al. 2014).

Key Applications

Woock (2012) has discussed what type of features is suitable for sonar-based underwater navigation.

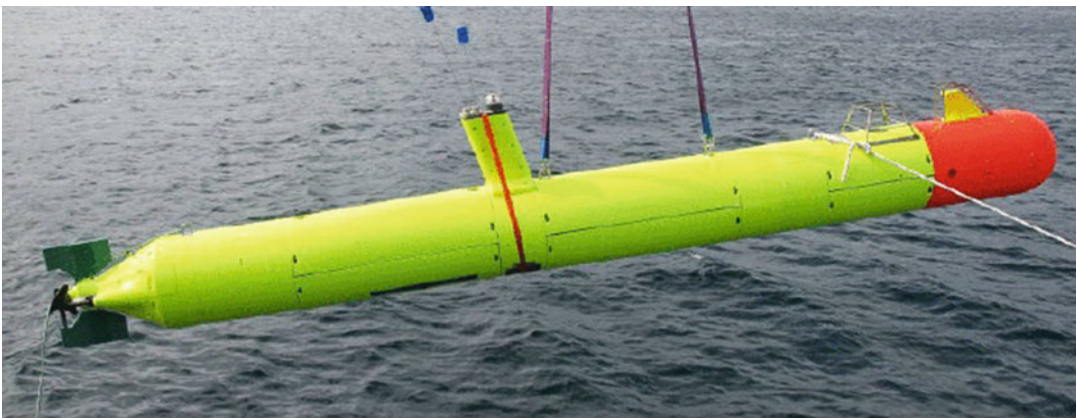
For evaluation, a simulation environment has been created (Fig. 3) which performs environment shape reconstruction based on side-scan sonar data while additionally utilizing vehicular inertial measurement data. A ray-tracing based forward model creates a synthetic sonar response for a given elevation map and vehicle pose. Then, a pose-aware inversion method estimates an elevation map of the surrounding environment. This map is the basis for navigation. Therefore, the characteristics of the reconstruction method need to be taken into account for the feature design as well as the fact that the surface will be sampled differently upon a re-visit (Woock 2012).



Environmental Perception for Underwater Vehicles, Fig. 3 The scheme of sonar perception method. (Woock 2012)

The TIETeK project is a joint Fraunhofer research project with the goal of showcasing technologies for deep-sea exploration and inspection tasks up to a depth of 6000 m. A prototype vehicle incorporating various Fraunhofer deep-sea technologies has been built and is shown in Fig. 4. The vehicle is of pressure-independent design and therefore features no pressure hull. With regard to sonar and data processing, the vehicle is equipped with two side-scan sonar sensors of switchable frequency (250 kHz, 500 kHz, 1 MHz), an Xsens IMU, a fiber-optic gyroscope, and a computing board (Woock 2012).

Hernandez et al. (2016) have presented a new end-to-end approach for autonomously mapping unknown underwater natural environments using AUV. They propose a framework composed of two main functional pipelines. The first provides the AUV with the capability for creating an acoustic map online, while simultaneously planning collision-free paths. Such functionality is essential for safe navigation in unknown and potentially dangerous areas, and to maintain a very short distance to the areas being imaged, as required by optical mapping underwater. Using the gathered image data, the second pipeline builds a photo-realistic 3D model, which can be used as base maps for environmental inspection and subsequent monitoring. In order to thoroughly validate the proposed approach, Hernandez et al. present the results obtained in a challenging real-

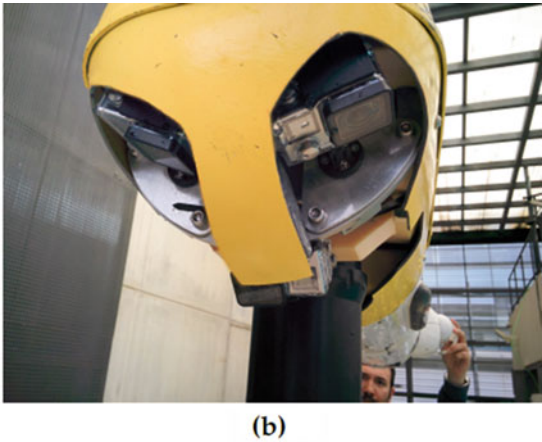
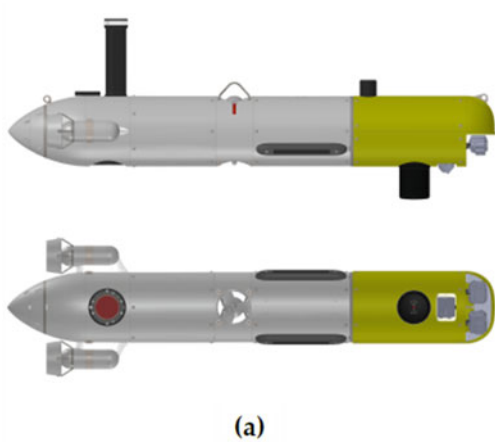


Environmental Perception for Underwater Vehicles, Fig. 4 The Fraunhofer prototype vehicle of the TIETeK project. (Woock 2012)

world natural scenario, in which the Sparus II AUV (Fig. 5) conducts several autonomous missions. The test scenario, containing an underwater canyon between two rocky formations, permitted the researchers to demonstrate the extent of the capabilities of the approach. By successfully navigating through the natural environment, the Sparus II AUV is able to acquire data subsequently used in the reconstruction of a complex textured 3D model of the area.

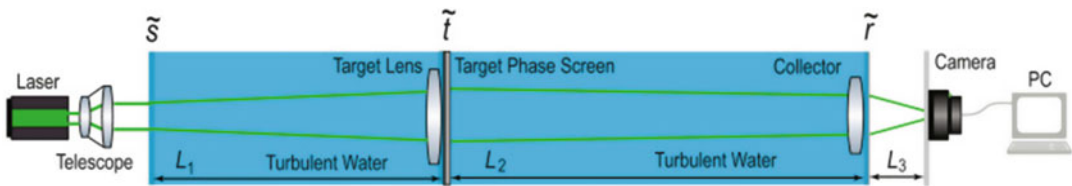
Korotkova and Yao (2020) have implemented a novel matrix method based on the extended Huygens-Fresnel integral for the analysis of optical beam propagation in the bi-static and mono-static LIDAR (Light Detection and Ranging) systems operating in the presence of underwater turbulence, as show in Fig. 6. By applying three transformations: for the beam passage from source to target, for its interaction with the target,

and for its propagation from target to the receiver, sequentially, they illustrate how the second-order moments of the beam can be predicted and compared at the source plane, just before the target, just after the target, and at the collecting lens of the receiving system. They have also carried out a very detailed numerical analysis of the beam evolution in a bi-static LIDAR system, using, without loss of generality, a linearly polarized and fairly coherent source. The results indicate that the underwater optical turbulence degrades the spectral density and the degree of coherence as follows: the lower value of the average temperature, a larger value of the temperature dissipation rate, or a smaller value of kinetic energy dissipation leads to more beam broadening and lower coherence. They have also illustrated that other parameters of turbulence affect the beam in a very complex manner.



Environmental Perception for Underwater Vehicles, Fig. 5 Sparus II AUV. (a) CAD model, where the three thrusters can be observed, as well as the profiling sonar and

cameras located in the payload area (yellow); (b) real-world vehicle's payload. (Hernandez et al. 2016)



Environmental Perception for Underwater Vehicles, Fig. 6 Bi-static underwater LIDAR system. (Korotkova and Yao 2020)

Cross-References

- [Environment Modeling](#)
- [Environmental Perception](#)

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Environmental, Social, and Health Impact Assessment (ESHIA)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Ergonomics

- [Human Factors in Ship Design](#)

ETLP (Extended Tension Leg Platform)

- [Tension-Leg platform](#)

Eutrophicated Water

- [Eutrophication](#)

Eutrophication

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Synonyms

Eutrophicated water; Eutrophication control;
Harmful ecological effects; High nutrients

Definition

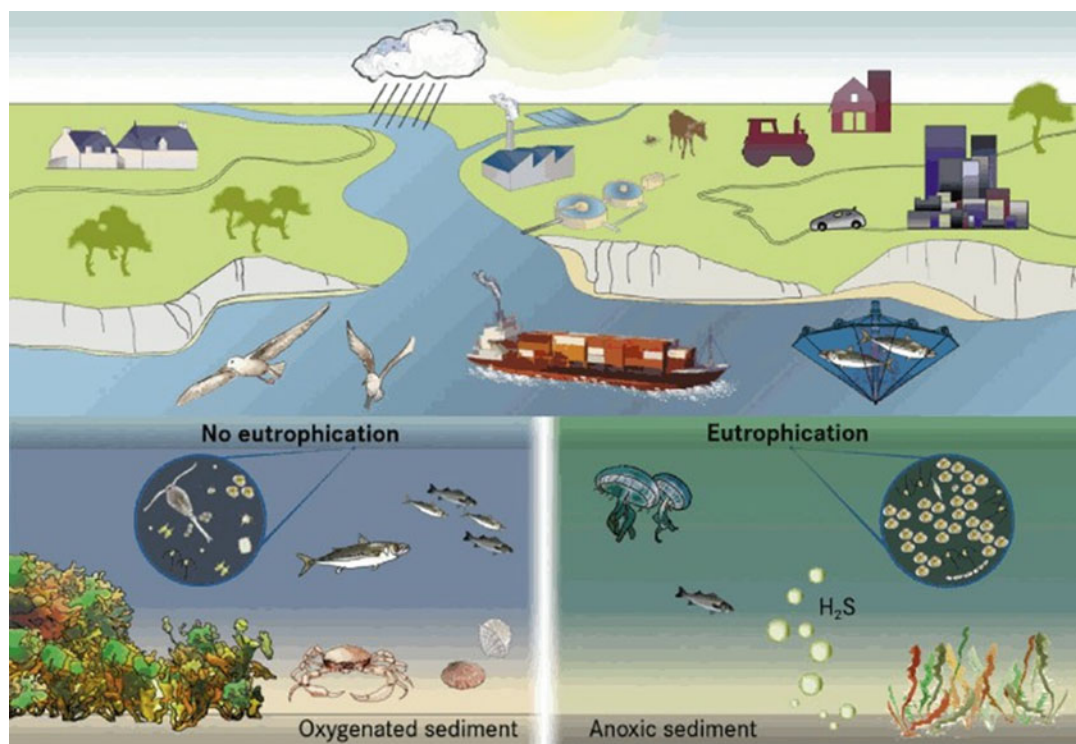
Eutrophication is the process of increased anthropogenic nutrient inputs to lakes, estuaries, and other aquatic systems over the natural supply and the subsequent excessive biological production of organic matters. Nutrient fertilization in

aquatic environments may result in a series of harmful ecological effects including the development of hypoxia ($[O_2] < 62.5 \mu M$), acidification, and die-off of fishes and benthic organisms in waters (Fig. 1).

Introduction

Eutrophication, the process of changing water system from nutrient-poor to nutrient-rich, negatively influences water quality. It profoundly increases the biomass of undesirable species. Even, it increases the occurrence of harmful algal blooms in the lakes, estuaries, and coastal regions (Zhou et al. 2008; Jin et al. 2010; Jiang et al. 2014; Wang et al. 2017; Chen et al. 2020).

Algae use sunlight, water, carbon dioxide, as well as nutrients to build organic matter for their own growth, development, reproduction, and other life activities. In considerable aquatic environments, nitrogen and phosphorus limit



Eutrophication, Fig. 1 Schematic view of eutrophication (OSPAR 2010, https://qsr2010.ospar.org/en/ch04_01.html)

phytoplankton growth. However, because of human activity, a large amount of domestic sewages and industrial wastewaters, which are rich in nutrients such as nitrate and phosphate, are discharged into lakes, estuaries and nearshore waters. Such enormous nutrient input even trigger explosive growth of phytoplankton in marginal seas (Wang et al. 2017). In a favorable water condition, biomass will accumulated due to intense algae bloom which blocks sunlight into water. Subsequently, oxygen dissolved in water are consumed significantly by decomposition of organic matters.

Eutrophication

Nutrient Source for Eutrophication

The nutrients needed by phytoplankton in seawater mainly include nitrogen, phosphorus, silicon, carbon, trace elements, and various vitamins. In general, the limiting factors for the phytoplankton growth are nitrogen and phosphorus, rather than silicon and carbon in seawater. Silicon limitation and iron limitation are mainly reported in open oceans. The main sources of nutrients in coastal waters include:

1. Terrestrial runoff. Terrestrial runoff mainly includes rivers, domestic sewages, and industrial and agricultural wastewater. Riverine nutrients are usually the most important nutrient source in the estuary and coastal waters. The extreme high nutrient concentrations in runoff contribute to the eutrophication of coastal waters and trigger intense algae bloom (Turner and Rabalais 1994; Grelowski et al. 2000; Wu et al. 2020).
2. Atmospheric nutrient deposition. Atmospheric input is an important source of nitrogen in coastal waters. The combustion of fossil fuel and automobile exhaust leave nitrogen oxides in the air, which enter aquatic environment by dry and wet deposition. By comparison, atmospheric deposition of phosphorus is much smaller than that of nitrogen (Beddig et al. 1997; Rendell et al. 2017).
3. Aquaculture wastewater. Aquaculture wastewater contain large amount of unused bait and fecal pellets. Degradation of the organic matters by bacterial release inorganic nutrients into waters, which also contribute to the eutrophication process

Ecological Effect

Algal Blooms and Red Tide

The eutrophication of lakes, estuaries, and coastal waters favors the growth of algae or aquatic plants. The floating plants may cover the surface waters or even filled the water column in shallow waters. The biomass accumulation of algae or aquatic plants in waters decreases transparency of water body, which blocks the penetration of sunlight. The eutrophication in coastal waters may even trigger red tide, during which algal blooms are so intense that they even discolor the coastal waters. Under a worse condition, harmful algae will release toxin that will kill fish, shellfish, marine mammal, and birds and threaten the health of human being.

Hypoxia and Subsurface Acidification

Algal blooms in coastal waters produce enormous organic matters during the exponential growth phase. As the nutrients are depleted, and sunlight is blocked by surface algae, phytoplankton growth slows. The decomposition of organic matter exceeds the production of organic matter eventually. Thus, the oxygen level will decrease continuously. In the river estuary and marginal seas, eutrophication results in intense phytoplankton blooms in upper layer, and the subsequent setting and decomposition of fresh organic carbon contribute to severe hypoxia and acidification in the bottom water (Cai et al. 2011). The extreme low oxygen level, or even the generation of H_2S , results in the death of fishes and benthic organisms. Thus, the hypoxic waters are also called “dead zone.”

Simplification of Food Web Structure

The process of eutrophication reduced the number of species and increased the abundance of mesoeutrophic to hypereutrophic species (Van

Dam and Mertens 1993), which will cause changes in species relationship in the community, simplified food web structure. A shift in primary producers from eelgrass to macroalgae in response to increased nutrient loading is found (Deegan et al. 2002). It altered the habitat, physicochemical structure, and food webs. The nitrogen decrease lowers the density and biomass of eelgrass. However, it promotes a record-increase of algal biomass. Serious eutrophication can also lead to the loss of important marine habitats such as seagrass beds and coral reefs.

Evaluation of Eutrophication Level

$$E = c(\text{COD}) \times c(\text{DIN}) \times c(\text{DIP}) \times 10^6 / 4500*$$

E is the nutrient index (First Institute of Oceanography Ministry of Natural Resources 2008).

$E \geq 1$ indicates the water area has exhibited the characteristics of eutrophication.

$c(\text{COD})$ is the concentration of chemical oxygen consumption, mg/dm^3 .

$c(\text{DIN})$ is the concentration of dissolved inorganic nitrogen, mg/dm^3 .

$c(\text{DIP})$ is the concentration of dissolved inorganic phosphorus, mg/dm^3 .

4500 is the product of single eutrophication threshold values of COD, DIN, and DIP for specific oceans or lakes. These three thresholds are distinct for various aquatic environments. Threshold ranges of COD $1 \sim 3 \text{ mg}/\text{dm}^3$, DIN $0.2 \sim 0.3 \text{ mg}/\text{dm}^3$, DIP $0.01 \sim 0.02 \text{ mg}/\text{dm}^3$ were suggested and recommended.

Eutrophication Control

Eutrophication is worsen in lakes, estuaries, and coastal regions all around the world, especially in the context of global change. Nutrient inputs to aquatic system are mainly from human activities (agriculture fertilization, sewage discharge, aquaculture, etc.). Thus, the effective treatment of eutrophication is “prevention and protection” strategy in lots of countries, including ecosystem-based management, reducing the use of nitrogen fertilization, implementing pollutant emission control, developing watershed protection, and increasing water self-purification capacity.

Biological control is one of the common methods to control eutrophication. Some aquatic macrophytes, such as *Alternanthera philoxeroides*, which improves the transparency of eutrophic lake water, are capable of purifying eutrophic lake water (Wang et al. 1999). *Hydrodictyon reticulatum* was reported to actively remove 67.3% nitrogen and phosphorus in 6 days under different environmental conditions (Liu et al. 2004). Some phytoplanktivorous fish can be utilized for weed management and counteracting eutrophication (Opuszynski et al. 1995). Several microorganisms are reported to be efficient for scavenging phosphates from sewage sludge (Lawson and Tonhazy 1980; Florentz and Hartemann 1984).

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Eutrophication Control

► Eutrophication

Evanescent Wave (EW)

► Fiber Optic Hydrophone

Excessive Accelerations

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Wuxi, China

Definition

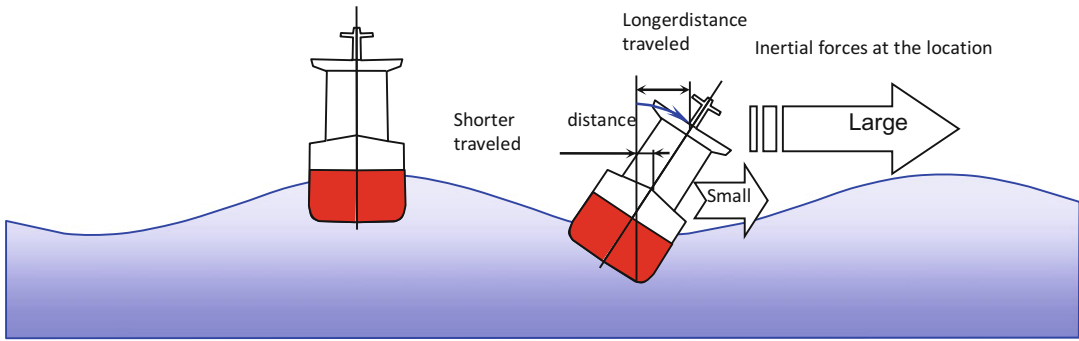
When a ship is rolling, objects in higher locations travel longer distances. A period of roll motions is the same for all the locations onboard the ship. To cover a longer distance during the same time, the linear velocity must be larger. As the velocity changes its direction every half a period, larger linear velocity leads to larger linear acceleration. Large linear acceleration means larger inertial force, see Fig. 1 (SDC 3/WP.5/Annex 7 2016).

Inertial forces acting in the horizontal plane are more dangerous for a human than vertical inertial forces. The vertical inertia forces cause brief overloading, while horizontal inertia forces cause humans to lose balance, fall, or even be thrown against walls, bulkheads, or other structures. Large acceleration is mostly caused by roll motions, so they have predominantly lateral direction.

If GM value is large, the period of roll motion is smaller. Thus, for the same roll angle changes of linear velocity occur faster, so acceleration is larger (SDC 3/WP.5/Annex 7 2016).

Typical Excessive Accelerations Accident

In the years 2008/2009 the world economic crisis caused a reduction in the number of transported TEU, due to a significant decrease in transported goods. Consequently, a great number of container vessels had been laid up or were forced to operate with a small amount of cargo on board. In this loading condition container vessels have very high stability. In many cases, their vertical center of gravity is even located below the lightship condition's coordinate. This is due to large amounts of ballast water in wing and double



Excessive Accelerations, Fig. 1 Scenario of stability failure related to excessive accelerations (SDC 3/WP.5/Annex 7 2016)

bottom tanks as well as no or only a small amount of cargo located at cargo hold bottom. A high amount of ballast water is needed to obtain an adequate hull and propeller immersion as well as a limitation of the longitudinal bending moments within the hull structure. In the typical loading conditions with higher amounts of cargo, less ballast water is needed, and the stability is lower. Under the described circumstances, several accidents have happened to container vessels during the last years. Accidents caused not only severe damages on ships but also heavily injured and even killed crew members. They have been thrown through the bridge due to high transversal acceleration caused by the heavy roll motions of the ship. This highlights that modern container ship designs face some problems concerning insufficient seakeeping behavior (SLF54/INF.7 2011).

The accident of the CMS Chicago Express happened on September 24th, 2008, during heavy weather in the South China Sea near Hong Kong. The maximum rolling angle was about 35° , and the maximum transversal acceleration on the bridge exceeded 1.0 g. One crew member was killed, one heavily injured, and several slightly injured. There were no noticeable damages to the vessel's hull (SLF54/INF.7 2011).

The accident of a 2468 TEU containership happened on September 15th, 2009, during heavy weather in the South China Sea near Hong Kong. The maximum rolling angle was about 30° . The maximum transversal acceleration on the bridge exceeded 1.2 g. One crew member

was killed. Heavy damages to the vessel's hull occurred (SLF54/INF.7 2011).

The accident of a 2500TEU containership happened on October 16th, 2009, during heavy weather in the North Sea near Borkum. The maximum rolling angle was about 30° . The maximum transversal acceleration on the bridge exceeded 1.0 g. One crew member was heavily injured. No damages to the vessel's hull occurred (SLF54/INF.7 2011).

Model Test for Excessive Acceleration

The existing accidents show that the excessive acceleration generally occurs in the ballast condition. The larger GM value and the smaller roll damping caused by a stall in waves will lead to the excessive lateral acceleration at the higher position of the hull under such loading conditions. In case of excessive acceleration, the rolling amplitude of the accident ship can exceed 30° . Due to excessive stability and short period, the maximum lateral acceleration could exceed 1.0 g. The speed of the hull was quite low when the accident occurred. It is reported that when the two accidents of Chicago express and Guayas occurred, the navigation speed was particularly low, only about 2–4 knots.

From the above analysis, it can be seen that under ballast or close to ballast and low speed, excessive acceleration is more likely to occur, which will be accompanied by large roll amplitude. Therefore, the beam wave and zero speed are

usually selected for the research of excessive acceleration. At this time, resonance rolling is easy to occur, which is easy to cause excessive acceleration.

Here is an example of a model test of excessive acceleration. One containership is selected, and the principal particulars are given in Table 1. Model test photos are shown in Fig. 2. Excessive acceleration model test was carried out in the seakeeping basin of China Ship Scientific Research Center. The model test is restrained by four springs.

The lateral accelerations of the three measuring points are measured by the acceleration sensor, and the coordinates of the measuring points are listed in Table 2. The layout of measuring points on the model are respectively at the port side of the center of gravity (P1), the starboard side of the cab (P2), and the starboard side of station 0 at the stern of the model (P3), as shown in Table 2.

Firstly, the model test of the free roll decay curve is carried out, and then the model test is carried out in regular and irregular beam waves at

zero speed. Eleven wave conditions in regular beam waves are carried out, with wavelength/ship length = 0.2, 0.3, 0.4, 0.6, 0.8, 1.0, 1.2, 1.4, 1.6, 1.8, and 2.0, respectively, and the wave steepness is fixed at 0.03. Irregular wave conditions are shown in Table 3, and ITTC 2-parameter spectrum is used for simulation.

Key Numerical Method for Excessive Acceleration

It is widely accepted that the degrees of freedom (DOF) of a rigid body in a three-dimensional space is six but can be reduced if constraints are added. The 3DOF coupled mathematical model (heave-roll-pitch) and 4DOF coupled mathematical model (sway-heave-roll-pitch) are usually used for direct stability assessment of excessive acceleration. In the simulation, the Froude-Krylov and hydrostatics forces were calculated by integrating the incident wave pressure around the instantaneous wetted hull surface. Radiation and diffraction forces were calculated on the basis of the initial averaged submerged hull. Roll damping coefficients were obtained from the free roll decay motions.

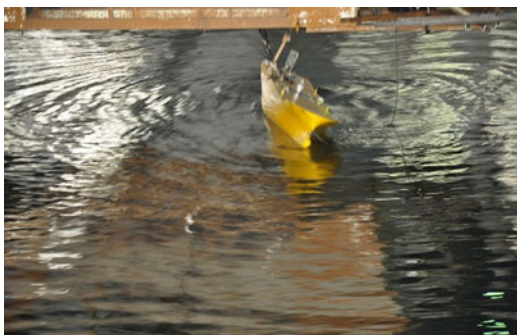
The schematic diagram of lateral acceleration in the calculation is shown in Fig. 3, and the formula is:

Excessive Accelerations, Table 1 Principal particulars of the sample containership

Items	Value	Items	Value
Length: (L_{pp})	251.88 m	Displ. (W)	27543.05ton
Breadth (B)	32.2 m	Draft (T)	6.225 m
Metacentric height (GM)	6.132 m	Natural roll period (T)	11.173 s

Excessive Accelerations, Table 2 Definition for three measuring points

No.	Coordinate		
	X(m)	Y (m)	Z (m)
P1	112.22	14.75	19.87
P2	54.35	−15.53	45
P3	0	−14.18	19.87



Excessive Accelerations, Fig. 2 Model test photo of the sample containership

Excessive Accelerations, Table 3 Wave conditions in irregular waves

No.		$H_{1/3}$ (m)	T_z (s)
1	ITTC 2-parameter spectrum	5.5	7.5
2			8.5
3		6.5	8.5
4			10.5

$$\begin{aligned} a_l &= (g - a_v) \sin(\varphi) + a_H \cos(\varphi) \\ a_n &= -(g - a_v) \cos(\varphi) + a_H \sin(\varphi) \end{aligned} \tag{1}$$

where, a_l and a_n are acceleration components along the deck direction and normal to the

deck, respectively. a_v and a_H are vertical and horizontal components, respectively.

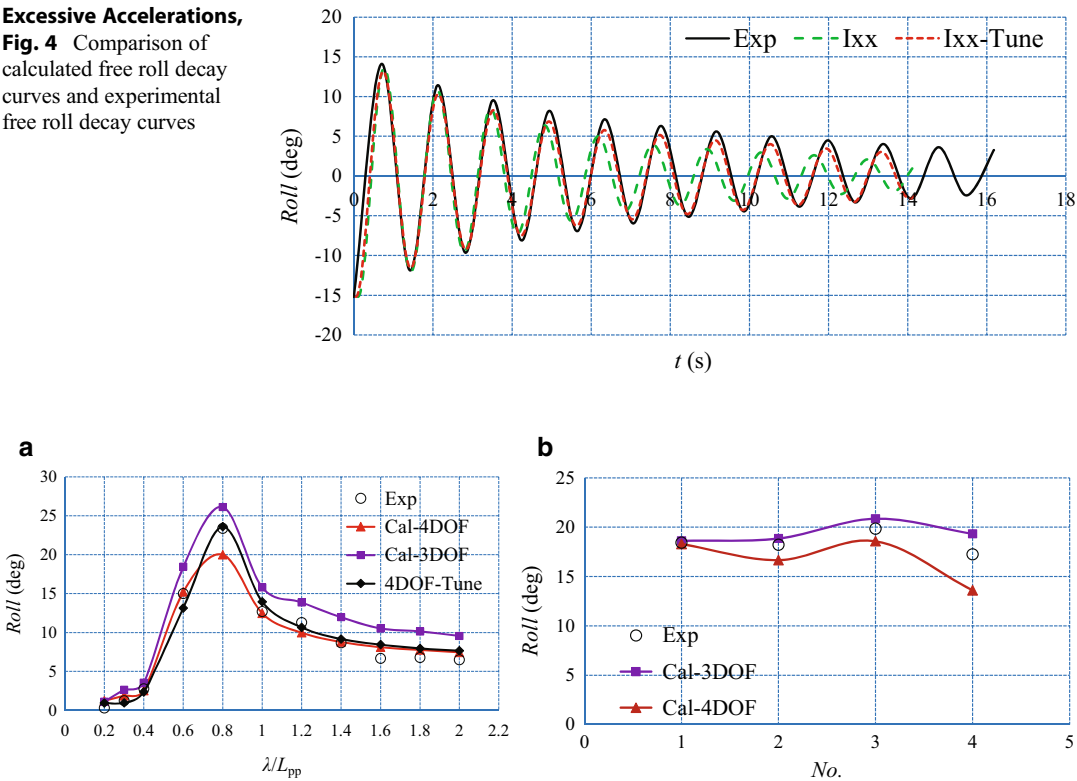
The Effect of Degree of Freedom on Excessive Acceleration

The effect of the degree of freedom on excessive acceleration is compared by taking the sample containership as an example. Firstly, the roll moment of inertia of the 4DOF coupled mathematical model is adjusted by using the free roll decay curve measured in the model test. The calculated free roll decay (Ixx-Tune) after adjustment is consistent with the free roll decay curve obtained by the test, as shown in Fig. 4.

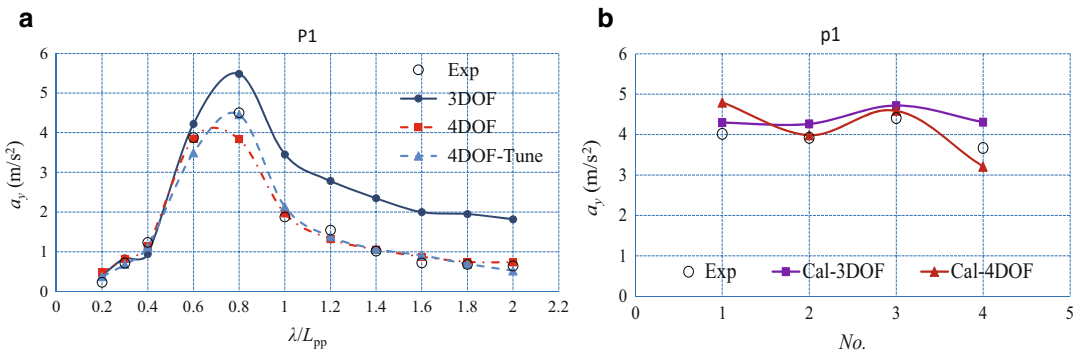
Figure 5 displays roll amplitudes under different wave conditions. Figures 6, 7, and 8 shows the lateral acceleration of three measuring positions in regular waves and irregular waves. The results show that the calculation results of the 4DOF

Excessive Accelerations, Fig. 3 Schematic for the calculation of lateral acceleration

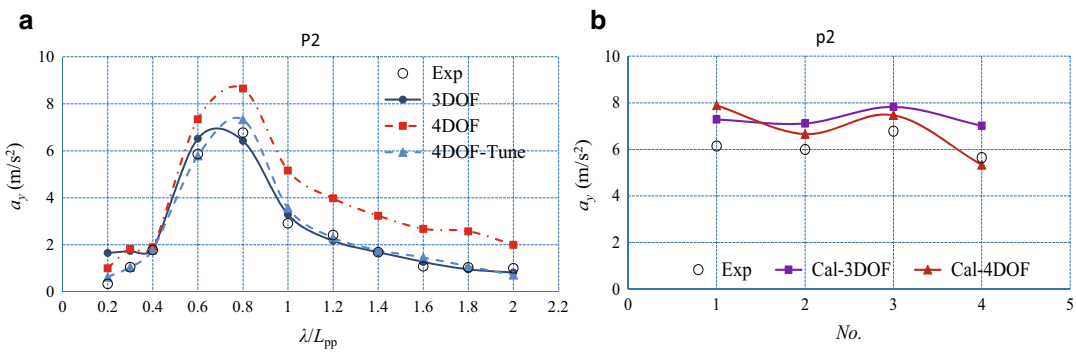
Excessive Accelerations, Fig. 4 Comparison of calculated free roll decay curves and experimental free roll decay curves



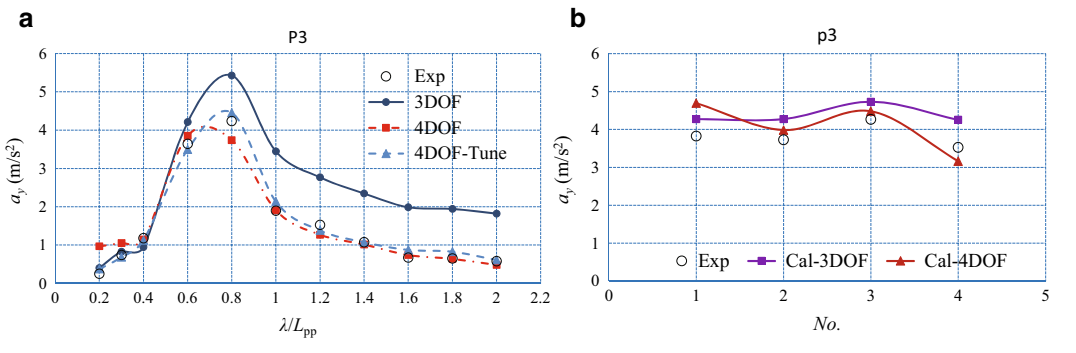
Excessive Accelerations, Fig. 5 Comparison of roll amplitudes in regular and irregular wave



Excessive Accelerations, Fig. 6 Comparison of maximum lateral acceleration at position p1



Excessive Accelerations, Fig. 7 Comparison of maximum lateral acceleration at position p2



Excessive Accelerations, Fig. 8 Comparison of maximum lateral acceleration at position p3

model agree well with the model test results, but the calculation results are slightly smaller than those of model test in several conditions. The calculation results of the 3DOF model are larger

than those of the model test results in the whole frequency range. The comparison results show that the roll moment of inertia has an influence on the accuracy of the 4DOF mathematical model.

The Effect of Calculation Position on Excessive Acceleration

The results of lateral acceleration are related to the ship roll amplitude and the position of the calculation point. The effects of the height of calculation position are studied by taking P2 as an example. Four calculation heights (H1 = 19.87 m, H2 = 30.73 m, H3 = 45.0 m, and H4 = 60.73 m) are chosen. From the calculation results shown in Fig. 9 (left), it can be observed that the lateral acceleration increases with the increase of the calculation height. Then draw the maximum lateral acceleration (resonance roll) corresponding to each height into Fig. 9 (right). It can be seen that the lateral acceleration basically changes linearly with the increase of height.

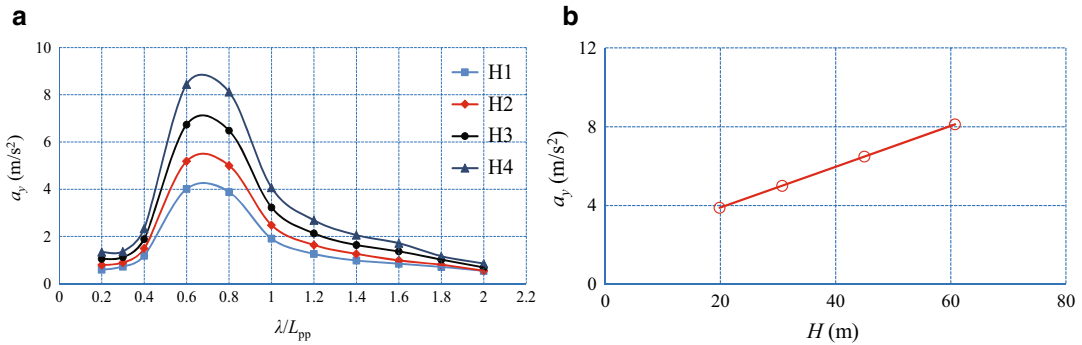
The different longitudinal positions and transverse positions are selected to calculate the lateral

acceleration at the roll resonance. The calculation results are shown in Fig. 10. Figure 10 (left) shows the change of lateral acceleration with longitudinal position and Fig. 10 (right) shows the change of lateral acceleration with transverse position. It can be seen that the change of longitudinal and transverse position has little effect on the lateral acceleration.

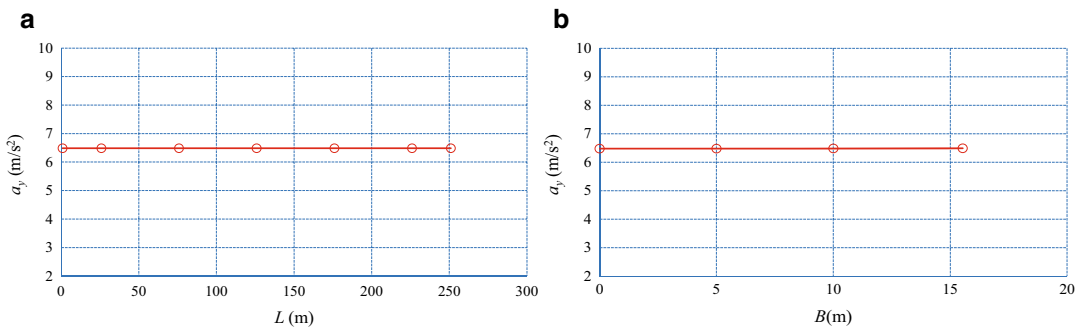
Simplified Method for Excessive Acceleration

The acceleration at any position of the hull can be calculated by the simplified formula (1), as proposed by IMO second generation intact ship stability criteria (IMO SDC 3/WP. 5 Annex 2)

$$a_{y_sim} = k_L (g \sin(\varphi_a) + h\omega_j^2 \varphi_a) \quad (2)$$



Excessive Accelerations, Fig. 9 Lateral accelerations along with the height of calculation position

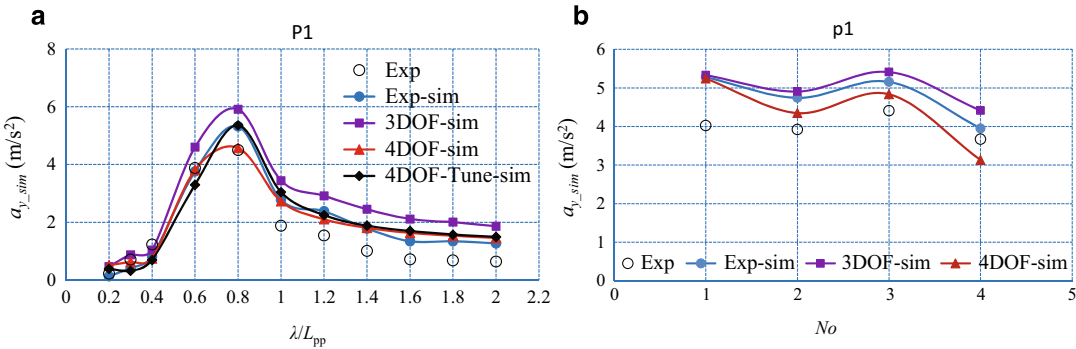


Excessive Accelerations, Fig. 10 Lateral accelerations along with longitudinal and horizontal positions

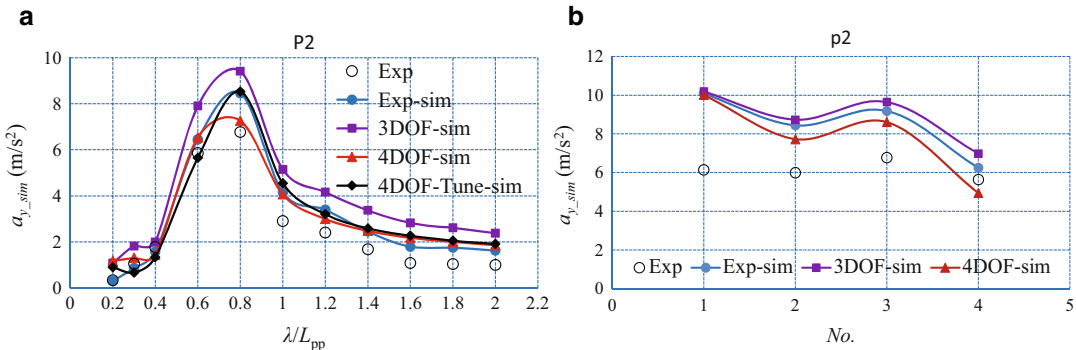
The reliability of this formula is verified by the comparisons between different methods and model tests results by taking the sample container-ship as an example.

The results are shown in Figs. 11, 12, and 13 in which the Exp represents the lateral acceleration measured by the model tests, Exp-sim represents

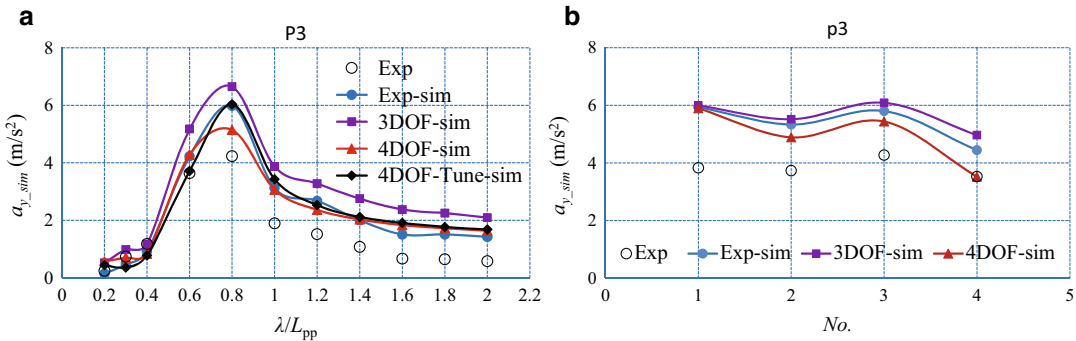
roll amplitude in formula (1) obtained from model tests, 3DOF-sim represents roll amplitude in formula (1) obtained from 3DOF coupled mathematical model tests, and 4DOF-sim represents roll amplitude in formula (1) obtained from 4DOF coupled mathematical model tests. We can see that the calculated results based on the simplified



Excessive Accelerations, Fig. 11 Comparisons of lateral acceleration by simplified method at position p1



Excessive Accelerations, Fig. 12 Comparisons of lateral acceleration by simplified method at position p2



Excessive Accelerations, Fig. 13 Comparisons of lateral acceleration by simplified method at position p3

formula for excessive acceleration are also in good agreement with the model test results, on the basis of accurate calculation of roll angle.

Cross-References

- ▶ [Dead Ship Condition](#)
- ▶ [Intact, Damage, and Dynamic Stability of Floating Structures](#)
- ▶ [Parametric Rolling](#)
- ▶ [Pure Loss of Stability](#)
- ▶ [Second Generation Intact Stability Criteria](#)
- ▶ [Surf-Riding and Broaching](#)

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- IMO SDC 3/WP. 5. Annex 2, Draft amendments to part B of the code with regard to vulnerability criteria of level 1 and 2 for the excessive acceleration failure mode

Experiment

- ▶ [Experimental Investigation of Offshore Renewable Energy](#)

Experimental Investigation of Offshore Renewable Energy

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Synonyms

[Experiment](#); [Model test](#); [Test](#)

Definition

Testing is a reliable approach to investigate the performance of a marine and offshore renewable energy system. The general concept of “testing” includes field testing and lab testing. Field testing is typically conducted in the oceans and employs a full-scale test model although the test model may be scaled occasionally. Comparatively, lab testing is usually model-scaled and carried out in experimental environment (wind tunnel, ocean basin, etc.) so that a scaling law must be developed and applied. When we refer to testing, it usually indicates the lab testing. The testing approach is commonly used when the physical response of the renewable energy system is too complicated to analyze numerically. A testing campaign may also be launched to validate numerical model.

Application of Testing in Renewable Energy Research

Test Facility

Unlike field test where wind, wave, and tidal current are available from the nature, lab test requires us to build up the experimental environment and generate the environmental conditions manually. Typical test facilities to build up the experimental environment are wave flume, wind tunnel, and ocean basin.

Wave Flume

Wave flume, or wave tank, is a laboratory setup used for the generation of surface ocean waves. Typical wave flume is a slender box filled with water, leaving open on top (see Fig. 1). Wave generation device is located at one end and the other end usually installs a wave-absorbing beach to eliminate the wave reflection. Wave flume is usually used in the model test of wave energy converter. Due to the geometry limit of wave flume in width, it is only applicable to the test of a single energy device. Nowadays, increasing attention is paid to the energy device array so that alternative facilities must be used, e.g., the ocean basin. Also, the wave in the tank only propagates along one single direction, meaning that only long-crested waves can be generated.



Experimental Investigation of Offshore Renewable Energy, Fig. 1 Wave flume at Kelvin Hydrodynamic Laboratory, UK (Kelvin Hydrodynamic Laboratory 2018)

Experimental Investigation of Offshore Renewable Energy, Fig. 2 NASA wind tunnel with the model of an aircraft (NASA 2017)



They can also be used to test tidal turbines, which are actually towed by a moving carriage to mimic the current effect.

Wind Tunnel

Wind tunnel is a tool used in aerodynamic research to examine the dynamic behavior of a solid object (a blade model, an aircraft model, etc.) in airflow. Fig. 2 illustrates the NASA wind tunnel. Wind tunnel is very similar to wave flume, consisting of a fan system to generate incoming flow at one end, a tubular passage and

flow-dissipating system at the other end. For decades, wind tunnel is mostly applied in the field of aerospace, and it is until the proposal of offshore wind turbine that it is introduced to the realm of offshore and marine technology. Nowadays, it is mainly used to measure the aerodynamic coefficients of a specific airfoil adopted by the offshore wind turbine. Cavitation tunnel can be seen as another kind of wind tunnel. In a cavitation tunnel, the fluid is water. Cavitation tunnel is mostly used in the model test of tidal turbine.

Ocean Basin

Ocean basin is probably the most representative facility in offshore and marine technology. It is basically a wave tank with width and length of comparable magnitude but distinguished by its wave generation ability. Ocean basin is able to generate multidirectional waves, multidirectional currents, and interactions of them. Wind generation system may also be installed to model offshore wind. Current generation system is frequently equipped to simulate tidal current. Fig. 3 sketches the ocean basin at Shanghai Jiao Tong University. Ocean basin can be used for test of various energy devices, ranging from offshore wind turbine, wave energy converter, wind turbine, or combination of them.

Scaling Law

The energy device in a test is typically scaled down from the prototype, even in a field test. It is straightforward to scale the device according to geometry similitude. For example, the prototype is a box with dimension $10\text{ m} \times 10\text{ m} \times 10\text{ m}$ and the model can be scaled down to $1\text{ m} \times 1\text{ m} \times 1\text{ m}$ with a scaling ratio of 1:10. Then, we let the prototype moves with a speed of 1 m/s. What the scaled speed in the test should be? 1/10 m/s, is it right? In practice, the scaling algorithm we apply in a model test could be far more complicated. In addition to geometry similitude, we must also maintain dynamic similitude. Depending on the

dominating load in the test, we introduce Reynolds number (Re), Froude number (Fr), and Euler number (Eu) to fulfill such goal.

$$Re = \frac{\rho u L}{\nu}$$

$$Fr = \frac{u}{\sqrt{gL}}$$

$$Eu = \frac{2\Delta p}{\rho u^2}$$

where u is the speed, L is a characteristic length dimension, ρ is the fluid density, ν is the dynamic viscosity of the fluid, g is the acceleration of gravity, and Δp is the gradient of fluid pressure.

Reynolds Number

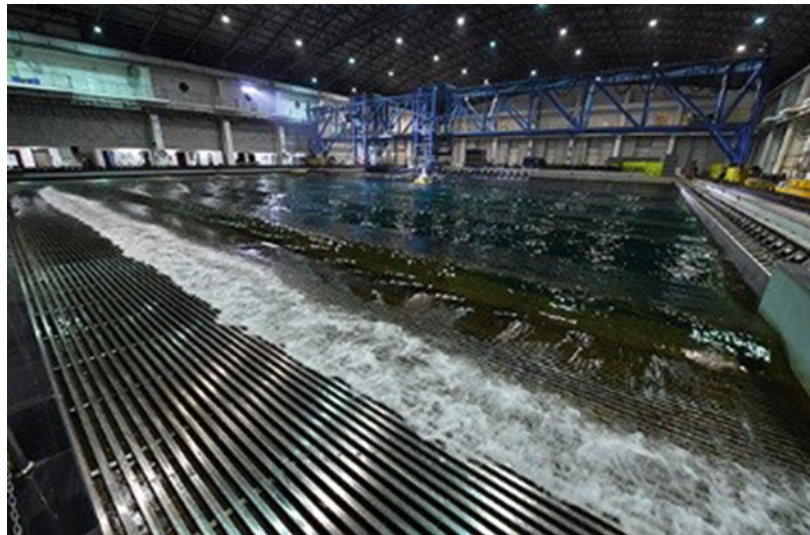
Reynolds number represents the ratio of inertial forces to viscous forces within a fluid (water or air). In low number case, sheet-like flow dominates. Otherwise, turbulence caused by the difference in the fluid's direction and speed is significant. If viscous force dominates the physical process, we tend to maintain Reynolds number. It is the case in the model test of wind turbine and tidal turbine.

Froude Number

Froude number represents the ratio of inertial forces to gravity. Froude number is maintained

Experimental Investigation of Offshore Renewable Energy,

Fig. 3 Deepwater offshore basin at Shanghai Jiao Tong University, China (State Key Laboratory of Ocean Engineering 2018)



when the inertial fluid forces play an important role. More specifically, we maintain Froude number if the wave force is the primary excitation load. Model test of wave energy converter adopts the Froude number similitude. Similarly, model test for hydrodynamics of floating wind turbines follows the Froude scaling law.

Euler Number

Euler number describes the ratio of inertial forces to pressure gradient. Euler scaling law is mainly used in the experimental investigation of cavitation and noise of tidal turbine so that it is sometimes named as the cavitation number. Euler scaling law is not often used in the model test is offshore renewable energy system.

Test Campaign

To deepen our understanding on the dynamic response of offshore and marine renewable energy system as well as to validate a suite of benchmark models and simulations, several test campaigns (both field test and lab test) have been launched.

Offshore Wind Turbine

In 2009, Equinor (formerly Statoil) launched the Hywind demo project (see Fig. 4), which is the first full-scale offshore floating wind turbine prototype in the world (Equinor 2017). The 2.3 MW Hywind concept (see Fig. 5), supported by a spar-type floating platform, is located 12 km off Norway's south-west coastline. The floating system is anchored to the seafloor with three mooring lines. Just 1 year later, Principle Power set up the field test of a full-scale floating wind turbine concept – WindFloat (Principle Power 2015). A 2 MW wind turbine is mounted on a semi-submersible floating platform, which is installed 5 km off the Portuguese coast. In 2017, the world's first floating wind farm – Hywind Scotland was opened about 30 km off the coast of Scotland. The Hywind Scotland earns experiences and technology from the Hywind demo project to provide power to the national grid. Since the above field tests are managed by industrial sector, the measured data are hardly open to the public.

In addition to field test, lab tests have been insensitively carried out in the academic



Experimental Investigation of Offshore Renewable Energy, Fig. 4 Hywind demo concept (Equinor 2017)



Experimental Investigation of Offshore Renewable Energy, Fig. 5 WindFloat concept (Principle Power 2015)

community. The OC4-DeepCwind test campaign is launched under the collaboration of a couple of participants. The lab test was performed at Maritime Research Institute Netherlands to assess the unique behavior of various platform concepts for floating wind turbine. This test campaign was primarily set up to generate benchmark experimental data to verify fully coupled tools for offshore floating turbine. The data recorded in the test are publicly available. In lab test, we have to focus on the major factors and ignore the minor factors. In most cases, we focus on wind load and wave load whereas structural dynamics can be somewhat simplified. It is ideal to scale aerodynamic and hydrodynamic loads accurately at the same time. Unfortunately, we cannot maintain Reynolds number and Froude number simultaneously. Please recall the question at the beginning. If we are to maintain Reynolds number, the scaled speed of the prototype should be 10 m/s. The speed is scaled down to 0.32 m/s if Froude number is maintained. What we typically do is maintaining Froude number to scale wave load properly and develop alternative approaches to acquire the desired wind load. For example, Martin et al. (2014) corrected Reynolds number effect by roughening the leading edge of the blades. Other research even abandoned the blades and used a disc to produce the wind load.

Wave Energy Converter

Compared with offshore wind turbine, wave energy technology appears to be less popular, but a large number of field tests have been conducted so far. The Wave Dragon concept (Wave Dragon ApS 2013), the world's first offshore grid-connected wave energy device, was first tested in 2004. This test unit supplied electricity to the grid for more than 20,000 hours. The Pelamis project (European Marine Energy Centre 2018) was launched by Pelamis Wave Power in 2014, which successfully provided power to the national grid of Scotland. The Pelamis is the world's first offshore wave power converter to successfully generate electricity into a national grid. However, most of the field tests did not last for a long duration, typically less than 1 year. Most recently, wave energy converter lab test begins to put its focus on array effect. When a set of wave energy converters are displaced together to constitute an array, the hydrodynamic interaction between them will have a significant influence on power production. Stratigaki et al. (2014) measured the dynamic response of wave energy converter for a range of geometric layout configurations at a water basin to provide data for understanding array hydrodynamic interactions. Fig. 6 gives a sketch of the wave energy converter array. The



Experimental Investigation of Offshore Renewable Energy, Fig. 6 Model test of wave energy converter array (Stratigaki et al. 2014)

recorded data may also be used to validate array interaction numerical models.

Tidal Turbine

Tidal turbine is very similar to a land-based wind turbine. Tidal turbine test is typically conducted in a wave tank or a cavitation tunnel. Since most industrial tidal turbine products are fixed to the seafloor, tidal turbine model test mostly focus on thrust force and power capture. Batten et al. (2007) measured steady power and thrust coefficients of a 1/20th scale tidal turbine. Recently, the research community shows great interest in the tidal turbine performance under wave-current interaction. In this circumstance, the model test must be performed in a wave flume or an ocean basin. Relevant work on this specific field was reported by Luznik et al. (2013).

Hybrid System

Hybrid energy system is proposed as a technical solution to the efficient exploitation of offshore and

marine energy recourses. The hybrid concept is mainly proposed for two objectives: (1) increase the overall power production; (2) enhance the dynamic performance (motion, structural load, etc.) via the combination of different offshore renewable energy devices. Until now, no field test is available although some lab test measurements have been reported. Wan et al. (2016) conducted a series of tests to assess the dynamic response of the STC, a combination of a spar-type floating wind turbine and a heaving-body wave energy converter. The SFC concept integrates a semisubmersible-type floating wind turbine and a flap-type wave energy converter. Michailides et al. (2016) experimentally measured its dynamic performance in the ocean basin in Ecole Centrale Nantes.

Cross-References

- [Analysis of Renewable Energy Devices](#)
- [Design of Renewable Energy Devices](#)

- Offshore Wind Turbines
- Tidal and Ocean Current Turbines
- Wave Energy Converters

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Experimental Techniques for VLFS

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Definition

Model test plays a very important role in the research process of very large floating structures (VLFS). Many challenges on the experimental techniques are encountered due to the large dimensions compared with the conventional marine structures. Firstly, due to the large horizontal dimensions of the VLFS, ocean environment has great changes in the area over the structure. There are a lot of problems unsolved in the experimental research of nonuniform ocean environment. Secondly, because of the extremely large dimension of VLFS, a large-scale ratio has to be adopted in the model tests, which could induce an obvious scale effect and thus make a great difference on the test results. In addition, in order to simulate and measure the hydroelastic properties of the VLFS accurately, the test design and measuring system are more complicated than that for conventional marine structures, which can be regarded as rigid bodies.

Scientific Fundamentals

Historical Development

Model Experiment on Hydroelasticity of VLFS

With the development of theoretical researches, experimental research on VLFS is also developing very fast. Many countries, such as Japan (VLFS '96), the USA (Ertekin and Riggs 1991, 1999), and Norway (Faltinsen 1996) among others, have paid much attention to the experimental research on VLFS, and they have made great progress.

Yago and Endo (1996a, b) carried out a model test in the basin of Ship Research Institute in Japan to study the hydroelastic behavior of VLFS. A box-typed model, which consists of two layers, is used. One layer is an aluminum alloyed honeycomb plate and the other is a polyethylene foam plastic plate. The two layers are agglutinated together. The model's dimension is $L \times B \times D \times d = 9.75 \text{ m} \times 1.95 \text{ m} \times 5.45 \text{ cm} \times 1.67 \text{ cm}$, and the scale ratio is 1:30.77. Where L is the model length, B the width, D the depth, and d the draught. The basin is 40 m long and 27.5 m wide. Four wave incidence angles are adopted: 0° , 30° , 60° , and 90° . In the model test, the wave period changes from 0.8 s to 2.5 s with four different wave heights: 1 cm, 2 cm, 6 cm, and 7 cm, respectively.

Ohmatsu and Namba (1996) also carried out hydroelastic, forced and free oscillation tests on a box-typed VLFS. The model's dimension is $L \times B \times D \times d = 50 \text{ m} \times 5 \text{ m} \times 4.4 \text{ cm} \times 1.0 \text{ cm}$, with a scale ratio of 1:100. It is made up with two aluminum plates with 1 mm in thickness. Polyethylene foam with 2 mm in thickness is filled in between the two aluminum plates to provide the rigidity needed. Below the plate, polyethylene foam with 4 cm in thickness is also agglutinated. The water basin is 140 m long and 7.5 m wide with water depth 0.45 m and 3.0 m, respectively, corresponding to the full scale of 45 m and 300 m. The purpose of the model tests is to observe and measure the hydroelastic responses of VLFS and determine its natural frequency, viscous damping coefficient, and the wave drift forces.

Shiraishi et al. (1996) (in VLFS '96) carried out a model test using six models with different widths, masses, and bending rigidities in a $19.0 \text{ m} \times 10.0 \text{ m}$ water basin. By measuring the motions and deformations, the hydroelastic performances of a box-typed VLFS are studied. It is shown that models with small bending rigidities have larger vertical displacement than those with greater bending rigidities, and all of these six models have large longitudinal displacement in waves.

Shin et al. (1999) carried out a model test, taking the model of the Technological Research Association of Mega-Float as the prototype. The dimension of the model is $L \times B \times D \times d = 7$

$\text{m} \times 1.4 \text{ m} \times 4.667 \text{ cm} \times 1.167 \text{ cm}$, and the scale ratio is 1:42.8. In the test, water depth is 18.7 cm and 250 cm, respectively, corresponding to the full scale of 8 m and 107 m. The model is made of foam plastic with alloyed frame. The foam is composed by many cells. Between every two adjacent cells there is a 3 mm wide space to avoid the influence of foam material on model's rigidity. The test results indicate that with the increase of incident wave length, the amplitude of VLFS deformation also increases. And the amplitude of the responses in shallow water is relatively smaller than that in infinite water depth. It also seems that the water depth has significant effect on the structure responses of VLFS in waves.

In 1999, Kagemoto et al. carried out a model test on hydroelastic response of VLFS (VLFS '99). The prototype is 5 km long, 1 km wide, and its depth is 5 m. The model is 3 m in length, 1 m in width, and 0.03 m in depth. In addition, the wave generator is able to generate waves with only 1 mm wave height and 10 Hz in frequency. And with optical technology and Moire method, the water surface and vertical displacements could be measured continuously during the model test.

A Mobile Offshore Base (MOB) model test has been carried out in Mc Dermott's in the US Naval Surface Warfare Center by Wu and Mills (1996). The model scale is 1:59. In the test, the model is connected by one, two, and five modules, respectively. The motions of the model, connector loads, and the mean drift forces in operational condition are measured. Results of the experiment are compared with that of calculations. According to the comparison, it shows good agreement while the number of modules is small, but with the increase of module number, the difference becomes increasingly great.

Model Experiment on Hydroelasticity of VLFS in Nonuniform Ocean Environment

Because of the large horizontal dimensions of the VLFS, ocean environment have great changes in the area over the structure. Therefore, uniform wave spectrum, which is based on the hypothesis of ergodicity and stationary stochastic process in both space and time, is not applicable here.

Instead, a new nonuniform wave spectrum varying with location needs to be established (Cui et al. 2001). However, most of the present researches on hydroelasticity of VLFS are based on the hypothesis of uniform ocean environment. Therefore, experimental research of VLFS in non-uniform ocean environment is quite essential.

There are a lot of problems unsolved in the experimental research of nonuniform ocean environment. First of all, because of the extremely large dimension of VLFS, and the topography variation of ocean bottom needed to be simulated in model basin, large-scale ratio has to be adopted in the model tests. Thus, the scale effect may have great influence on the model test results. Besides, there are few model tests on nonuniform ocean environment reported publicly. To describe the nonuniform ocean environment and its influence on motion and deformation of VLFS properly, it may be necessary to put forward a method to analyze and express the model test results.

Model tests of VLFS in nonuniform ocean environment have been performed in the model basin of State Key Laboratory of Ocean Engineering, SJTU (Lu et al. 2004). The VLFS model in these tests is box-typed. Its dimension is $L \times B \times D \times d = 10 \text{ m} \times 0.6 \text{ m} \times 3 \text{ cm} \times 1 \text{ cm}$, and the scale ratio is 1:100. The water depth changes between 0.2 m ~ 0.4 m. The model is made up with two layers. The upper layer is an aluminum plate 2.5 mm in thickness, while the lower one is foam plastic plate 27.5 mm in thickness. The two layers provide the required bending rigidity and buoyancy, respectively.

In the experiment, nonuniform ocean environment is simulated by setting a shoal model on the bottom of the basin. Altogether there are three shoal models with different shapes and dimensions used in the model tests. Two of them are basically 2-D shoals with different dimensions, and the third one is a 3-D shoal model. The 2-D shoal is a column with an elliptic section, while the 3-D one is a part of an ellipsoid.

The model test results show that the varying depth bottom of the ocean has great influence on the deformation of VLFS in shallow water. Compared with that of uniform ocean environment, VLFS has larger vertical displacement at the

location above the shoal. Moreover, the longer wave period, the smaller water depth, and larger shoals on the bottom have greater effect on VLFS.

Mooring System Experiment of VLFS

In exploitation of ocean space resources, VLFS are commonly moored to an offshore location for a long period over 100 years. Consequently, its mooring system has to face many problems. During the design of VLFS mooring system, the integrate effect of wind, wave, and current should be considered carefully.

The involved problems mainly include the influence of hydrodynamic forces, external forces, etc. With large horizontal dimension and relatively small draft, the horizontal motions of VLFS are quite different from ordinary floating structures. Since breakwaters are often set around VLFS for near-shore applications, the shape and propagation of waves are altered. And because of the asymmetry of VLFS superstructure, external forces acting on VLFS are also asymmetrical. These forces and the effect of wind and current also increase the difficulty of the mooring system design (Kinji et al. 1996).

The purpose of mooring system experiment is to study the forces on it. Ministry of Communication in Japan carried out a model test (Kato et al. 1996) to study the mooring system's reliability in the case that one dolphin of the multipoint mooring system fails.

In Kato's experiment (Kato et al. 1996), the VLFS model is box-typed. It has a dimension of $L \times B \times D = 10 \text{ m} \times 2 \text{ m} \times 0.04 \text{ m}$ with the scale ratio of 1:50. The model is made of vinyl foam. The mooring system is composed of link devices and dolphins. The link devices operate with a fracture mechanism, which is able to disconnect when the compress load reaches a certain level. Springs are installed on the dolphins to simulate mooring stiffness. Universal joints are set at two ends of the link device, and they are able to rotate freely.

The model is fixed by seven dolphins. Four of them are set along the longitudinal side, while three others are located along the transverse side. Motions of VLFS are measured by two CCD cameras and LED (light emission diodes).

Accelerations at mooring point and forces on each dolphin are measured by accelerometers and force transducers, respectively.

Experimental Techniques of VLFS

The Law of Similarity

Ocean engineering model tests are commonly carried out in a basin including the following respects: At first, the characteristics of the prototype and the environment conditions must be simulated correctly. Then test data are acquired with scientific and effective measurement method. And the data acquired are carefully analyzed to achieve the conclusion needed.

Model tests use the physical model to obtain data needed for the design and construction of the ocean engineering structures. Therefore to obtain the correct information from the collected data, model test results must be transformed to corresponding values of the prototype. And this transformation must follow the law of similarity.

It is known that if all the parameters measured from one system (linear dimension, velocity, force, moment and time interval, etc.) could be achieved from another system by multiplying certain constant numbers, then the two systems may be regarded as similar (Liu et al. 1990). Mathematically, it can be expressed as:

$$\frac{L''}{L'} = C_L, \frac{V''}{V'} = C_V, \frac{T''}{T'} = C_T \dots$$

where, L, V, T – local or characteristic parameters relative to physical process;

Superscripts ' and '' indicate the parameters corresponding to the first and second system respectively;

C – constant similarity coefficients.

If the two systems are similar in physics, the law of similarity in kinematics and dynamics must be satisfied, i.e., the two systems are not only similar in geometry, but also in velocities, forces, and other physical constants in the problem. For vectors, such as velocity and force vectors, their directions at corresponding points must be consistent, in addition to their absolute values proportional to each other.

For hydroelasticity problems in ocean engineering, we know already the mathematical model to describe the interaction between structure and flow around it both for the model scale and full scale, therefore, we can obtain the similarity law from the formulation of the problem.

For simplicity and clarity, we use Timoshenko nonuniform beam or von Karman's isotropic thin plate as the structural model, and the motion equations of the freely floating beam or the plate can be taken as the dynamic boundary condition on the wetted body surface. If the two flow fields and structures for model and prototype are similar, the formulation of the problem posed above would be suitable for both of them and is complete.

For similar flow issues, we can use the relations given above to transform one system to another. The governing equation, kinematic boundary condition, seabed boundary conditions, and radiation conditions are self-similar, no required restriction is obtained. Thus, we need only focus our attention on the free surface condition and the motion equations of Timoshenko beam or von Karman plate.

Table 1 lists the characteristic variables between model and prototype usually used in ocean engineering experiment.

Design and Construction of VLFS Model

Before model tests, the scale ratio of model should be determined to obtain the test data needed. The scale factor is decided based on the balance among the wave-making condition of the basin

Experimental Techniques for VLFS, Table 1 Variables between model and prototype

Item	Symbol	Scaling ratio
Linear dimension	L_p/L_m	λ
Linear velocity	V_p/V_m	$\lambda^{1/2}$
Linear acceleration	a_p/a_m	1
Angle	φ_p/φ_m	1
Angle velocity	φ'_p/φ'_m	$\lambda^{-1/2}$
Period	T_p/T_m	$\lambda^{1/2}$
Area	A_p/A_m	λ^2
Volume	∇_p/∇_m	λ^3
Moment Inertia	I_p/I_m	$\gamma \lambda^5$
Force	F_p/F_m	$\gamma \lambda^3$

(the range of wave length L_w and wave height H), the required test condition of wave length L_w /ship length L_s , wave height H /ship length L_s , the possibility of reducing the structural rigidity of the model to the factor of $1/\lambda^5$, and the measurable strain range of the model in test wave conditions. Obviously, it is impossible to reduce the thickness of hull plate of the model exactly to the factor of $1/\lambda$, and to find the material with the Young's modulus of $E_m = E/\alpha$. Therefore, to simulate the global effect, the detailed layout and dimensions of the plates and stiffeners should be well designed to allow for the resultant flexural rigidity $E_m I_m$ and torsional rigidity $G_m L_{cm}$ satisfying the similitude conditions.

While transforming the test data from model scale into the full scale, the laws of similarity must be satisfied. To what extent the similarity conditions should be satisfied depends on the purpose of the test. To examine the loads and the structural responses of the structure in waves, no matter the interest is focused on the global reactions or local stresses, the geometry of global similarity of wetted surface, the hydrodynamic similarity of global motions, and the similarities of mass and stiffness distributions of the hull girder along the length, together with the radius of longitudinal gyration, must be satisfied (Wu et al. 2003).

In practice only part of the similitude conditions may be held, especially when an entire ship hull is to be modeled to the small size, due to the restrictions and limitations arising from the available materials and possible dimensions of the model.

During the wave tests, the gross weight of model should remain constant. So after model tests the model weight should be checked to ensure that the error between two measurements is less than 0.5%.

The gross weight of VLFS model consists of two parts: the VLFS model weight and the ballast weight. It should be noted that the model weight must be less than the gross weight. So during VLFS model construction, the model weight should be controlled carefully. After the model being constructed, the load conditions, the position of the center of gravity, longitudinal radii of gyration, and rolling periods can be adjusted at the

trimming table by adding or subtracting and/or shifting the weights so as to meet the specified requirements. For box-typed VLFS it can be regarded as an elastic body, and its longitudinal bending rigidity must be measured before storm tests and be transformed into full scale to simulate the prototype correctly.

After the model is adjusted, the weights should be numbered and fixed to ensure that the positions of the weights remain motionless during tests.

Main Facilities Needed in VLFS Model Tests

Main facilities needed in VLFS model tests include wave generator, wave absorbing beach, towing carriage, and noncontact optical systems for measuring motion, etc.

Wave Generator There are many types of wave generators all over the world. In VLFS tests in SJTU, a dual flap type hydraulic wave generator is adopted. The dual flap type hydraulic wave generator is made up of two flaps. One of the flaps is immersed in water and connected with the pedestal and swings around it. Another upper flap can swing around the fulcrum at the joint with the lower flap. The motions of two flaps are controlled by computers. Then complex waves of long crest can be simulated in water basin.

This wave generator is relatively advanced. But it is also complex and expensive. And it is also difficult for maintenance. In addition it must be water proof between the generator and the tank wall in order to avoid the water flowing behind the back of flaps.

Wave Absorbing Beach In order to absorb waves and avoid wave reflection, wave absorbing beach is set along the side opposite to the wave generator. There are many types of wave absorbing beaches in which the most common type is the parabola inclined plate. The gradient of inclined plane influences the wave absorbing performance directly. The wave absorbing beach with small gradient has better performance, but it will also cover more length of the basin. In addition the length of wave absorbing beach should not be too long. In order to obtain better performance, other types of wave absorbing beaches, such as dual

layers and multilayers are also adopted in some basins.

Moreover the wave absorbing beach should be easy to move, i.e., it would be not too large or heavy.

Noncontact Optical Systems for Measuring Motion Motions of VLFS model with six degree of freedom can be measured by the noncontact optical system. The system uses light emitting diodes and position sensitive detectors to measure the positions of several points of the model. Several LED are installed on the model and their motions are measured by noncontact optical system. Signals acquired are separated into data in 2-D multichannels, i.e., the projection location of LED (Yang et al. 1999).

This noncontact measuring system measures motions of VLFS model without forces acting on the model. Thus high precision can be obtained.

Examples of VLFS Model Tests

In this section, three model tests of VLFS are given. All these tests are carried out in the State Key Laboratory of Ocean Engineering, SJTU. The water basin is 50 m long, 30 m wide, and the main facilities include dual flap type hydraulic wave generator and large area false bottom. Water depth of the basin can be adjusted from 0 m to 5 m. The wave absorbing beach is located opposite to the wave generator and its performance is quite satisfactory.

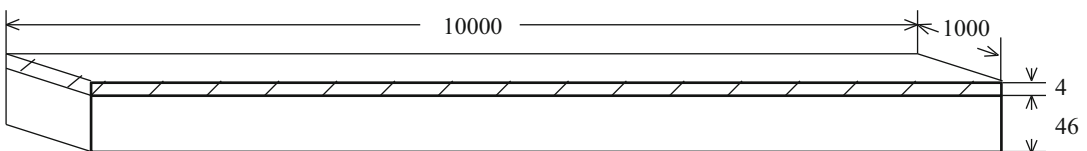
Hydroelastic Model Test on Box-Typed VLFS (Chen et al. 2003; Chen 2003)

Design of the Test Model The VLFS model is of box-type, and its scale ratio is 1:100. The model is 10 m in length, 1 m in width, 0.05 m in depth, and

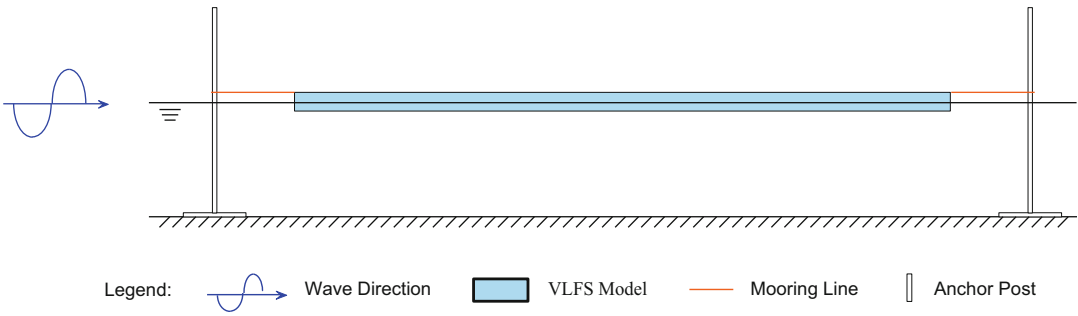
0.02 m in draft. Its bending rigidity is $EI = 5.79 \times 10^2 \text{ N} \cdot \text{m}^2$. According to the scale ratio, its prototype is 1000 m long, 100 m wide with a depth of 5 m. And the bending rigidity is $EI = 5.79 \times 10^{12} \text{ N} \cdot \text{m}^2$. The model consists of two layers. The upper layer is an aluminum plate with 4 mm in thickness, which simulates the rigidity needed and the lower layer is made of foam plastic with 4.6 cm in thickness supplying the required buoyancy. Outline of the model is showed in Fig. 1. Since its ratio of thickness to length is only 0.5%, so the influence of inhomogeneity due to different materials may be ignored. The model's exterior surface is covered with a waterproof layer, and its water permeability is about $1.4954 \times 10^{-5} \text{ kg}/(\text{m}^3\text{s})$.

Test Program The test mainly includes two parts: one part is the experiment of hydroelastic responses in regular waves and the other is the effect of water depth on hydroelastic behavior of VLFS. For the experiment in regular wave, water depth is 0.7 m, corresponding to the full-scale water depth of 70 m. Wave height is 2 cm, and the corresponding real wave height is 2 m and the wave period is 0.8, 1.13, 1.6, 1.96, 2.26, and 2.53 s, respectively, corresponding to the real wave period of 8, 11.3, 16, 19.6, 22.6, and 25.3 s. The ratio of wave length to the length of VLFS is, respectively, 0.1, 0.2, 0.4, 0.6, 0.8, and 1.0. Wave incidence angles are 180°, 165°, and 135°. The tests for the effect of water depth are carried out in regular waves. Wave direction is 180° only, and the ratio of wave length to VLFS length is limited to 0.4. Water depth is 0.2 m, 0.45 m, 1 m, and 2 m, respectively, corresponding to the wave depth of 20 m, 45 m, 100 m, and 200 m in full scale.

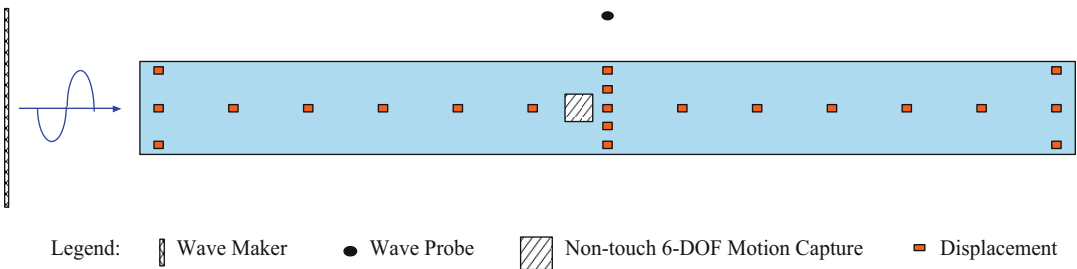
In this test VLFS model is moored as is shown in Fig. 2. The test setup is shown in Fig. 3.



Experimental Techniques for VLFS, Fig. 1 The VLFS model (unit: mm)



Experimental Techniques for VLFS, Fig. 2 Mooring of the VLFS Model



Experimental Techniques for VLFS, Fig. 3 The Setup of the Model Test

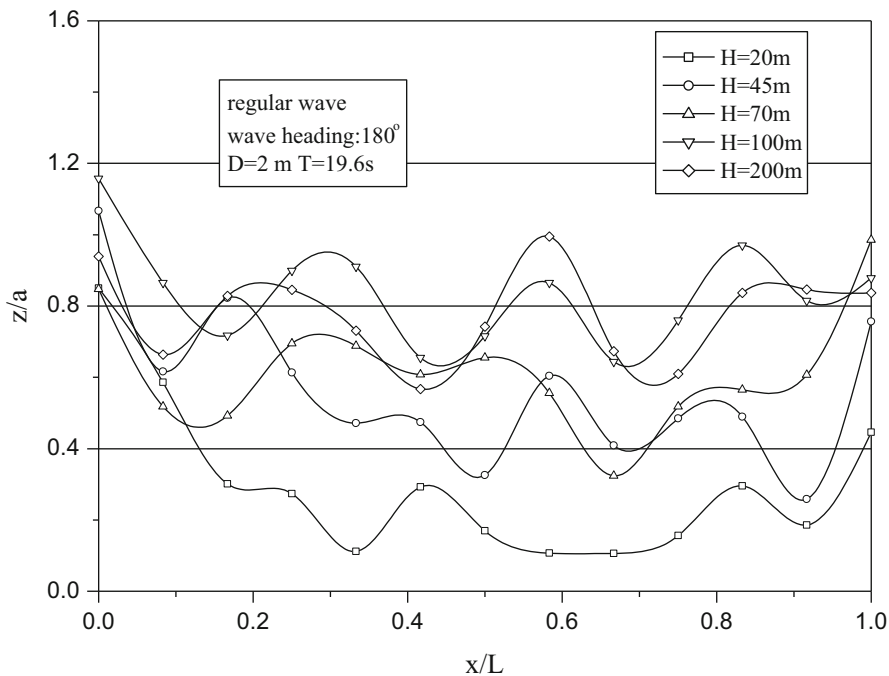
Altogether there are 21 displacement sensors placed on the model in longitudinal direction, 13 of them are placed in the mid-line of model. For the further research of transverse transformation, 5 others are placed along the transverse direction. And there are 4 sensors located at 4 corners of VLFS, respectively. The range of displacement sensor for measuring is about 0 ~ 10 cm and the nonlinearity is less than 0.2%. The six degree-of-freedom of VLFS model motions are measured by the noncontacted optical system. Wave height and period are acquired by the wave probe of resistance type. The effective duration of each wave test is 10 min at least in model scale corresponding to 100 min in full scale. All of the measurements are recorded in computer with a sampling rate of 20 per second. Part of the experiments results are shown in Fig. 4.

Model Test on Hydroelastic Responses of VLFS in Nonuniform Ocean Environment (Lu et al. 2004)

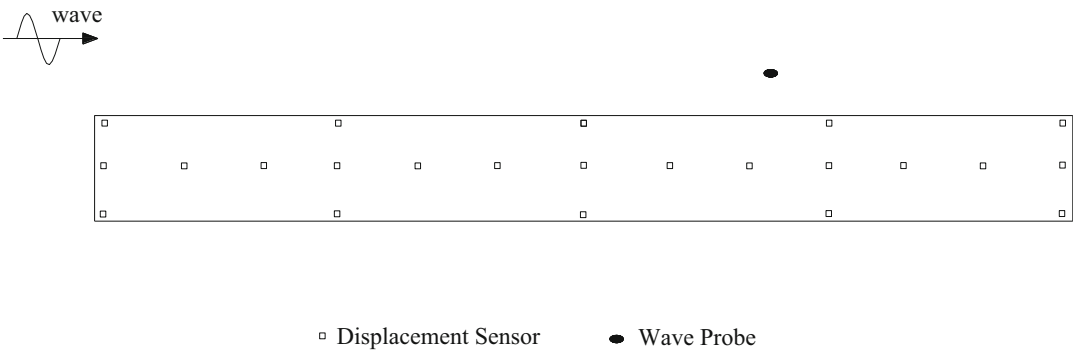
Design of Test Model The model used in this test is similar to the one described in the last

section. It also contains two layers with different materials. The upper one is a 2.5 mm aluminum layer, used to simulate the rigidity of the prototype, and the lower is a 27.5 mm foam plastic layer, supplying needed buoyancy. The VLFS model is 10 m long, 0.6 m wide, and 0.03 m in depth with a 0.01 m in draft. Its bending rigidity (EI) is $EI = 1.2573 \times 10^2 \text{ N} \cdot \text{m}^2$. While the full scale is 1000 m in length, 60 m in width with 3 m in depth and a 1 m in draft. Its bending rigidity is $EI = 1.2573 \times 10^{12} \text{ N} \cdot \text{m}^2$. In order to keep the weight of the model constant in storm tests, the surface is covered with waterproof layer, which has satisfactory performance of waterproof.

Some shoal models are set on the bottom of the basin to simulate the nonuniform ocean environment (see Photo 3 and Photo 4). Altogether there are three shoal models adopted in this experiment, which have different shapes and dimensions. Two of them are 2-D shoals with different dimensions, and the third is a 3-D shoal model. The 2-D shoal is a column with an elliptic section, while the 3-D one is a part of an ellipsoid.



Experimental Techniques for VLFS, Fig. 4 Distribution of maximum vertical displacements in different water depth (D -Draft, H -Water depth, T -Wave period, a -Wave amplitude, z -Vertical displacement)



Experimental Techniques for VLFS, Fig. 5 Arrangement of displacement sensors on VLFS model

Test Program VLFS is moored in the same way as shown in Fig. 2. There are 23 displacement sensors placed on the VLFS model. In the longitudinal direction, 13 displacement sensors are ranged in the mid-line of model, and there are 5 sensors located along the two sides. Motions and deformation are studied by measuring the

vertical displacement of every part of VLFS. The arrangement details are shown in Fig. 5.

Different wave incidence angles, water depths, and regular wave lengths are adopted in the experiment, and the relative position between shoals and VLFS model is also altered to study the influence of varied environmental conditions on the

hydroelasticity of VLFS. Wave height in model scale is 2 cm with the wave length in the range of 0.5 ~ 4.0 m, corresponding to the full scale 50 ~ 400 m respectively. The tests are run in waves with 180°, 150° and 120° headings. To study the effect of water depth on VLFS, water depth in experiment varies from 0.2 ~ 0.6 m corresponding to the real water depth of 20 ~ 60 m.

Multi-Module Test of Semi-Submersible VLFS (Yu et al. 2004)

Model Design A semi-submersible VLFS model is used in this test. The whole VLFS model is flexibly connected by three rigid modules (RMFC). This kind of connector system allows modules relative angular displacements in three directions, but their relative linear displacements are limited.

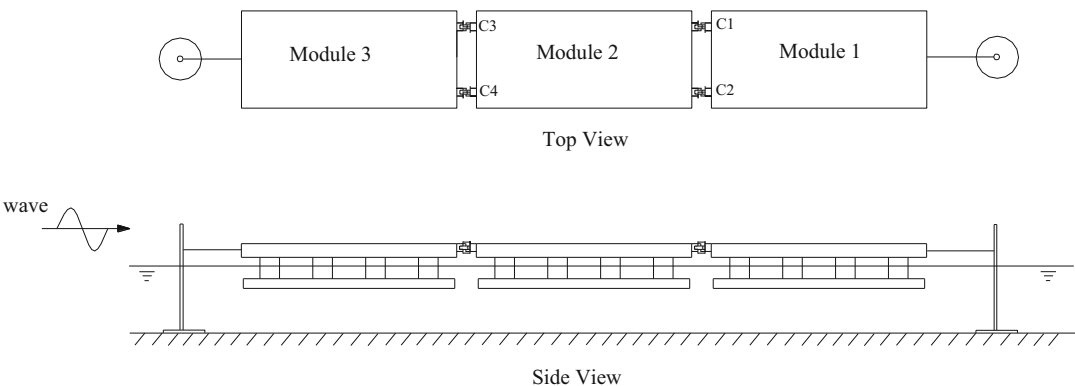
Each module is 2.75 m long, 1.25 m wide. Its depth is 0.575 m, and the draft is 0.288 m. The dimension in full scale is 275 m long, 125 m wide, and its depth and draft is 57.5 m and 28.8 m, respectively. The model is made of fiber glass reinforced plastics. Its load conditions, the position of center of gravity, longitudinal radii of gyration, and rolling periods are adjusted at the trimming table by adding and shifting the weights so as to meet the specified requirements. The setup of the model and connectors are shown in Figs. 1, 2, 3, 4, 5, and 6.

Test Program Semi-submersible VLFS in regular waves and irregular waves are tested. In regular wave tests the influence of wave period and direction on the motions and load of VLFS has been studied. And the irregular wave tests mainly focuses on the characteristics of multi-module VLFS in different wave directions and different connector rigidities.

In the test, noncontact optical system for measuring six degree of freedom motions is located on each module to measure their motions. Meanwhile, eight sets of 3-D force transducers are installed at the ends of four connectors to measure the connector loads. Its range for measurement is 0 ~ 50 kg and the accuracy is about 2.5% of the range. Wave heights and periods are measured with wave probe of resistance type. Its range is ± 30 cm and nonlinearity is less than 0.3%. Before the experiment, all these sensors are calibrated carefully.

Different water depth and wave heights are simulated according to the real operation environment and the theoretical calculation. And the water depth is 3 m corresponding to the full scale of 300 m.

Regular and irregular waves are adopted in the experiment. Regular wave height is less than 6 cm and the periods are 0.63 s, 0.70 s, 0.79 s, 0.90 s, 1.05 s, 1.26 s, 1.57 s, and 2.09 s. The wave height in full scale is not over 6 m



Experimental Techniques for VLFS, Fig. 6 Setup of the three-module semi-submersible VLFS model test

Experimental Techniques for VLFS, Table 2 Photos of model experiment of VLFS

Photo 1 The box-typed VLFS model



Photo 2 The VLFS model in waves

Photo 3 The VLFS model in 3-dimensional non uniform sea environment



(continued)

Experimental Techniques for VLFS, Table 2 (continued)

Photo 4 The VLFS model in non-collinear waves

E

Photo 5 Model test of a 3-module Mobile Offshore Base

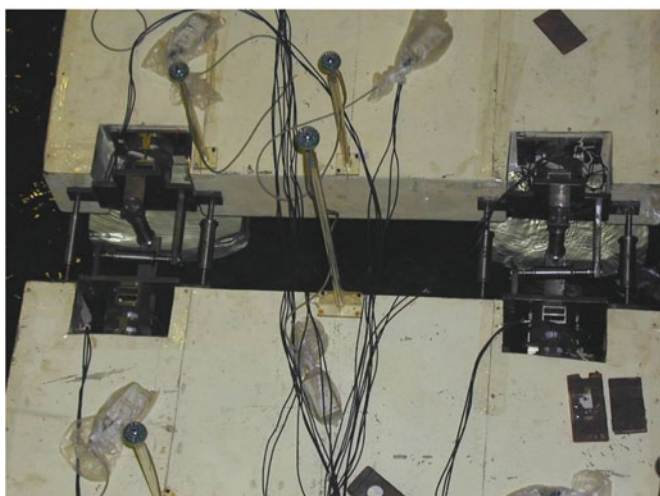


Photo 6 The connector model between modules

and the periods are 6.3 s, 7.0 s, 7.9 s, 9.0 s, 10.5 s, 12.6 s, 15.7 s, and 20.9 s correspondingly. The wave direction is also changed in the experiment to study the effect of wave incidence angle on the module motions and the connector loads. According to the test results of different wave height, RAO curves of VLFS motions and connector loads are obtained and compared.

Irregular waves are simulated according to the real environmental condition in operation and theoretical calculation. The Breschneider two-parameter spectrum is adopted as follows:

$$S_{\xi}(\omega) = \frac{1.25}{4} \left(\frac{2\pi}{T_p} \right)^4 \omega^{-5} H_s^2 e^{-1.25 \left(\frac{2\pi}{T_p \omega} \right)^4}$$

where T_p means spectrum peak period, H_s means the significant wave height.

In the experiment, the significant wave height of irregular waves is 1.88 cm and 3.25 cm, and the spectrum peak period is 0.88 s and 0.97 s, respectively. The significant wave height in full scale is 1.88 m and 3.25 m, respectively, and the peak period is 8.8 s and 9.7 s. Some photos are shown in Table 2.

Cross-References

- [Connectors of VLFS](#)
- [Floating Bridge](#)
- [Hydroelasticity Theory](#)
- [Mega-Float](#)
- [Station-Keeping System for VLFS](#)
- [Structural Analysis and Design of VLFS](#)
- [Very Large Floating Structures \(VLFS\): Overview](#)

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Exploitation of Gas Hydrates

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Definition

Gas hydrates are ice-like crystalline solids which consist of small molecules (typically gases) that are trapped inside “water cages” formed via hydrogen bonding (Sloan and Koh 2008).

Introduction

“Gas hydrates” is a common term for natural gas hydrates which are believed to be one of the most promising sources of fuel in the twenty-first century due to the enormous and untapped resources (Sloan and Koh 2008, Boswell and Collett 2011). A frequently quoted estimate of the global in-place natural gas resource volume contained in gas hydrates is about $2.1 \times 10^{16} \text{ m}^3$ (Milkov 2004), which is about twice the total proven worldwide resources of coal, oil, and natural gas. Natural gas hydrates have been found in both regions of permafrost and subsea sediments; however, the occurrence and amount of subsea gas hydrates (also known as marine gas hydrates) are much more than those in permafrost environment as shown by the US geological survey (Ruppel 2011). Gas hydrates can be found in different geologic environments, and their morphologies are different (Letcher 2013): They can be hydrate-filled networks of pores, massive hydrate lenses, grain-filling gas hydrate in marine sands, sea-floor mounds, grain-filling gas hydrate in marine clays, and grain-filling gas hydrate in onshore arctic sands/conglomerates. Marine gas hydrates can be found on continental slopes, land rocks (deep-water platforms), and basins in marginal waters of oceans, which have a water depth more than 350–600 m in the muddy seabed or in the loose sediment layer below the seabed. The

actual depth of hydrates occurrence depends on the temperature of water at these locations. Marine gas hydrates are not nice clear crystals but rather are muddy mixtures with sand, which is partly the reason that it is challenging to produce natural gas from them.

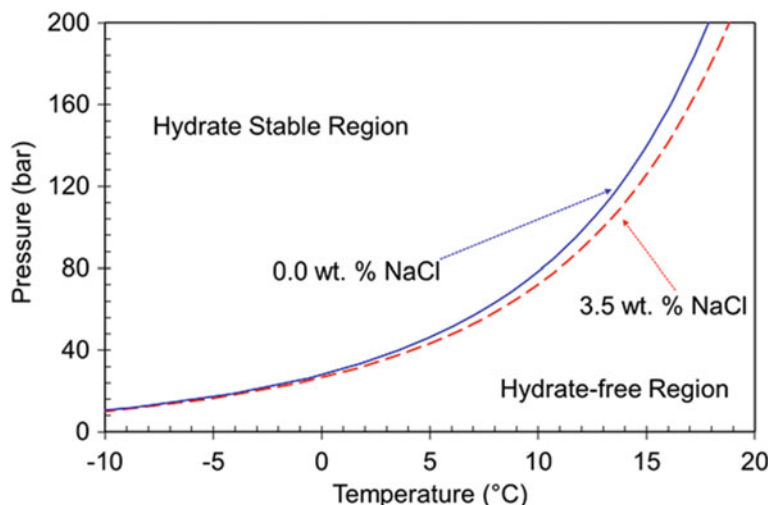
Structure and Thermodynamics of Hydrates

Natural gas hydrates are crystalline inclusion compounds, and they are nonstoichiometric. The methane molecules are caged by the hosting water molecules when the gas-water system is at high pressure and low temperature (Sloan and Koh 2008). The polyhedral cage-like hydrate lattices are formed via hydrogen bonding of water molecules and stabilized by the guest molecules. There are three common structure types (structures I, II, and H) for gas hydrates, which are largely determined by the type and size of guest molecules in the hydrate cages (Sloan and Koh 2008). For structure I, each unit cell is composed of 46 water molecules and 8 methane molecules. Other small gas molecules such as ethane and carbon dioxide can also form the type I structure. For structure II, each unit cell has 136 water molecules and 24 hydrocarbon molecules (Sloan 2011). The guest molecules in this structure can be propane and isobutane in addition to methane. For structure H, there is no consensus on the number of water molecules and gas molecules in each unit cell, and this type of structure could hold heavier hydrocarbons with more than five carbons in the molecules. Structures I and II are the most common gas hydrates in the oil and gas industry, while structure H is less common in either naturally occurring or artificial hydrates (Sloan et al. 2011).

Natural gas hydrates mostly exist in the form of structure I, as reported by Sloan et al. (2011), and they can be found at different combinations of temperature and pressure. Figure 1 shows the hydrate phase boundary curves of a pure water and methane system as well as a brine water (3.5 wt.% NaCl) and methane system, which was produced based on the predictions from

Exploitation of Gas Hydrates,

Fig. 1 Examples of structure I methane hydrate phase diagram for a pure water + methane system and a brine water (3.5 wt.% NaCl) + methane system



InfoChem's Multiflash 7.0 (KBC 2018). The "Hydrate Stable Region" is the temperature and pressure region wherein hydrates are in a thermodynamically stable state, while the "Hydrate-free Region" is the temperature and pressure region wherein gas and water coexist in different phases. The hydrate phase boundary is affected not only by the gas pressure and temperature, but also by the chemical composition of the water solution. The presence of salt decreases the required pressure for hydrate formation. It has to be cautioned that Fig. 1 is only an empirical phase equilibrium, and it should not be used directly for the prediction of hydrate dissociation because the hydrate formation process also involves kinetic effects (Lim et al. 2020). However, Fig. 1 could provide some guidance for the production of natural gas from methane hydrates, namely via either lowering the pressure of the gas in the hydrates system or increasing the temperature of the system to dissociate the hydrate to release the trapped gas. And for practical production planning, reliable gas and water solution compositions have to be obtained to decide proper exploitation routes.

Exploration and Production Trials

As they consist mostly of water (85 mol% water +15 mol% guest molecules), the physical appearance and density of gas hydrates are close to those

of ice (Koh et al. 2011). However, other physical properties, such as the shear coefficient, dielectric constant, and thermal conductivity, are lower than those of ice. In addition, the acoustic propagation velocity of natural gas hydrates is significantly higher than that of conventional gas sediments and saturated water sediments (Gabitto and Tsouris 2010; Wang et al. 2019). These differences are the theoretical basis for the exploration of natural gas hydrates by geophysical methods (Thakur and Rajput 2011). Geochemical methods could also be used for the identification of possible locations of hydrate formation (Yang and Sun 2011). Geochemical methods rely on the detection and analysis of the changes in the geochemistry of water bodies and sediments, which otherwise would be different from their normal properties if there were no hydrates because of the decomposition and crystallization of hydrates. Emerging marine technologies such as underwater imaging have also been applied for the location of hydrates deposits (Liu et al. 2020). Suitable methods should be selected for different environments and conditions, and sometimes a combination of methods should be considered to ensure the accuracy of exploration. The US geological survey shows that marine gas hydrates are most abundant in the Pacific marginal waters, followed by the Atlantic Ocean, on the edge of continental shelves (Ruppel 2011), which are favorable for gas hydrate formation because the sediments are

more developed and richer in organic matter and there are sufficient sources of methane-based hydrocarbon gases.

The production of methane from natural gas hydrates requires the release of methane from the hydrates formation in the seafloor before extracting the gas. Methods for dissociation of hydrates are largely based on the principles shown in Fig. 1, mainly including depressurization, thermal stimulation, and chemical stimulation (Moridis et al. 2009; Lee et al. 2011). The depressurization method is relatively simple to implement. It can be done by drilling a well for depressurization to favor the shift of the hydrate-gas phase equilibrium to the free gas side. However, the production rate of the depressurization method is slow because of the limited mass transfer and heat supply in the subsea environment. In addition, the depressurization method is not suitable for gas hydrate reservoirs where the reservoir temperature is near or below 0 °C, because the water may form ice at temperatures below 0 °C and the ice can retard the dissociation of hydrates and delay the transport of gas released from the hydrates (Yousif et al. 1991; Li and Han 2019). Thermal stimulation is another method to help release gas from the hydrate formation. This method requires the supply of heat to decompose the hydrates; therefore, it is also called thermal excitation method or thermal decomposition method. The heating methods mainly include the injection of steam, hot water, and hot brine as well as electromagnetic heating. This method can release gas faster than the depressurization method, but the limitation is the high heat loss and low thermal efficiency (Lee and Holder 2001; Uchida et al. 2004). Chemical stimulation method involves the injection of chemicals that will break the molecular cage formed by the water molecules, and they are essentially the same as those used for flow assurance purposes (Koh et al. 2002). However, extra care must be taken to ensure the injection of chemicals would not contaminate the subsea ecosystem. In recent years, an additional approach of releasing CH₄ from natural gas hydrates is proposed, and it is commonly referred to as carbon dioxide hydrate replacement for methane hydrate (Zhao et al. 2012; Komatsu et al. 2013). This method recovers CH₄ through the

dissociation of methane hydrates caused by the heat released from the formation of carbon dioxide hydrates, which is because carbon dioxide preferentially (compared to methane) forms hydrate since the temperature and pressure conditions required for CO₂ to form hydrate are much lower than those for methane. This method is being actively investigated at the laboratory-scale because of its potential of sequestering CO₂ and producing CH₄ at the same time.

To date, well-known trial productions from natural gas hydrates have only been carried out in two permafrost areas and two marine areas. Chronologically, the first production trials were from permafrost gas hydrates in the Mackenzie Delta region in Canada, which were carried out three times in 2002, 2007, and 2008; the second was the 2012 permafrost trial in the Alaska North Slope region in the United States; trial productions from marine natural gas hydrates were carried out in the Nankai Trough in Japan in 2013 and 2017; then more trials from marine hydrates were tested by China in the Shenhu area in the South China Sea in 2017 and between October 2019 and April 2020.

Both thermal stimulation and depressurization methods were used in the Mackenzie Delta trials (Dallimore et al. 2012). In 2002, the thermal stimulation method was used and 463 m³ gas (at standard conditions) was produced in 5 days. In the 2007 and 2008 trials, the depressurization method was used. In 2007, 830 m³ gas was produced in 12.5 h; and in 2008, 13,000 m³ gas was produced in 6 days. The trials were eventually aborted due to the blockage of the pipeline by the coproduced sand. The Alaska North Slope trial used the CO₂ and methane hydrate replacement method, which was the first notable field trial (Boswell et al. 2017). A mixture of CO₂ and nitrogen was injected into a reservoir, and CH₄ was produced along with water and sand. Some exchange of CO₂ with CH₄ in the hydrate phase had occurred; however, the low exchange rate of CO₂ and CH₄ has limited the overall CH₄ production rate and recovery. Therefore, depressurization was later used as an auxiliary method and the trial produced 24,000 m³ gas in a month. The 2013 Nankai Trough trial extracted 120,000 m³

gas in 6 days using the depressurization method, and the trial was forced to stop due to the sand production problem. Another two trials in the same area were carried out in 2017. In May, 35,000 m³ gas was produced in 12 days, and the operation was shut down due to the same sand production problem. In June, 200,000 m³ gas was produced in 24 days. Although there was no sand problem, the production rate of methane was low (Oyama and Masutani 2017). In May 2017, China conducted its first test of gas production from marine gas hydrates using the depressurization method in the Shenhu area in the South China Sea. In 60 days of continuous gas production, 309,000 m³ gas was produced, which includes both methane from the decomposition of natural gas hydrates and the free moving gas below the hydrate layer. China conducted a second test in the same area between October 2019 and April 2020, and 86.14×10^4 m³ gas was produced in a 30-day continuous operation (Li et al. 2018; Ye et al. 2020).

Technical and Environmental Challenges

The highest daily average rate of all the production trials is 28,700 m³/day, achieved in the second test in the Shenhu area. However, this production rate is still far less than the commercially viable rate, which needs to be more than 100,000 m³ gas per day for a single gas well. The low gas production rate from gas hydrates mainly arises from the slow decomposition of the hydrates and the coproduction of sand and the blockage of gas and water pipelines (Letcher 2013). Therefore, it is critical to understand the geological parameters of hydrate reservoir in addition to the thermodynamics of hydrates so that appropriate production technologies could be selected (Moridis et al. 2011; Lau et al. 2018). Meanwhile, the gas extraction technology and process also need to be improved to increase production efficiency, especially the way of preventing or reducing the intrusion of sand into the production line. Although potentially improved methods could be tested in laboratory, they must be validated in real production trials, which is a

potentially major obstacle for deploying because these field tests may require significant resources and collaboration. Another technical challenge to produce gas from marine hydrates is the unconsolidated nature of the hydrate reservoirs because of the dissociation of gas from hydrates, which may cause the reservoir to lose its mechanical strength such that the production well may collapse and cause the loss of production equipment. To understand and overcome these challenges, both experimental and numerical simulations of the hydrate reservoir conditions need to be performed. It was reported that experimental and numerical simulations have showed that new technologies such as horizontal wells and multi-branch wells (Maricic et al. 2008; Reagan et al. 2008), which have been widely used for coalbed methane extraction, can increase the production capacity by five to six times. However, as most subsea hydrates are shallow and soft sediments, further geological data are required to support the construct of such wells in practical operations.

The environmental impact of producing methane from natural gas hydrates has attracted the attention from governments and scientists, with concerns such as sudden release of large amount of methane from the seafloor, damage to the marine ecosystem, impact on the global warming, and subsea geological disaster (Li and Han 2019). It is possible that a large amount of gas is suddenly released from the subsea hydrates when the stability of the hydrate area is disrupted by the exploitation of gas. Such a gas release will cause the seawater to fluctuate drastically, producing an effect like an earthquake in the seafloor. To avoid this, proper real-time monitoring of the production site and the adjacent area is essential. However, it must be noted that there is no evidence that gas hydrate production could lead to uncontrolled hydrate dissociation and gas release to the sea water (Ruppel and Kessler 2017).

The marine ecological environment may also be adversely affected by the leaked methane into the water body (Yamamoto et al. 2014). Most of the methane that enters the seawater will react with oxygen to produce carbon dioxide when there is sufficient oxygen in the water body. The consumption of oxygen in the seawater will result

in the depletion of oxygen, which is a direct cause of the extinction of certain marine life. Therefore, there is a concern that the exploitation of natural gas hydrates may affect the ecological balance of the ocean and lead to the extinction of some marine organisms.

Another concern about the potential environmental impact of developing natural gas hydrates is the impact on climate. Although methane consists a small percentage of the atmosphere, it has a much greater greenhouse effect than carbon dioxide; it is 28–36 times greater than that of carbon dioxide over a 100-year period (Balcombe et al. 2018). The extraction of natural gas hydrates will inevitably result in the release of methane into the atmosphere, just as much as the conventional gas industry if not more. The warming effect of methane decreases with time due to chemical reactions in the atmosphere (Holmes 2018); however, proper measures should be taken to reduce the emissions since any more methane in the atmosphere is going to have an impact on the oceans and could lead to a global warming trend exacerbated by the dispersion of gas hydrates.

Natural gas hydrates affect the physical properties of the sediment (Gabbito and Tsouris 2010); thus, hydrates will have an impact on the stability of the seafloor. The hydrates are stable under certain pressure and temperature conditions, but their dissociation for gas production may cause geological disaster (Maslin et al. 2010). Gas hydrates are also related to the safety of offshore oil and gas facilities. A small change in pressure or temperature caused by oil and gas production can cause fracture of the gas hydrate layer, resulting in well blowouts, subsea collapse, and coastal landslides, which are not rare scenarios in the gas production history. Therefore, geological monitoring of the natural gas hydrate production site is important to avoid potential problems (Sun et al. 2020).

Concluding Remarks

Natural gas hydrate is a crystalline substance formed between various hydrocarbon and non-hydrocarbon substances and water molecules

under low temperature and high-pressure conditions. Methane is the most common and the most abundant gas in natural gas hydrates, which could be an alternative energy source in the twenty-first century, and their successful development may provide energy security to countries with large hydrate resources. Marine gas hydrates are overwhelmingly dominant, and they are found in water depth below 350–600 m in the muddy seabed or in the loose sediment layer below the seabed. As the hydrates are commonly mixed with muddy sand, it is challenging to efficiently produce natural gas from them. Most natural gas hydrates have type I structure, which has relatively well-understood phase equilibrium properties that can be used as a guide to produce gas. Natural gas hydrates have been found in many locations around the world, but there is no commercial production operation yet although several trial productions have been carried out. A few technical and environmental challenges need to be tackled before commercial production of methane from hydrates. As a new energy source with great potential, natural gas hydrates undoubtedly have a very broad development prospect. If the production and environmental challenges are solved, natural gas hydrates will become a new treasure in the energy field in the future.

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Explosive Dismantling

► Explosive Removal

Explosive Removal

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Synonyms

Explosive dismantling

Definition

To remove a fixed platform, the installation steps are essentially reversed. Topside equipment such as living quarters, generators, and processing equipment is taken off by crane and returned to shore for scrap or to be reused. The deck sections are then lifted from the platform and placed on cargo barges for transportation to their disposal site. Conductors, flare piles, well conductors, submerged wells, and caissons are structures subject to explosive removal (MMS 2004).

Scientific Fundamentals

The History of Explosive Removal

Nowadays, a large number of offshore structures are in operation to exploit oil and gas all over the

world. However, the operator may make a decision to terminate and remove the offshore productions from a producing field if they are not uneconomic any longer. Therefore, most of the platforms carried out to be removed are located in shallow water depths (less than 200 meters). Considering the movement of offshore production platforms and recent interests in deepwater development are increasing, it led to a thinking about how to remove the platform in deeper OCS (Outer Continental Shelf) waters (e.g., over the continental slope) (MMS 2004).

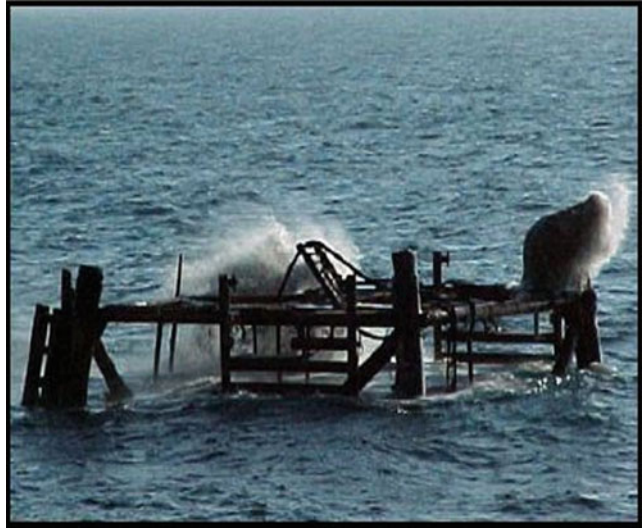
Explosive removal is considered as a conditional and common method to remove the useless offshore structures (shown in Fig. 1). The removal of the offshore structure (platform or ship) usually requires explosives to remove structural components in a few feet below the bottom of the sea, such as wellhead, pile, etc. In addition, considering the presence of endangered species or threatened species which are listed or proposed to be included in the America's Endangered Species Act in protected species in the ocean as well as the requirements of reducing or eliminating the influence on these species, the environmental impacts associated with explosive platform removal needs to be fully understood and attached important to.

Procedure in removing a fixed platform is basically reversed compared to the installation steps. Topside equipment such as living quarters, generators, and processing equipment is removed with a crane and back to the shore for scrap or reused. Then the deck sections are lifted from the platform and placed on cargo sailing to the disposal site in the transportation. Conductors, flare piles, submerged wells, and caissons are also structures to be removed with the use of explosives. The regulations of Minerals Management Service (MMS) in the USA make a requirement that piles and well conductors shall be severed at least 5 m below the mud line (MMS 2004). The piles and conductors are lifted and board on a cargo barge for the use of ultimate disposal once they are separated from the seabed.

The current researches focus on collecting and synthesizing existing information for explosive removal of offshore structures in aquatic

Explosive Removal,

Fig. 1 Explosive removal of offshore structure (MMS 2004)



environments. A more detailed illustration of explosive removal will spread, including the techniques, characteristics of underwater explosive removal, and impacts on the environment in this paper.

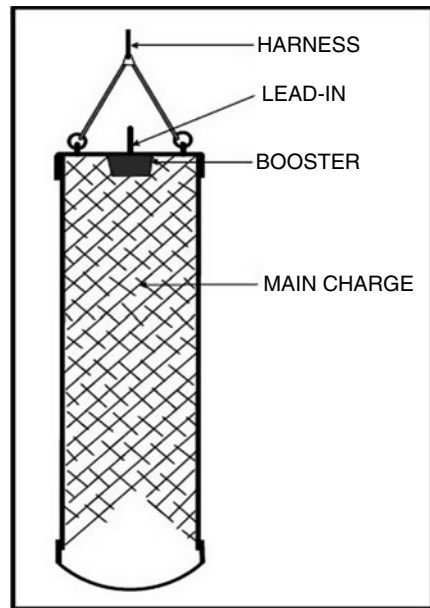
Explosive Removal Techniques

Placing an explosive charge inside the offshore structure at the desired depth and severing the explosively is the safest and easiest way of cutting procedure (MMS 2004).

Severing of piles and conductors in the underwater environment is a slow and dangerous process for the reason that it requires mass of diving time and divers will bear enormous risks. The explosive removal technique is the key point of the development of explosive removal. At present, it can be summarized into bulk explosive, configured bulk explosive, and cutting charges.

Bulk Explosive

Bulk explosives such as C-4 or Comp B (Fig. 2) are the most common techniques for explosive cutting of piles and conductors. These explosives are plastic and fabricable which have a high velocity on detonation and a high shattering power. Compared to other types of high explosives, they are less dangerous to handle. In addition, bulk explosives can be molded in the field and made into specific sizes and shapes. Compared with



Explosive Removal, Fig. 2 Bulk explosive (DEMEX 2002)

conventional cutting techniques, the important advantage of explosive technology is plasticity. During the construction of the platform, piles are welded together with a steel guide (called “stabbing guides”) on the inner bottom of each pile to facilitate the matching of the next part. In this way, the inside diameter of the pile is reduced inside

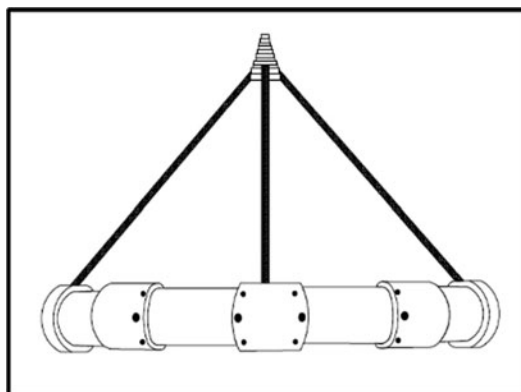
and outside the weld. The reduction in inner diameter of the pile is of great importance when conventional cutting techniques are used where the equipment shall be lowered along the pile and retrieved after cutting is finished. Variations in pile diameter are relatively inconvenient when bulk explosives are used. Bulk charges can be resized to sever a pile on the scene and do not have to be retrieved. Bulk charges can also be reshaped on the scene when the difference between the “planned” and the “as built” diameters of piles or well strings is encountered. The operation proceeds with relatively little delay owing to the flexibility; thus the flexibility is very important. Another advantage of bulk explosive is its high success rate, which is about 95% if sized properly. Furthermore, the increase in water depth has no adverse impact on the success rate of bulk explosive cuttings.

Configured Bulk Explosive

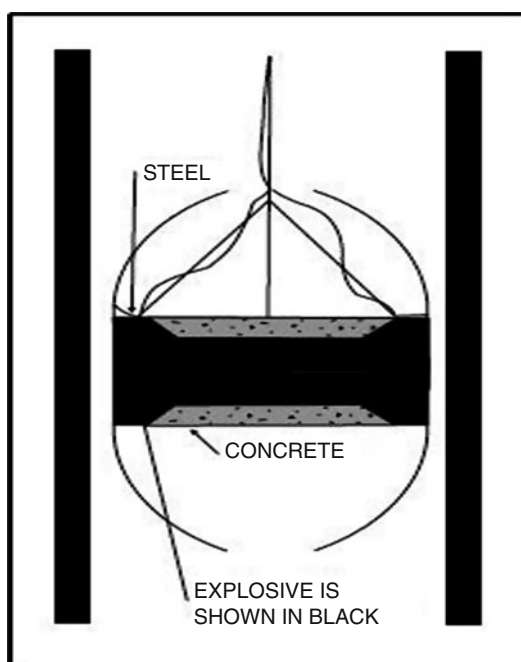
Configured bulk explosive includes “ring explosive” and “focusing explosive.” It is designed to collide the detonation front in order to concentrate more energy along the fracture line; the size of the charge carried to sever a piling is reduced consequently. The charges of these types proved to be effective in reducing belling or bulging out of the severed piling.

Ring Explosive The materials of ring explosive (shown in Fig. 3) are the same with that used in bulk explosive (shown in Fig. 3), which are mainly C-4 or Comp B and lowered into the piling. These rings concentrate the explosive closer to the inside of the piling’s walls, making the explosion process more effective. The total weight of charges required to sever a specific piling can be reduced by 10–15% approximately with ring charges (NRC 1996).

Focusing Explosive Focusing charges (shown in Fig. 4) are the sandwiched structures consisting of explosives between two tamping plates, one above and the other below. The use of smaller charges is accepted because these tamping plates have the function of delivering more power of explosive to the piling walls horizontally.



Explosive Removal, Fig. 3 Ring explosive (DEMEX 2002)



Explosive Removal, Fig. 4 Focusing explosive (DEMEX 2002)

Reductions in explosive weight with focusing plates are comparable to reductions in weights associated with ring charges. The weakness with both ring and focusing charge arrangements is that they must be prefabricated and sized to fit each application. Generally, each configuration varied in size which will allow for minor variations of inside pile diameters or obstructions. The

drawback with both ring and focusing charge arrangements is that the charges must be pre-fabricated. It is difficult to employ the configured charges to cut off wells due to the small diameter of the inner casing (NRC 1996).

Cutting Charges

Cutting charges include linear-shaped charges and cutting tape, each of which will be discussed with more details below. Compared to explosive cutting, mechanical cutting is less reliable, and the risk of failure is much higher (Kaiser et al. 2002). In addition, potential future explosive cutting techniques cover contact plaster, shock wave focusing, and radial hollow.

Linear-Shaped Charges Linear charges use high-speed explosives as energy to accelerate the v-shaped strip of cutting materials, through the pile-driving steel with high-speed jet flows. The linear-shaped charges can be used in a specially manufactured container which is assembled around the outside of the piling to sever the pile from the outside (shown in Fig. 5a). With the help of a positioning tool, linear-shaped charges can also be placed inside platform legs (Fig. 5b).

The performance of linear-shaped charges is relevant to the presence of an air space between the charge and the target. This standoff distance required is a function of the thickness of the material to be severed. When the standoff distance between charge and target is accurately positioned

and precisely calculated, the linear shape of the charge produces a very smooth cut (NRC 1996).

Lead time of linear-shaped charges is quite long, and accurate engineering data of piling or caisson to be cut is needed. If the thickness of a pile section is unknown or the pile is not round, the cut produced by a linearly shaped charge will not be clean. If linear-shaped charges are placed outside a piling, the explosive power provided by soil will not attenuate. Linear-shaped charges must be hinged to pass through the stabbing guides to be used inside a piling. A clean cut cannot be obtained if a stabbing guide is located at the height of the proposed cut. Furthermore, it will lead to a great performance degradation of charging if water or soil permeates the air space at a distance.

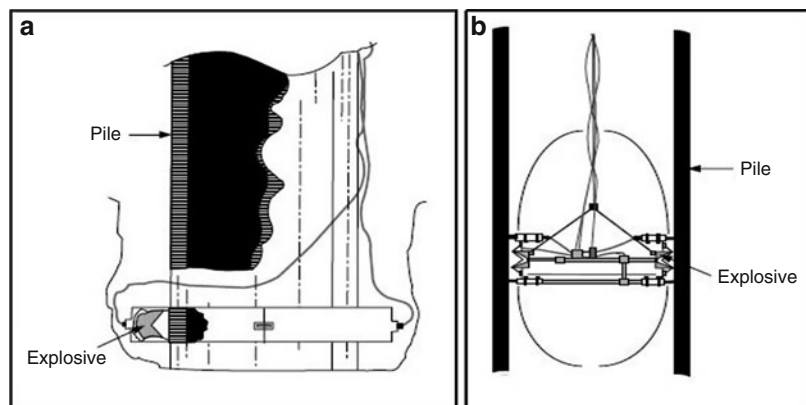
Cutting Tape Explosive cut tape is a flexible form of linear charge (Fig. 6). The explosive and the liner are squeezed into a shape charge that is fixed in a flexible conductor, allowing the tape to form a profile with the pile, thus keeping a proper distance around the entire pile.

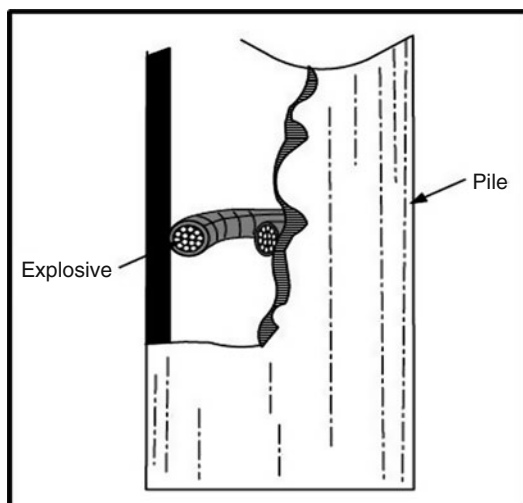
There are two disadvantages of cutting tape (MMS 2004):

1. The diver is required to place the tape around the target pile.
2. As the charge and the liner are located on the surface of the conductor and the depth is greater than 91 m, the environmental pressure may cause the conductor to deform, thus

Explosive Removal,

Fig. 5 (a) External linear-shaped charge (DEMEX 2002) (b) Internal linear-shaped charge (DEMEX 2002)





Explosive Removal, Fig. 6 Flexible linear-shaped charge (DEMEX 2002)

changing the spacing and reducing the charge cutting efficiency.

At this point in time, cutting tape is less efficient than linear charge removal platforms, especially in the sea areas where the depths are more than 91 meters (NRC 1996).

Key Applications

Explosive Process

Most explosives used in underwater explosions contain oxygen and have relatively unstable chemical bonds (MMS 2004). When the explosion began, oxygen was easily released and recombined with other atoms to form more stable molecules. The explosive reaction is the breaking down of the original molecules into derivatives, which is accompanied by the release of large amounts of heat.

Physical impact caused by detonating explosives, which trigger the release of chemical energy, so as to realize self-sustaining wave, is known as the detonation wave. It is different from the shock wave propagating by non-reacting media, in which conditions the chemical transformation of energy release affect the shock front. The wave is not driven by the motion of the

boundary surface behind the shock wave, but the internal conditions of the matter behind it. At a relatively close distance from the explosion, the charge diameter of the explosive is generally considered to be three to ten times, and the direct explosion effect generated by the explosion can be neglected. The main source of shock around this distance is the gas reaction product of shock wave and expansion.

Shock Wave and Propagation

Pressure Waveform

The main shock pulses have rapid rising time and exponential decay, which are followed by a series of bubble pulses (MMS 2004). This high-pressure pulse is formed by the rapid conversion of solid explosives into gas form. The negative phase pressure after the impulse is due to overshoot. When the bubble reaches its maximum volume with the minimum of the negative phase, it begins to collapse. When the bubble reaches the minimum volume with the second peak in the pressure waveform, it would be called the first bubble pulse. This oscillating behavior continues, although the amplitude pulse decreases as the energy dissipates.

Sea Surface Reflections

The sound wave energy of sea surface reflection coefficient is close to negative 1 atmosphere, which indicates that the reflection phase is reversed 180° and the incident pressure pulse is reflected in reverse polarity. Surface roughness causes reflected scattering. In addition, bubbles near the surface can absorb sound waves or scatter reflected energy as a result of the wave's rupture. Therefore, for high-frequency sound, roughness is usually more important than low-frequency sound (MMS 2004).

Sea Bottom Reflection

For the seafloor, the impedance contrast between surface and seabed material is smaller than that between surface and air. As a result, a large number of acoustic energy events occurring on the seafloor will be transmitted to the seafloor, where it may be reflected back from beneath the

seafloor. A good summary can be found in Brekhovskikh (1980). From the research of Brekhovskikh, the bottom reflection coefficient under steep angle is true and positive, while the bottom reflection coefficient under shallow angle is much more complex.

Impacts to the Marine Environment

The environment problem is serious and non-negligible to the global world, while the impacts to the marine environment may be the biggest disadvantage of explosive removal, which have been paid close attention by people from all walks of life. Explosive removal of ocean structures affects marine life, including fish, turtles, and marine mammals. A large number of studies have been conducted to study the effects of explosive removal on the marine environment. Dzwilewski and Fenton (2003) proposed a method that can accurately simulate the shock wave effect of explosive under the mud line of piles, legs, pipes, or other structural units. A method for calculating explosive effect by using the function of pile diameter, wall thickness, and weight of explosive is presented.

Impacts to the Fishes

Young (1991) conducted experiments to apply impact pressure to different types of fish in order to establish a damage prediction model. These studies show that the cube root scale is effective for the near distance of electric charge, but the effect of sea surface sparse waves and seafloor reflection may play a greater role in the larger distance, which should be considered in the long-distance prediction.

Impacts to the Sea Turtles

The researchers predicted that the eardrums of turtles are sensitive and that small turtles are more vulnerable to shock waves than large turtles. The dominant sea turtle in the USA observed during the removal of the explosive structure is the loggerhead, which is listed as an endangered species. On account of sea turtles' characteristic of behaviors, it is difficult to observe and estimate the impact of explosive removal on sea turtles. Sea turtles in temperate regions generally spend less than 10% of their time on the surface of the water

(Byles 1989; Kemmerer et al. 1983; Nelson et al. 1987; Renaud and Carpenter 1994). The injured turtles, who were able to swim, returned to the sea, and the dying ones sank to the bottom. In addition, explosives can detonate within days, weeks, or even months as the platform is dismantled, and bodies can be carried away by water or predators from the area. Without a thorough investigation of the seabed after each explosion, only a conditional assessment of the impact can be made (Gregg et al. 1997).

Edward (1988) counted and analyzed the impact of offshore oil platform explosion on sea turtles and dolphins, during the period of high and low sea level explosion from March to April 1985 to March 1988. The results showed that there was a positive correlation between the frequency of stranding and explosion of sea turtles. Dates and locations for the use of underwater explosives in offshore oil and gas structures are available throughout industry documents. No government agency or agency has a complete database for explosions in offshore waters and coastal areas, and even if autopsies are carried out, the cause of death cannot be determined.

Impacts on Sea Mammals

O'Keeffe and Young (1982) hypothesized that shock waves damage the lungs and other organs for the sea mammals. Research at Lovelace Institute for biomedical and environmental research on the effects of underwater explosions on dogs, sheep, and monkeys has shown that as a function of sample size, species respond similarly to shock waves. Two types of damage caused by underwater explosions are hemorrhage in the lungs and around the lungs and stimulation of radial vibrations in small bubbles that normally exist in the intestines (Yelverton et al. 1973, 1975).

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Extended Kalman Filter (EKF)

- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)

External Tieback Connector (ETBC)

- [Steel Pipelines and Risers](#)

External Turret

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Definition

The *external turret* is a system where the “CALM buoy” is fixed to a cantilever yoke at the bow or often on an increased height above the bow (gooseneck structure) of the moored tanker. It is widely used to maintain the floating production, storage, and offloading (FPSO) on station especially in shallow water.

Scientific Fundamentals

Introduction

FPSO, which is first applied in 1977, is generally used to separate and process the oil and gas from the extracted oil, storage, and transport of crude oil products (Liu and He 2006). In order to maintain the operational capability, the turret mooring system is often adopted for FPSO station keeping. Depending on the type of vessel and its operations location, the turret system can be divided into internal and external turret. It includes a turret system that is located at the extreme end of an outrigger structure attached to the bow of the

External Turret,

Fig. 1 External turret mooring system (www.sofec.com)



vessel, as illustrated in Fig. 1 (Bluewater 2009; Wichers 2013).

An advantage of the external turret system over other options for floating installations such as spread mooring is that it allows the vessel to rotate and adopt the optimum orientation in response to weather and current conditions. This rotation of the vessel about the turret is known as weathervaning. It is less expensive than internal turret designs and can be delivered in a shorter period of time, making it an excellent choice for many applications (Shimamura et al. 2006).

Components and Function of the External Turret System

Main Components

As shown in Fig. 2, the external turret system usually includes following four parts (Wall et al. 2002)

- **Turret (T):** This provides the single-point mooring and allows weathervaning of the vessel. It includes the turret shaft, turret casing, main and lower bearings, and the mooring spider.
- **Fluid transfer system (FTS):** This is typically a multiple swivel assembly. It transfers the process fluids and other signals from the turret to the process plant on the vessel. It is positioned above the turret and linked via the turret transfer system.

- **Turret transfer system (TTS):** This refers to the intermediate manifolding linking the turret and the FTS. It is often positioned on a turntable on top of the turret. It rotates with the turret. It is otherwise referred to as the turntable or turntable manifolding.
- **Interfacing systems (IS):** This includes swivel access structure, mooring lines and flexibles below the turret, and other ancillary equipment.

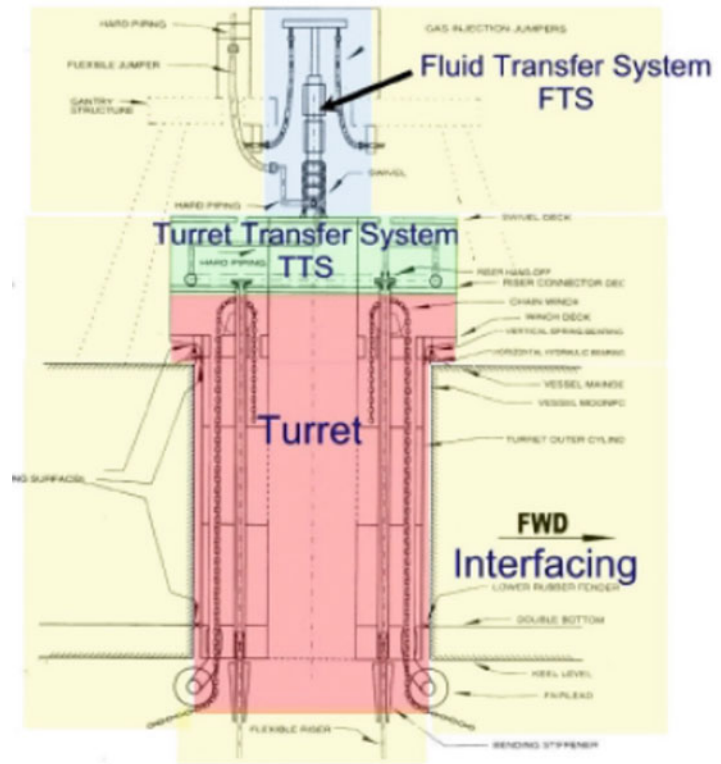
Function

The external turret system performs four main functions in a typical FPSO:

- Maintaining the vessel on station through single mooring
- Allowing weathervaning or rotation of the vessel to adjust to climate conditions
- Fluid transfer from the risers to the process plant
- Providing transfer of electrical, hydraulic, and other control signals

External Cantilevered Turret

It is possible to classify three main types of external FPSO or FSU turret, including external cantilevered turret (ECT), external disconnectable turret (EDT), and external wishbone (EW). Of the three main types of external turret designs, only the first two are currently widely used (Wall et al. 2002).

External Turret,**Fig. 2** Main components of external turret system (Wall et al. 2002)

The external turret is cantilevered from the bow of the vessel in a similar manner to a conventional external turret as shown in Fig. 3. The system will usually use a single or multipath swivel as a fluid transfer and have a simple turret bearing system.

Device Components

The ECT turret might typically comprise the following devices:

- Cantilever support
- Conventional multipath swivel
- Turntable and manifolding
- Bearing support and turret bearing
- Risers
- Chain table
- Winch for connection
- Mooring chains

The swivel is assembly mounted on a cantilevered support structure attached to the bow of the vessel. Mooring connections to

the turret provide single-point mooring to the seabed and allow the vessel to rotate freely around the turret adjusting to climatic conditions. The risers from the well are connected to the swivel. FPSO vessels of this type with conventional or multipath swivels can take oil production from several wells.

Design Principle

The turret design is generally smaller and handles fewer flow paths than internal turret. The main difference is that the turret is located at the end of a cantilevered support structure and not in a cavity in the vessel itself.

Pipework and turret are exposed, so corrosion of manifold pipework and inaccessible pipework in the cantilever area is of specific concern. Bearings will usually be of a roller design.

The principal features include:

- External cantilevered turret elevated above water level at the bow of the vessel
- Multipath swivel for FTS

External Turret,

Fig. 3 FPSO with external cantilevered turret (ECT) (www.sofec.com)



- Conventional turret bearing and support
- Multiple wells
- Articulation via universal joint to accommodate riser movement
- Mooring by conventional anchor chain and wire catenary mooring
- Isolation valves on each riser

Safety Concerns and Maintenance

The process plant and accommodation on the vessel is downwind of the turret and is a safety consideration. Consequences of hydrocarbon leakage could escalate if not controlled and appropriate evacuation procedures put in place. The turret is exposed at the bow of the vessel and susceptible to green water or wave damage and corrosion from seawater.

Typical maintenance for an external cantilevered turret may comprise:

- External visual examination of the turret for corrosion damage
- Functionality check on turret bearing and including torque checks
- Grease monitoring of turret bearings
- Examination of the swivel and dynamic seal performance
- Replacement of static seals
- Examination of flexible risers as recommended by the manufacturer particularly connections
- Leak monitoring of the riser connections, the interfaces, the manifolding, and the swivel
- Diver examination, or recovery and examination, of the mooring chains and connections
- Visual inspection of pipework in cantilever structure for corrosion damage

Disconnectable External Turret

Figure 4 is an external disconnectable turret. This is similar to the external cantilevered turret but has a tensioned riser connector that enables the riser and moorings to be quickly disconnected and allows the vessel to sail away from the station (Shimamura et al. 2006).

Device Components

The components are the same as for the external cantilevered turret (ECT) with the following additions:

- Connection/disconnection interface
- Tensioner
- Locking mechanism
- Universal joint to control riser orientation
- Riser column
- Winch for reconnection

This disconnection interface between the riser column and FPSO is a structural connector usually based on a hydraulic collet-type connector



External Turret, Fig. 4 FPSO with external disconnectable turret (ETD) inset (Wall et al. 2002)

typically 50–60 in. in diameter. The articulations of the riser column are accommodated by a universal joint mounted on the rigid arm turntable structure at the bow of the FPSO. The mooring loads are transmitted from the structural connector to the main bearing via the universal joint to the bearing support structure.

The universal joint is adopted to control the orientation of the risers relative to the turret. It is an important load bearing component, and any damage would affect the weathervaning of the FPSO.

The riser column supports the chain table, and the anchor legs once disconnected form the FPSO flexible, or conventional risers may be used.

The typical configuration of the external turret is that a gear ring located on the rigid arm turntable controls the orientation of the main connector and the fluid transfer connectors. Risers between the seabed PLEMs and the riser RTM are flexible hoses in a steep wave configuration. Risers are routed through J-tubes or to hard-pipe connections at two levels on the riser column. The disconnection interface is at the top of the riser column at the main structural connector level. All flow lines are routed through flow line

connectors with integral isolation ball valves through jumper hoses around the universal joint.

The riser column is reconnected by means of a linear winch. The reconnection wire rope is winched through the connector, universal joint, and swivel stack over the top pulley mount onto the support structure above the rigid arm. Once the connector flanges are face-to-face, the turntable structure is rotated to align the flow line connectors, and the structural connector is locked. Each flow line connector can then be connected individually.

Design Principle

The principle of operation is the same as for the external cantilevered turret (ECT) with the additional feature that the turret system can be disconnected from the FPSO in severe storms/cyclones, or for scheduled dry dockings. This allows the vessel to sail away under its own power. Disconnection of the turret system can be carried out from the FPSO bridge or locally at the bow. Reconnection is performed without the need for external assistance.

Such a turret design may be used on a site subject to seasonal cyclones. While connected the turret may be able to endure a sea state equal to a 10-year non-cyclonic storm (e.g., significant

wave height 4.5 m, current 1 m/sec). While disconnected, the mooring buoy is able to withstand a 100-year cyclonic condition with one damaged chain (i.e., significant wave height 10 m, current 2.5 m/sec).

Safety Concerns and Maintenance

The operational and safety concerns are the same as for the external cantilevered turret (ECT) with additional issues associated with connection and disconnection. The ability to connect and disconnect is a key part of safety cases. There is the potential for hydrocarbon release during connection and disconnection. The production and the swivel would be shut down and isolated before disconnection. Disconnection is a safety measure in the event of hydrocarbon leakage in the turret or manifolding systems.

Maintenance and inspection practices are the same as for the external cantilevered turret (ECT) with the following additions associated with the disconnection interface and universal joint:

- Check functionality of connection/disconnection interface
- Check functionality of turret bearing and universal joint including torque checks
- Check integrity of all riser connections and interfaces
- Diver examination or recovery and examination of mooring chains and connections
- Check function and maintain winch for connection/disconnection

Conclusions

The external turret is a vital component of the single-point mooring systems for the ship-based floating production systems in moderate to harsh environments, and it allows the floating systems to weathervane and align itself in the direction of minimum loadings. Furthermore, the external turret can be designed to be disconnectable so that the floating system could be moved to a protective environment in the event of a hurricane or typhoon. Due to the complexity of the external turret, further research on structural optimum

design and safety assessment and management is still significant to ensure the safe, reliable, and high efficiency operation of the ship-based floating system.

Cross-References

- [External Turret Single Point Mooring System](#)
- [FPSO](#)
- [Internal Turret Single-Point Mooring \(SPM\) System](#)
- [Single Point Mooring](#)

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External Turret Single Point Mooring System

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Synonyms

[Mooring system](#); [Positioning system](#); [Single point mooring system](#)

Definition

The device is a type of single point mooring system and belongs to the turret type single point mooring device. This type of single point mooring is moored to the seabed by a turret outside the bow (tail) through a uniform or grouped anchor chain. The FPSO hull rotates 360° around the external turret under wind, wave, and flow environmental loads, with a weather mark effect, which allows the hull to adjust to the direction of the least force. Because the single point mooring buoy and turret are installed outside the hull, it is called the external turret single point mooring system. The external turret single point mooring system is divided into a riser type and a cantilever type based on the structure type. The crude oil enters the oil storage tank through the submarine pipeline into the rotary sealing joint in the external turret.

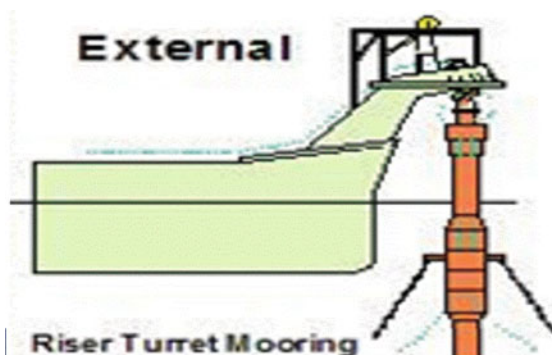
Development History

In the early 1980s, SBM first proposed the concept of a turret based on the principle of “wind-mark effect.” In 1985, an external turret was installed in the stern of a 140,000-ton floating storage tanker, applying this concept to reality (Yifeng 2012; Liu and He 2006).

The turret mooring works in the same way as other single point mooring methods. It also fixes the tanker to the mooring point at sea through a certain connection, so that it can rotate freely in all

directions with the action of wind, waves, and water current. The weather mark effect is formed to avoid the destructive force caused by wind and waves, thereby reducing the environmental load and motion response of the system under the external disturbance force of wind and waves. The turret mooring system is mainly composed of four parts: turret, liquid transmission system, turret rotation system or rotary table, and interface connection system. The turret is not only the mooring point of the tanker but also the passageway of the riser and other umbilical system to the hull deck through the seabed. The relatively fixed characteristics of the pontoon and the deck facilities ensure that the oil tankers can also achieve continuous oil and gas transport between the seabed wellhead under complicated sea conditions, which is the core of turret mooring technology. Compared with many other mooring methods, the turret mooring system has the following advantages: (1) simple structure; (2) less influenced by waves, suitable for severe sea conditions; (3) applicable to various water depths; and (4) it has a quick release and reconnect function for easy maintenance (Liu and He 2006; Liu 2012).

The traditional external turret mooring system is the external disconnectable riser turret mooring system. As shown in Fig. 1, it first appeared in 1986, mainly composed of overhanging turret, articulated head connector, cylindrical mooring riser, mooring chain, universal joint, rotary joint, and mechanical connecting device. It was applied to the development of the H59IJ3 oil



External Turret Single Point Mooring System, Fig. 1 Riser-type detachable external tower mooring system

field in the hurricane area of the northwest coast of Australia, thus opening the door to the development and utilization of turret mooring technology.

The advantages of the retractable riser-type external turret mooring system are less investment, easy to manufacture and demolition, release and return function, and mitigation of mooring characteristic of the mooring characteristics. The shortcoming is that after the system is released, because the riser displacement is much larger than the weight of a single point, there is a large residual buoyancy. The existence of the residual buoyancy makes the movement of the riser under the action of waves difficult to control, which is not conducive to the return installation of the riser type external turret mooring system.

After several decades of development, the external tower mooring system has been greatly improved. The external turret currently used is mostly an external cantilevered turret (ECT), as shown in Fig. 2, which is a permanent mooring system. The main feature is that the bow or stern is fitted with a large outrigger structure that can

accommodate the rotating bearing and the external turret. This supporting structure makes the turret out of the water (Lian-zheng et al. 2008; <http://image.baidu.com/search/index?tn=baiduiimage&ps=1&ct=201326592&lm=-1&cl=2&nc=1&ie=utf-8&word=FPSO>. (In Chinese)).

Operation Principle and Component

The cantilever external turret mooring system is mainly composed of a flexible riser, a liquid rotary joint, a mooring buoy, a mooring line, etc., wherein the liquid rotary joint is a core component.

Liquid rotary joints include:

1. Well flow rotary joint: The inlet end is connected to the underwater flexible hose, and the outlet end is connected with the inlet line of the FPSO production processing module to form a crude oil output passage.
2. Electric rotary joint: Power is supplied to the platform through electric rotary joints and



External Turret Single Point Mooring System, Fig. 2 Cantilever external turret single point mooring system

- power cables for use on equipment and equipment on the platform.
3. Hydraulic rotary joint: Hydraulic rotary joint provides hydraulic power control emergency shut-off valve.
 4. Emergency shut-off valve: The emergency shut-off valve is used to shut off the oil passage in an emergency state.

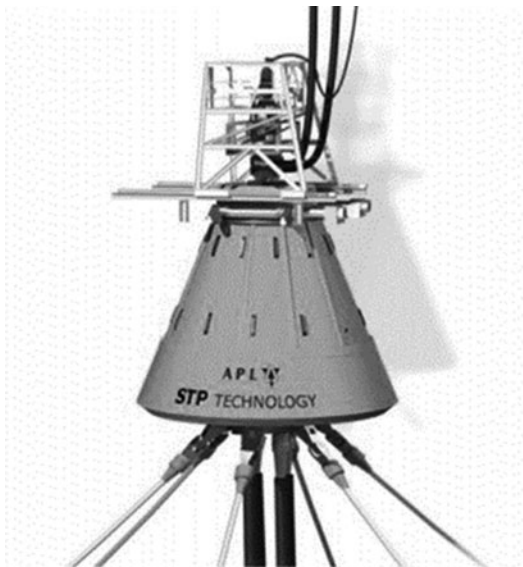
The mooring buoy is also a key component. Once the mooring buoy is detached, the entire mooring system fails. The pontoon has a conical structure, and a bearing assembly is arranged on the upper part of the pontoon, and the lower part of the pontoon is connected with the mooring line through the eye plate to transmit the mooring tension. The upper part is fixed to the overhanging structure by a special hydraulic locking device. Through the bearing assembly on the pontoon, under the action of environmental loads such as wind, waves, and current, the mooring center rotates or drifts. The structure of the pontoon is shown in Fig. 3. The locking device is an important component for ensuring that the pontoon does not fall off. The locking device clamps and locks the pontoon, so that the

overhanging structure is integrated with a single point. The locking device is the most critical device for a single point system. In actual production operations, the locking device must maintain sufficient preload.

Application

The advantage of the single-point mooring of the external turret is that the tanker weather mark effect is remarkable, it is easy to rotate to the orientation with the smallest environmental load (Geng-xian 2009). It does not occupy the hull storage space, and the oil tanker has high oil storage efficiency. The shortcoming is that the bow movement is the most responsive, especially in deep water mooring, the riser and turret of the external turret device are all within the range of wave slamming, which easily causes the damage of the riser and the turret. Therefore, the single point mooring device of the external turret is only suitable for medium deep waters with better marine environment.

At present, the world's largest single point mooring device for external turret is a cantilevered outreach tower designed and manufactured by FMC SOFEC. The mooring system was installed in the BIJUPIRA AND SALENA field in Brazil in August 2003. The unit can provide 18 riser and umbilical cable systems. Each mooring leg is composed of 154 mm diameter polyester cable. Under unloading conditions, the single point mooring system can withstand 360,000 tons of oil tanker and tail shuttle tanker.



External Turret Single Point Mooring System, Fig. 3 Buoy structure scheme

Cross-References

- [Internal Turret Single-Point Mooring \(SPM\) System](#)
- [Mooring System](#)
- [Positioning System](#)
- [Single Point Mooring System](#)
- [Soft YOKE Single Point Mooring System](#)

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F

Fabry-Perot Interferometer (FPI)

► [Fiber Optic Hydrophone](#)

Fatigue of Mooring Lines

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Definition

Fatigue of mooring lines is one of the factors that must be considered during the design of permanent mooring systems, and it is not required for mobile mooring systems. Different from the strength analysis, the maximum stress that causes fatigue damage has not yet reached the ultimate strength of material, sometimes even smaller than the yield strength. For all of the mooring line components, such as steel chain, wire, synthetic fibers, connecting hardware, etc., the fatigue analysis should be conducted in order to satisfy the fatigue design life.

Scientific Fundamentals

Historical Development

Under the long-term marine environment, the floating structures and their mooring systems are subjected to the effects of variable wind, current, and wave loads, so the alternating motion and stress/tension are caused in the mooring system. With the alternating stress/tension gradually accumulated, the cumulative fatigue damage of mooring line would occur. In recent years, many fatigue failures of mooring line in the practical projects have happened, such as during 2010–2013 where there were 15 offshore mooring line failures on the Norwegian continental shelf, according to Kvitrud (2014), and one third of them were fatigue failures, which indicates that the fatigue failure is still one of the critical failure modes of mooring system.

For the mooring chain, the first attempt to understand its fatigue behavior was conducted by a chain manufacturer Ramnäs (de Laval 1971), who used three-link samples of 2-in. oilrig chain to be tested at room temperature in dry conditions. Several companies also carried out research on the mooring chain fatigue tests, such as Shell Offshore Research (van Helvoirt 1982) who used 3-in. stud chains and connecting links to be tested under high-cycle fatigue range in artificial seawater aiming to investigate the fatigue

strength; Det Norske Veritas (DNV, Lereim 1985) who summarized the S-N curves of mooring chain links based on the 4-year research project about mooring lines; and Nippon-Kokan Kabushiki Kaisha (Asano et al. 1986) who tested samples of 40 and 44 mm of nominal diameter chain in air and water with and without cathodic protection.

For the mooring wire ropes, the first published paper related to its fatigue was conducted by Matanzo (1972), who investigated the relationship of the axial fatigue resistance of wire rope in air/seawater with the rope construction and core, as well as the effect of the applied mean load and load range, through the laboratory experiments. Lucht and Donecker (1977) also used the laboratory tests to investigate the axial tension fatigue under different field environments and deployment techniques. Hanzawa et al. (1981) presented the fatigue data for 50 mm diameter ropes with different constructions and socketing materials.

For the mooring fiber, Parsey (1982) first puts forward that the possible reason of its failure is fatigue through a SPM mooring hawsers case study. Hearle et al. (1998) divided the fiber fatigue into five effects: tensile fatigue (Bunsell and Hearle 1971; Oudet et al. 1984), hysteresis heating (Banfield and Hearle 1998), creep rupture (Hearle et al. 1993), internal abrasion (Seo et al. 1990), and axial compression (Flory 1996).

Some representative joint industry projects (JIPs) about mooring line fatigue were conducted, such as “Design Guidelines of Anchor Chains” led by Billington Osborne Moss Engineering Limited (Bolt et al. 1995); “Fiber Tethers 2000” led by Noble Denton Europe (1995); “Fatigue Testing of Large Diameter Mooring Anchor Chains” also led by Noble Denton and Associates Incorporated (Stiff et al. 1996); “Heat Build-up and Mechanical Property Behavior of Full Scale Polyester Rope” led by National Engineering Laboratory (1998); and “Mooring Integrity for Floating Offshore Installations” cooperated by 35 industry participants (Health and Safety Executive 2006, 2017).

Based on these series researches, the American Petroleum Institute (API) edited the first edition of mooring design rules including the mooring line

fatigue in 1993 (API RP 2FP1 1993) and then updated the new version in 1996, 2005, and 2008. Furthermore, API edited the mooring design rules especially for synthetic fiber ropes in 2001 (API RP 2SM 2001) and updated in 2014. Meanwhile, DNV also edited the first edition of mooring design rules in 2001 (DNV-OS-E301) and then updated the new version in 2004, 2008, 2010, and 2015.

Principles of Fatigue Analysis of Mooring Lines

In the fatigue analysis of floating structures, the most common recommended method is spectrum fatigue analysis, and typically only the wave-induced fatigue is considered. During the calculation process, the precise finite element model of floating structure should be established, and the hot spot stress position should be determined through the strength analysis of wave forces on the floating structure, and then the short-term wave state is assumed as Rayleigh distribution, so the stress of hot spot could be calculated through transfer function which will also follow Rayleigh distribution, finally the S-N curve is chosen, and the linear cumulative fatigue damage rule is used to calculate the total fatigue damage.

Different from the fatigue analysis of floating structures, the stress/tension of mooring line is calculated directly through the dynamic analysis of mooring line rather than transfer function, meanwhile all of the environmental forces induced in the mooring line tension could be considered, including the wave, current, and floater motion. As the same, the linear cumulative fatigue damage rule is also used. So, the key issues of fatigue life estimation are the fatigue resistance of mooring line components, the amplitude of cyclic stress/tension, and the loading cycle times. The specific fatigue damage analyses of mooring lines are shown as below.

Linear Accumulated Fatigue Damage Rule

The commonly used calculation method of fatigue life is the Palmgren-Miner (P-M) linear

accumulated damage hypothesis rule, which is firstly proposed by Palmgren (1924) and then popularized by Miner (1945). The P-M rule states that the fatigue damage of components or materials caused by the different stress/tension magnitudes is independent and unrelated, so the total damage is the linear superposition of fatigue damage caused by the different stress/tension magnitudes, shown as Eq. (1).

$$D = \sum_{i=1}^n D_i = \sum_{i=1}^n \frac{n_i}{N_i} \quad (1)$$

where D is the cumulative fatigue damage, and the material will be damaged when $D = 1$, n is the discrete number of stress/tension range, D_i is the fatigue damage under a different stress/tension range, n_i is the number of stress/tension cycle times in this stress/tension range, and N_i is the number of stress/tension cycle times that would lead to failure in this stress/tension range.

The design fatigue life is $1/D$, and it should be larger than the mooring system design service life multiplied a safety factor. For different mooring line components, the safety factor is different for the reason of uncertainties.

Meanwhile, there are also some other double-linear damage rules and nonlinear damage rules (Fatemi and Yang 1998), which could also be used in the fatigue design as needed.

Fatigue Resistance

The fatigue resistance gives the relationship of number of stress/tension cycle times that would lead to failure at different stress/tension range, and it should be based on a large amount of fatigue test data.

Generally, the fatigue design curves for tension-tension fatigue failure condition have been recommended by API (API RP 2SK 2008; API RP 2SM 2014) and Det Norske Veritas and Germanischer Lloyd (DNVGL OS E301 2015), and the tension-tension fatigue should be considered by all of the mooring line components. For the other types of fatigue failure, such as the out-

of-plane bending (OPB), torque and twist, and axial compression, the test data are insufficient for generating design curves.

Testing Qualification

It is also permissible to apply the actual test data for the design of a specific mooring line component rather than using design curves. During the test qualification, the test specimen, the test machine and apparatus, the test procedure, and the posttest examination should also match the requirements of corresponding standard. A linear regression analysis should be used to give the design curve, which generally located a little more than two standard deviations below the mean line, and the aim is to satisfy the prediction interval and exceedance probability. If the test is conducted in the air rather than in the seawater, a reduction of the fatigue life should also be considered.

T-N Curve

The T-N curve is recommended by API (API RP 2SK 2008; API RP 2SM 2014) and presents the number of cycles to failure for a specific mooring component as a function of constant normalized tension range, shown as Eq. (2).

$$NR^M = K \quad (2)$$

where N is the number of cycle times, R is the ratio of tension range to reference breaking strength (RBS), and M and K are the slope and intercept parameters in T-N curve, respectively, which are determined by material properties.

In Eq. (2), the stud chain links with loose stud cannot be applied, and the mean load should be included in the parameter K because it has important influence on the wire rope fatigue life.

For the common stud chain, studless chain, six/multistrand rope, spiral strand rope, and polyester rope, API has recommended the parameters value of M and K in the T-N curves.

For the other types of fiber ropes, the fatigue design should use the test data to qualify the fatigue life from the manufacturer with certification, which shall approve the test procedure.

S-N Curve

The S-N curve is recommended by DNVGL (DNVGL OS E301 2015), and it is the same approach as described in Eq. (2) with the change that the tension range R is replaced by the nominal stress range S . However, the parameters value of M and K in the S-N curve is different from T-N curve. The other considerations are the same as T-N curve.

Fatigue Damage Calculation

The fatigue life of mooring line components are estimated through comparing the long-term cyclic stress/tension with the fatigue resistance of that component. So, the long-term environment loading should be considered in the fatigue analysis, rather than the extreme environment loading used in the strength analysis.

The fatigue design under long-term environment can be represented by a number of short-term discrete sea states. Each short-term sea state has the probability of occurrence and consists of wind, wave, current parameters, and their incident directions, which could be required to calculate the mooring system motion responses. Generally, 8–12 reference directions and 10–50 reference short-term sea states provide a good representative of the long-term environment. However, the fatigue damage is sensitive to the number of short-term sea states, and the sensitivity studies should be conducted.

Annual Fatigue Damage

Based on the Eqs. (1) and (2), the fatigue damage of each short-term sea state D_i could be calculated by Eq. (3).

$$D_i = \frac{n_i}{K} E[R_i^M] \quad (3)$$

where n_i is the number of stress/tension cycle times in the sea state i , R_i is the normalized tension range using T-N curve or nominal stress using S-N curve, and $E[R_i^M]$ is the expected value.

The number of stress/tension cycle times n_i in each short-term sea state is determined by Eq. (4).

$$n_i = v_i \cdot P_i \cdot T_D \quad (4)$$

where v_i is the zero up-crossing frequency (Hertz) of the stress/tension process in the sea state i , P_i is the probability of occurrence of the sea state i , and T_D is the annual time (second).

Normalized Tension/Nominal Stress Range Calculation Method

The normalized tension/nominal stress range under the combined environmental loads could be calculated through the dynamic analysis method of mooring line or the model test method, and the static or quasi-static analysis method is not recommended. The dynamic stress/tension of mooring line could be divided into low frequency (LF) range and wave frequency (WF) range, and the fatigue damage could also be divided into LF fatigue damage and WF fatigue damage. The following four methods could be considered, in which the Methods I–III is based on the dynamic analysis of mooring line in the frequency domain and the Method IV is in the time domain.

1. Method I – Simple summation: the total fatigue damage is assumed to be the sum of LF and WF fatigue damages, which are calculated independently.

Method I is acceptable when the standard deviation stresses/tensions of WF divide the standard deviation stresses/tensions of LF is larger than 1.5 or smaller than 0.05. However, Method I may underestimate the fatigue damage when the LF and WF stresses/tensions both dominate the total stresses/tensions.

2. Method II – Combined spectrum: the total fatigue damage is calculated using the standard deviation of combined LF and WF spectrum, in which the combined LF and WF spectrum are calculated firstly.

Method II is always conservative and may significantly overestimate the fatigue damage.

3. Method III – Combined spectrum with dual narrow-banded correction factor: a correction factor is applied to the result of Method II, which is based on the two frequency bands that are present in the stress/tension process.

Method III is an improvement of Method II in order to decrease the conservative predictions, and it is suitable when the LF and WF stresses/tensions both dominate the total stresses/tensions. However, Method III will overestimate the fatigue damage when the LF stresses/tensions dominate the total stresses/tensions.

4. Method IV – Time domain cycle counting: the fatigue damage is calculated directly using a cycle counting method, such as the rainflow counting method, to obtain the fatigue load spectra from the time series of mooring line stress/tension. In other words, the histogram of stress/tension range, including the number of stress/tension cycles and the expected value of stress/tension range, could be obtained.

Method IV is the most accurate, but the analysis is relatively time-consuming.

Fracture Mechanics Approach

The fracture mechanics approach could also be used to conduct the fatigue damage estimation by modeling the crack growth under cyclic loading. The crack growth could be modeled by linear fracture mechanics based on the Paris-Erdogan equation (1963). The chain link could be treated as a round bar, and an initial surface crack is assumed to propagate at the surface, and then the stresses of chain link could be obtained through the finite element analysis (Xue et al. 2018). Meanwhile, the parameters sensitivity studies should be conducted when using the fracture mechanics approach, such as the initial crack shapes, the initial crack sizes, and the critical crack depths.

Fatigue Design Procedure

The fatigue design procedure for the mooring line is recommended as the following steps: (1) Discrete the long-term environmental into a number of short-term sea states. (2) Conduct the motion responses analysis of mooring line, and obtain the stress/tension process of mooring line. (3) Determine the fatigue design curves. (4) Calculate the

annual fatigue damage of each short-term sea state. (5) Sum of the step (4) results based on the P-M rule, and so it is the total fatigue damage. (6) Compute the fatigue life together with the step (5) and safety factor.

When selecting a fatigue design curve for the fatigue life assessment, the safety margin in all three integral parts (design curves, safety factor, and analysis method) could not be against experience and available test data.

For the polyester rope used as the free span in the mooring line, the fatigue design don't have to consider the degradation and should focus on the fatigue life of steel components because the fatigue life of polyester rope usually exceeds too much than that of steel wire or chain. However, the load-elongation properties of polyester rope have significant influence on the tension calculation of steel components.

Other Fatigue Conditions

Some of other fatigue conditions should also be considered in the fatigue design of mooring line, but not limited to the following.

OPB and Twist

The interlink rotations combined with the pretension of mooring chain could result in OPB fatigue failure at the terminations, such as the fairleads and hawse pipes (Morandini and Legerstee 2009). The wear of chain induced by the friction in the bearing must be considered, and the Archard theory of wear is suggested. The contribution to fatigue due to OPB could be solved by finite element analysis or test. The precautionary measures should be taken to avoid OPB fatigue failure, such as using a seven-pocket fairlead design, or keeping the diameter ratio of fairlead to line is large enough.

The mooring chain may cause twist under the torque effect in the adjacent link or even be inadvertently installed in a twisted condition, especially in the chain-wire/fiber-chain mooring line. The finite element analysis considering the wear or test could also be used to predict the twist fatigue damage. Meanwhile, the imposed twist

fatigue during the installation should be considered for the fiber ropes.

Axial Compression

The axial compression fatigue should be considered in an aramid rope when it is exposed to cyclic loading with a low trough tension, but it is not a concern with polyester and HMPE fiber ropes. The axial compression fatigue damage could cause strength loss, and keeping sufficient tension is benefit to prevent the axial compression fatigue damage.

Corrosion

The corrosion brings the fatigue into a corrosive environment, which could introduce mechanical degradation of a material under the combination of corrosion and cyclic loading. The simplest method of considering corrosion effect is to decrease the area of mooring line components, and the different corrosion velocities and models have been recommended. The coupled corrosion and wear effect may have significant influence on the fatigue damage estimation.

Cross-References

- [Design of Mooring System](#)
- [Dynamic Analysis Method](#)
- [Mooring System](#)

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FE, Finite Element

- [Numerical Simulation of Ice-Going Ships](#)

FEM – Finite Element Method

- [Analysis of Renewable Energy Devices](#)

Fiber Bragg Grating (FBG)

- [Fiber Optic Hydrophone](#)

Fiber Optic Based SPR (FO-SPR)

- [Fiber Optic Hydrophone](#)

Fiber Optic Hydrophone

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Synonyms

[Evanescent wave \(EW\)](#); [Fabry-Perot interferometer \(FPI\)](#); [Fiber Bragg grating \(FBG\)](#); [Fiber optic based SPR \(FO-SPR\)](#); [Fiber-optic sensor](#); [Hollow core PCFs \(HC-PCFs\)](#); [Index matching fluid \(IMF\)](#); [Mach-Zehnder interferometer \(MZI\)](#); [Photonic crystal fiber \(PCF\)](#); [Prism based SPR \(P-SPR\)](#); [Sagnac interferometer \(SI\)](#); [Solid core PCFs \(SC-PCFs\)](#); [Surface plasmon resonance \(SPR\)](#); [Surface plasmon wave \(SPW\)](#); [Ultra-violet \(UV\)](#)

Definition

Fiber optic hydrophone is a sensor used for underwater sound measurement. Its working principle is to obtain underwater sound-related characteristics by measuring the changes in the photosensitive properties of the optical signal transmitted in the optical fiber.

Scientific Fundamentals

For the past few decades, immense research has been carried out to monitor ocean parameters

adopting trending techniques. Among them, one of the notable approaches falls in the domain of optical fiber technology. The optical fiber sensing technique has an exclusive feature of single-ended operation, that is, one end of the fiber is connected to the instrumentation node, whereas the other end is left free. The free end of the fiber is allowed to interact with the water body. This feature allows in situ examination of ocean properties rather than testing a sample of seawater in the laboratory. In a nutshell, fiber optic sensing techniques allow real-time in situ measurements providing infinite immunity to electromagnetic interference. Added conveniences in using fiber sensors are that they are less in weight, occupy a small area, and ease of availability. Hence, almost all the sensing techniques cited in this entry are applicable to ocean observation.

In general, any sensing device while deployed in the ocean should be considered for its ability to withstand high pressure and temperature. Ocean pressure increases at the rate of 145 Psi for every 100 m depth. Commercial silica fibers have the ability to withstand pressure up to 20×10^4 Psi. Also, silica fibers can endure temperature up to 900°C beyond that the core and cladding begin to drift. Since the maximum temperature of seawater is 30.5°C (as with the Red Sea, the world's warmest sea), fiber sensors can operate without any degradation in its sensing property. In seabed and polar regions, the water temperature reaches 4°C and further below. Glass fiber and photosensitive fiber can withstand cryogenic temperature up to -170°C . Abovementioned features of fiber sensors make them a befitting candidate for ocean observation and also in cryogenic temperature sensing applications (Gupta et al. 1996).

Even though fiber sensors have a lot of attractive merits, there exist few technical challenges while designing them for specific applications. In oceanography, bio-fouling and corrosion are two main issues that are to be considered while deploying any underwater module for sensing applications. To overcome these contaminants, the surface of fiber sensors should be coated with a corrosion-resistant lubricant. By doing so, the inherent sensing properties of fiber could be altered, and hence extreme care should be taken

while designing ocean optics sensors. In order to protect fiber sensors from external impurities and also to increase easy handling, the packaging is essential. Practical encapsulations for fiber sensors are made of steel, glass fiber reinforced plastic, and carbon fiber reinforced plastic. A dedicated phase control system needs to be accompanied by optical fiber sensors to maintain the finest sensitivity throughout the deployment period (Abbas et al. 2018; Wang et al. 2018).

Researchers at the Naval Physical and Oceanographic Laboratory performed the packaging of fiber optic hydrophone using polymer shell. Hydrophones are underwater acoustic signal detectors that listen and records sound signals in a water medium. Apart from sensing ocean parameters, fiber optic sensors are also involved in structural health monitoring of marine structures. Due to their tiny size and flexibility, the optical fiber could be easily inserted in composite structures. Optical strain sensors are fixed along the hull of a ship to continuously monitor stress and strain components thus ensuring the reliability of marine structures. The underlying concept of any fiber optic sensors is their ability to modulate intensity, wavelength, phase, or polarization of the optical signal concerning variations in the sensing parameter (Doyle and Staveley 2003; Vivek et al. 2017; Kumari et al. 2019).

Historical Development

In 1977, Bucaro et al. have demonstrated the feasibility of employing directly acoustic-optic interactions in an optical fiber to produce a sensitive acoustic detector for the first time. The technique utilizes the phase modulation of an optical beam in a submerged optical fiber coil by sound waves propagating in a fluid. Analysis of the results indicates that the sensitivity of this technique compares well with that of the best available hydrophone. This research has brought a revolutionary change to the development of fiber optic hydrophone technology (Bucaro et al. 1977).

In the 1980s, Interferometric fiber sensors had been widely investigated for use in a range of application areas including acoustic

pressure and magnetic fields. As with conventional electronic sensor technology, the use of arrays of sensors was often desirable for such systems, and a number of multiplexing techniques for interferometric sensors have been described (Kersey 1991).

In the 1990s, fiber optic hydrophones had been demonstrated to have better performance than conventional hydrophones, and continuing research in fiber-optic acoustic sensor heads and the associated signal modulation/demodulation system had been actively pursued. To recover the phase information from the output intensity of the fiber-optic acoustic hydrophone, different signal detection and processing schemes had been developed. These schemes included passive homodyne, active homodyne, true heterodyne, synthetic heterodyne, pseudo heterodyne, and differential delay heterodyne (Lim et al. 1999).

In recent years, fiber optic hydrophones have developed rapidly. Fiber optic sensors play an important role in the application of underwater detection systems. Especially in military applications, fiber optic hydrophone arrays have become one of the most important means for submarine detection.

Fiber Optic Sensor Technology

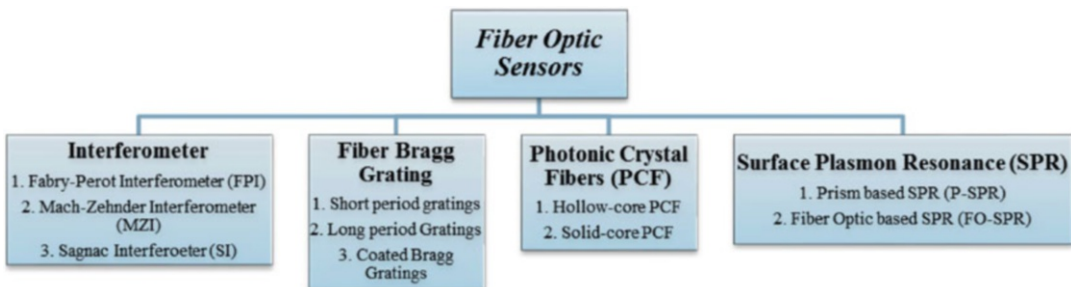
Figure 1 depicts a clear classification of optical sensors based on sensing methodology. The following sections focus on these sensing approaches in oceanographic applications.

Interferometric Sensing

Fiber optic interferometry is a technique that makes use of the fundamental principle of light interference for measuring changes in the physical and chemical properties of sensing medium. An interferometer could be designed using precise fiber optic components such as fibers and waveguides that control optical path. Optical fibers are preferred as the basic component to build any interferometric sensors because of their sensitivity towards elongation (change in physical length), less power consumption (in the range of micro- and milli-watts), and ability to reduce the effect of wavefront distortion caused by atmospheric turbulence. In addition to optical fibers, photonic splitters and combiners should also be accommodated to obtain a working model of Interferometric sensors.

In general, interference is defined as the superimposition of two or more light beams and interferometry is the measuring technique that evaluates phase difference between combined light beams. Variation in the sensing parameter could be estimated from the interference pattern. Based on the operating principle, interferometric sensors are classified as Fabry-Perot Interferometer (FPI), Mach-Zehnder Interferometer (MZI), and Sagnac Interferometer (SI).

FPI is an optical resonator comprising of two reflectors. The reflectors may be either planar mirrors or curved mirrors with 90% reflectivity. The distance of separation between the mirrors determines the bandwidth of resonance and wavelength resolution of FPI. Resonance bandwidth increases with the decrease in spacing between



Fiber Optic Hydrophone, Fig. 1 Classification of fiber optic hydrophone techniques (Kumari et al. 2019)

the reflectors, whereas the resolution of wavelength increases with increased spacing. Hence, a tradeoff should be made between these two factors (bandwidth and resonance) to successfully realize an FPI as a sensor.

MZI makes use of a beam splitter to split an input light into two different beams. To recognize MZI as a sensor, one of these two beams is allowed to interact with the surrounding environment whose parameters are to be sensed. The other beam is enclosed within a vacuum chamber thus behaving as a reference arm. These two beams are then combined at the output end using a beam combiner and the phase difference between them is measured. The measured phase difference is then interpreted to study the desired environmental parameter.

SI can be designed using two optical fibers with multiple wires to increase the path length to achieve high sensitivity. Like MZI, the bottom line for SI is measuring the phase difference between two beams that come out of two multi-loop optical fibers (Kumari et al. 2019).

Fiber Bragg Grating (FBG) Based Sensing

FBG can be defined as a series of in-fiber selective mirrors with specific refractive index modulation along the axial length of the fiber core. These mirror structures are inscribed inside the fiber core using femtosecond lasers or Ultra-Violet (UV) lasers. FBGs fabricated with femtosecond lasers have a central wavelength in the range of 790–850 nm and 1520–1590 nm, whereas using UV lasers FBGs could be obtained with a central wavelength ranging from 970 to 1620 nm. Germanium and boron-doped fibers are preferred for FBG fabrication as they have inherent photosensitive properties.

Grating sensors are mainly classified as a short or a long period based on the length over which grating is written. Bare FBG has a built-in sensitivity to changes in temperature and pressure. Hence to make FBG respond to a single parameter, coated FBGs were fabricated and their characteristics were analyzed. FBGs coated with metals and polymers possess excellent repeatability and sensitivity while used for sensing the

physical and chemical parameters of the ocean (Kumari et al. 2019).

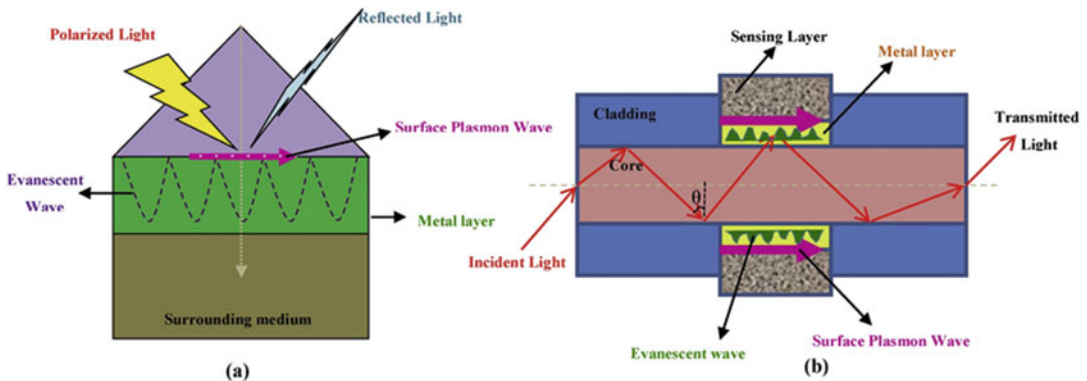
Photonic Crystal Fiber (PCF) Based Sensing

PCFs are a class of fibers that are made of pure fused silica. The cladding of these fibers is formed of air holes that are distributed in a well-arranged manner around a central hole (the core). PCFs are mainly used as optical tunable devices by filling the air holes with suitable Index Matching Fluid (IMF). The Refractive index of IMF varies with respect to the surrounding temperature and electric field. This attractive feature of PCF makes it suitable for environmental sensing applications. Liquid to be filled should have a greater index of refraction compared to that of the enclosing medium. IMF can be injected inside specific air holes using a fusion splicer or by locating the tip of the air hole under a microscope (Xiao et al. 2005; Witkowska et al. 2006).

Based on the core structure, PCFs are classified into two types: hollow-core and solid-core PCF. As the name indicates, Hollow Core PCFs (HC-PCFs) have air-filled cores that are mainly used in the generation of Raman comb frequencies, high field optics, and coherent optics. HC-PCFs are also employed in discharge-based laser design and laser metrology. Liquids or gases can be filled in the core of HC-PCFs making them suitable for sensor design, the transmission of high-power optical pulses, and also in the bio-medical stream. On the other hand, Solid Core PCFs (SC-PCFs) are made of air holes surrounding a thick core of silica and work on the basis of total internal reflection (Ramsay 2008).

Surface Plasmon Resonance (SPR) Based Sensing

SPR is an optical phenomenon in which resonance of free electrons occurs on the surface of a metal-dielectric interface. The resonating wavelength varies with respect to variation in the sensing medium. Based on the system configuration, SPR sensors are classified as Prism based SPR (P-SPR) and fiber optic based SPR (FO-SPR).



Fiber Optic Hydrophone, Fig. 2 SPR based sensing methodology, (a) prism based SPR and (b) fiber optic based SPR (Sharma et al. 2007)

As the name indicates, PSPR uses a prism for reflection of resonance signal, whereas in FO-SPR the fiber itself involves the transmission of a resonant wave. The resonance of Surface Plasmon Wave (SPW) is related to the wave vector of Evanescent Wave (EW) that travels along the surface of the interfacing medium. In P-SPR, p-polarized light is incident on a metal-coated prism. EW starts to originate on the metal-prism interface. This EW component excites surface plasmon on the metal-dielectric (metal-air) layer (Fig. 2a).

In FO-SPR, the prism is replaced by an optical fiber in which cladding is chemically etched using 40% hydrofluoric acid. The cladding-less portion of the fiber is then coated with a metal (usually with gold or silver) (Fig. 2b). Along with the core-cladding interface, EW starts to propagate when input light is incident into the fiber with an optimum angle of incidence " θ " such that total internal reflection eventuates. As in P-SPR based sensors, SPW originates along with the interface of a metal and surrounding sensing layer. Any change in parameters of the sensing medium induces a shift in resonant wavelength (Sharma et al. 2007).

Even though the fundamental sensing technique is the same for both SPR methods, P-SPR uses angular interrogation, whereas FO-SPR relies on spectral interrogation techniques to obtain a calibration curve for specific sensing applications. Compared to P-SPR, FO-SPR is

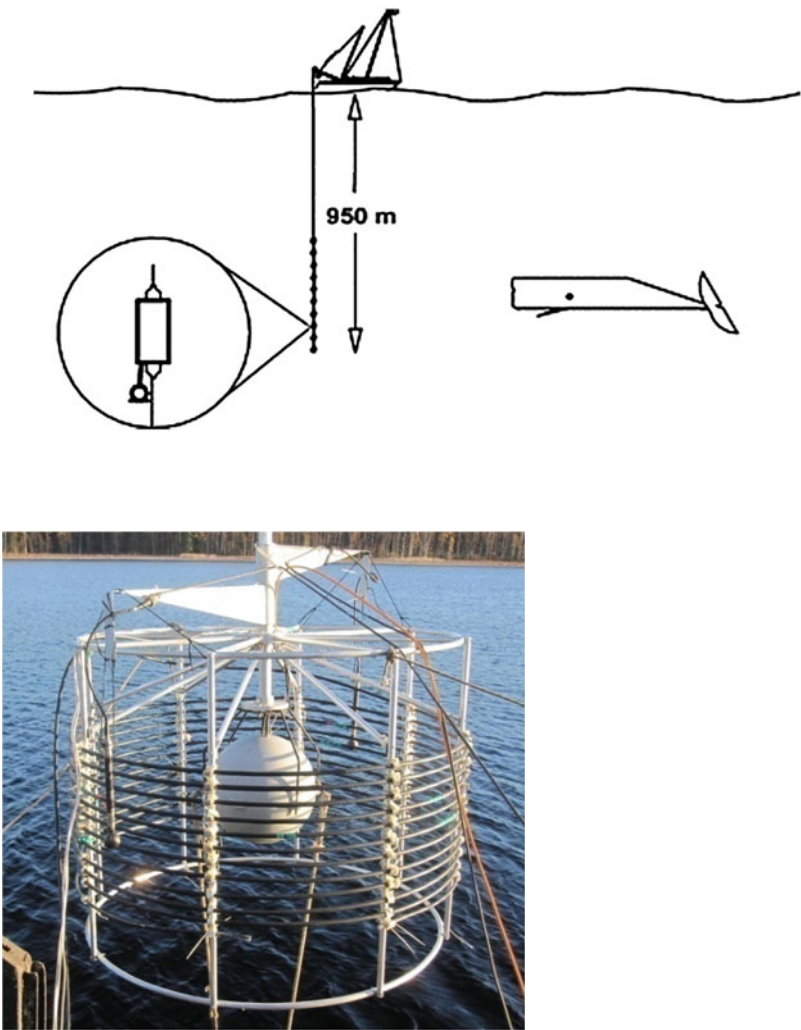
preferred because of its lightweight and ease of examination. The design of the sensor head includes U-shaped and side-polished fibers (Gupta and Verma 2009).

Key Applications

A ten-element, 950 m long, vertical hydrophone array based on fiber optic data transmission has been developed primarily for studying the beam pattern from deep-diving cetaceans emitting sonar pulses, as shown in Fig. 3. The array elements have a configurable sampling rate and resolution with a maximum signal bandwidth of 90 kHz and a maximum dynamic range of 133 dB (Heerfordt et al. 2007).

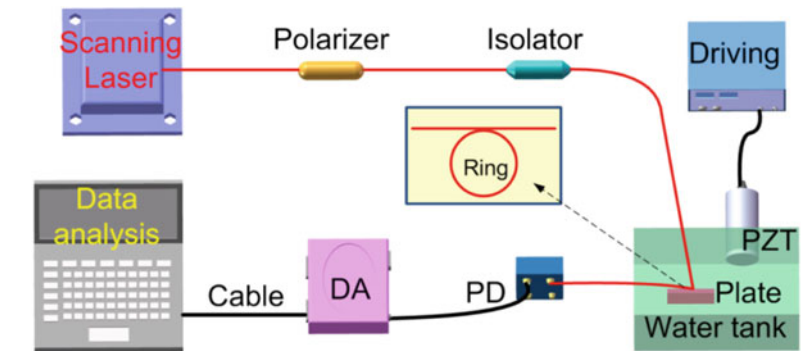
Lavrov et al. (2017) have presented the results of experimental investigations of the fiber optic hydrophone array consisting of six sensors, placed in one thin sensitive cable, as shown in Fig. 4. Sensors were formed by pairs of Bragg gratings spaced 1.5 m apart and recorded in a birefringent optical fiber with the elliptical stressed coating. To form an extended sensor array, the optical fiber was additionally covered with silicone material RTV655 and protective coatings. The obtained results might be used for developing and producing long thin hydroacoustic arrays for geophysical investigations and other hydroacoustic applications.

Fiber Optic Hydrophone, Fig. 3 The concept of the fiber optic linked deep-sea hydrophone array. The enlarged section shows a canister with a hydrophone attached below it (Heerfordt et al. 2007)



Fiber Optic Hydrophone, Fig. 4 Measurement stand for fiber-optic hydroacoustic sensor array (Lavrov et al. 2017)

Fiber Optic Hydrophone, Fig. 5 Schematic configuration of the experimental setup for acoustic measurements. PD is the optical power detector, and DA is the data acquisition card (Chen et al. 2019)



Chen et al. (2019) have proposed a compact fiber-optic hydrophone and use it for acoustic wave measurements, as shown in Fig. 5. The sensor probe comprises a microresonator formed by simply tying a tapered fiber into a loop structure. After fabricating the sensor, the knot exhibits low-absorption loss underwater and, thus, the device can possibly be used as a hydrophone. The simple fabrication and small size structure of the sensor make it a good candidate for high-resolution scanning imaging.

Cross-References

► Fiber Optic Hydrophone

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Fiber Optics

► Fiber-Optic Cable

Fiber-Optic Cable

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Synonyms

Center pipe; Fiber optics; Laminated structure; Mode fibers; Optical; Optical fibers

Definition

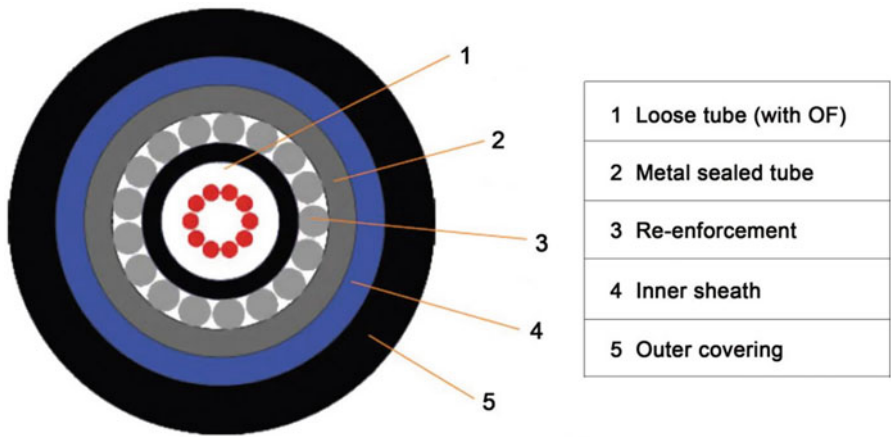
It is with the purpose of meeting the performance specifications of optics, machinery, and the environment that optical fiber cables are made. Communication cable assemblies with one or more optical fibers in the cladding sheath of optical cables are made used as the transmission medium and can be used individually or in groups. In the structure of optical cables, a certain amount of fibers is used to form the cable core in a certain way, some of which are covered with outer sheath. Generally speaking, optical fiber cables are communication lines used to transmit optical signals (Hogari et al. 2008). Although there is certain tensile strength of optical fibers, the strength is not strong enough to withstand the practical situations of bending, twisting, and side pressure. Thus, the traditional cable which contains technology should be used, such as fitting, overlaying plastic, metal tape armoring, and so on, just like the copper cables of communication. And then, in order to make a variety of optical cables which can be used in different environments, the strength component materials are placed in the cable core, so that it can meet the laying conditions required by the projects; achieve the mechanical performances of resistance to tension, impact, bending, distortion, and so on in practical situations; and ensure the original

great transmission performance of the unchanged optical fibers (Shimizu 2008).

Scientific Fundamentals

Basic Structure

Optical cable is mainly composed of optical fiber (glass which is as thin as hair), plastic protective sheath, and plastic outer layer. It can be divided into fiber core, reinforcing steel wire, filler, and sheath. In addition, there are waterproof layer, buffer layer, insulation metal wires, and other components according to the demand. To some extent, the performance of the optical fibers has an impact on the performance of optical cables. Fiber core is the most core component in optical transmission materials of optical fibers, which can be of the glass or the plastic kind. It has a high refractive index, which is used to transmit light (Sato 1991). The thicker the fiber core is, the more the light can be transmitted through the optical fiber. The wrapping provides a low refractive index at the core interface to form the waveguide effect, forming a total reflection with the fiber core. Therefore, it can restrict the light in the optical fiber. The fiber coating/protective sheath is usually made of multilayer plastic, which has a high strength and can withstand large impact so that it can maintain the strength of the optical fiber, absorb vibration, and protect fiber core and wrapping (Li et al. 2009) (Fig. 1).



Fiber-Optic Cable, Fig. 1 The structure of optical fiber cable (http://www.orientcable.com/v_pros.asp?/35.html)

Classification

Optical cables can be classified into the following five common types according to the shape and the number of fiber cores.

Optical Cable with Center Pipe

This optical fiber is usually in the center of the center pipe, which has small cross section and is light weight. The loose pipe is located in the physical center of the optical cable, which is helpful for bending of the cable (Kurashima et al. 1991). The reason why it is usually filled with moisture-proof ointment is that it is for ensuring moisture and no longitudinal water seepage. The fiber cores of this optical cable are few, less than 12 cores.

Optical Cable with Laminated Structure

The optical cable with laminated is usually made of optical fiber with loose sheath. The structure of this kind is widely used all over the world. It is a common optical cable which is used in early optical communication and which is usually made up of 6–12 core optical fibers (Hataji 1991). The protective layer is slightly different according to the laying requirement of pipe, overhead and directly buried. All in all, the structure is complex, and there are much various steps in the manufacturing process.

Optical Cable with Framework

The optical fiber of this type locates in the frame slot. Framework materials are made of low-density polyethylene, while reinforcing cores are made of multistrand steel rope or reinforcing plastic. The number of fiber core in this optical cable is large so that it can solve the problem of transmitting congestion and shorten the connection time of optical fibers. Meanwhile, there are few wrong connections because of the optical fibers which are easy to identify (Hogari et al. 2010).

Optical Cable with Bundle of Tube

The structure of optical cable with bundle of tube style is the most reasonable for the protection of the optical fibers. There are two single steel wires in this structure which are placed in the sheath and which are used as two reinforcing cores. The

strength of this optical cable is great, especially the resistance of the lateral pressure (He et al. 2013). The number of the optical cable with bundle of tube style is flexible. In other words, the range of the number of the optical fibers is large.

Optical Cable with Tape

The optical cable with tape can hold a lot of optical fibers. The tape units of this cable are stacked into a rectangle and twisted into a cable with a certain pitch so that it can have a better bending performance.

Basic Property

Attenuation

Attenuation refers to the attenuation of optical per unit length with dB/km. As a result of the attenuation, the energy signal in optical fiber can be continuously attenuated. In order to realize long-distance optical communication and transmission, it is necessary to set up a relay station at a certain distance to enhance the attenuated signal repeatedly. Generally, attenuation is caused by two factors, and they are the internal factor and the external factor, respectively. The internal factor is mainly due to the absorption attenuation and scattering attenuation of materials. There are some additional elements in general optical glass, many of which are impurities. And most of them have electron states with low excitation energy (Kojima et al. 1982). Meanwhile, there are also some metal ions whose electron states are more easily excited than the intrinsic properties of glass. Their absorption band can appear in the visible and infrared areas of the spectrum with strong absorption attenuation. The attenuation which is caused by impurity-absorbing energy is called absorption attenuation. The scattering attenuation mainly comes from the fabrication defect and intrinsic scattering of optical fiber, which is mainly due to the fluctuation of refractive index. In refractive index, microcosmic local changes may be resulted from the random molecular structure in optical fiber materials and so do the defects and the impurity atoms. In addition, the attenuation also comes from the bending of the optical fiber and the attenuation caused by the coupling of

optical fiber when the system is constructed. Bending attenuation of optical fiber includes macro and micro bending attenuation (Hodgson et al. 2004).

Dispersion

The optical signals (pulse) with different frequency components or mode components transmit in the optical fiber at different speeds. And the signals are affected by delay distortion, which is determined by factors such as refractive index distribution of optical fiber, dispersion characteristics of optical fiber materials, pattern distribution in optical fiber, and spectral width of light source. And when a certain distance is reached, the signal distortion (pulse broadening) occurs, which is called dispersion. Dispersion can mainly be classified into material dispersion, waveguide dispersion, polarization mode dispersion, and inter-mode dispersion. Among them, inter-mode dispersion has the characteristic of the multimode fiber (Komizo et al. 2008).

Bandwidth

Bandwidth is the product of the modulation frequency and the fiber length of the maximum modulation frequency pulse that a fiber can pass through. Also, it is a comprehensive index which is used for characterizing the optical properties of multimode fiber. The narrower the bandwidth is, the wider the digital signal is spread when transmitting. And error codes are generally caused by the overlapping of code elements. In order to meet error index, the code element spacing has to be increased, leading to the decrease of the communication capacity (Toge et al. 2009). On the other hand, the extent of fiber to pulse broadening increases with the increase of transmission distance. Therefore, transmission distance cannot be too long so that it can reduce error codes.

Cutoff Wavelength

Another important parameter to measure optical cables is the cutoff wavelength, which is related to the length and the bending state of fiber. Cutoff wavelength refers to the fact that single-mode fiber usually has a certain wavelength. When the

wavelength of light transmitted is over that, the fiber only transmits lights with one mode. And when the wavelength of light transmitted is below that, the fiber can transmit light with a variety of modes (Takai and Yamauchi 2009).

Mode Field Diameter

Mode field diameter is an important performance parameter of single-mode fiber. It can be derived from the parameters of equivalent step fiber, estimate the connection and bending loss of single-mode fiber, and characterize the distribution of fundamental mode light in the core area of single-mode fiber (He et al. 2013). The light intensity of the base mode is the maximum at the axis line in the fiber core area, which decreases with the distance from the increasing axis line. Generally, mode field diameter is related to the wavelength, which increases with the increase of wavelength (Chu et al. 2004).

Active Area

The optical energy is not completely concentrated in the fiber core, part of which is transmitted in the cladding in optical fiber. Since the core diameter cannot reflect the energy distribution in the fiber, the concept of active area is put forward. When the active area is small, the density of cross section of optical fiber is large. Too excessive density will lead to the nonlinear effect, which will reduce the optical signal-to-noise ratio of optical communication system and affect the performance in the system. Therefore, the larger the active area is, the better the performance is for transmission fiber.

Numerical Aperture

Numerical aperture refers to the light that enters the end of optical fiber, which cannot be transmitted by the optical fiber completely. It is just the incident light at a certain angle which is called numerical aperture of fiber (Boscher et al. 1982).

Mechanical Property

The mechanical properties of optical fiber consist of stretching, flattening, impact, cycle bending, torsion, sheath wear, and loose casing bending. And

there are some factors that should be measured, such as tensile strength, stripping force, and dynamic fatigue parameters (Nishio et al. 1999).

Key Applications

In general, optical fiber cables are the cables that contain optical fibers. In 1976, a 144-fiber cable made by Western Electric Company was used in Atlanta by Bell Institute of America, representing that the first optical fiber communication experimental system was built. In 1980, commercial optical fiber cables made of multimode fibers began to be used for intercity truck lines and a few long-distance lines. The commercial optical fiber cables made of single-mode fiber have been used for long-distance lines since 1983. The first transoceanic fiber-optic cable between the United States, Great Britain, and France was TAT-8. It was constructed in 1988 by a consortium of companies led by AT&T Corporation, France Télécom, and British Telecom. AT&T Bell Laboratories developed the technologies used in the cable. It was able to serve the three countries with a single transatlantic crossing with the use of an innovative branching unit located underwater on the continental shelf off the coast of Great Britain. The cable lands in Tuckerton, New Jersey, USA; Widemouth Bay, England, UK; and Penmarch, France. Since then, the first subsea optical fiber cable across the Atlantic was successfully built. Until 2018, the length of the global subsea fiber-optic cables has reached 900,000 km totally, carrying more than 90% of the communication data transmission in the world (Toge and Hogari 2007).

Human society has developed into an information society so far. The amount of information exchanged is so enormous (such as sound, images, and data) that traditional means of communication can no longer meet the demands of the modern society, providing the condition of optical fiber communication widely used due to its advantages of large information capacity, good confidentiality, light weight, small volume, and long distance of non-relay segment. The applications of optical cable almost cover all industries, which

are moving to a wider and deeper level (Haro and Horche 2013).

Cross-References

- Center Pipe
- Laminated Structure
- Optical Fibers

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Fiber-Optic Gyro (FOG)

- [Optical Compass](#)

Fiber-Optic Gyro Attitude Reference System

- [Optical Compass](#)

Fiber-Optic Gyro Motion Sensor

- [Optical Compass](#)

Fiber-Optic Sensor

- [Fiber Optic Hydrophone](#)

Field Development

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Synonyms

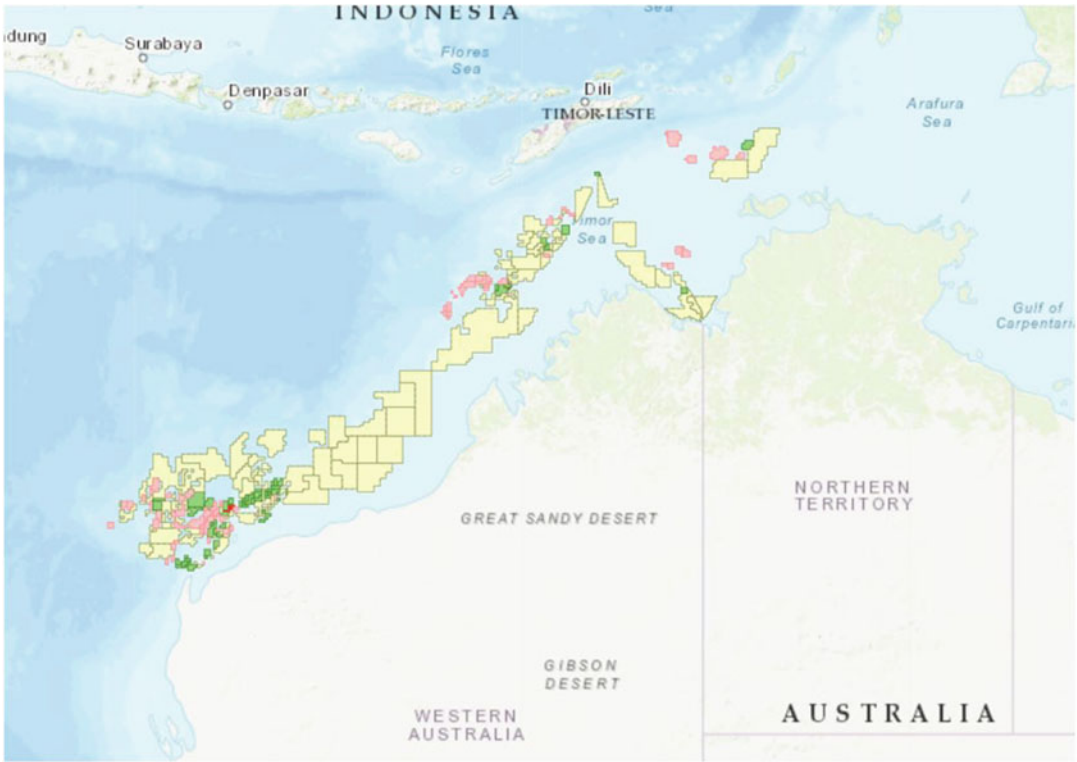
[Offshore project development](#); [Project execution plan](#)

Definition

Field Development is defined as the process by which an offshore deposit of hydrocarbon fluids is identified and developed through a structured process. While the process varies in detail between different parts of the world (influenced by differences in government processes, the nature of the hydrocarbons and the nature of the marine environment, including water depth), many of the processes are very similar while being named each with a plethora of inconsistent acronyms. Where examples are given herein, they are intended to be illustrative and not infer any preference, advantage, or disadvantage over alternates.

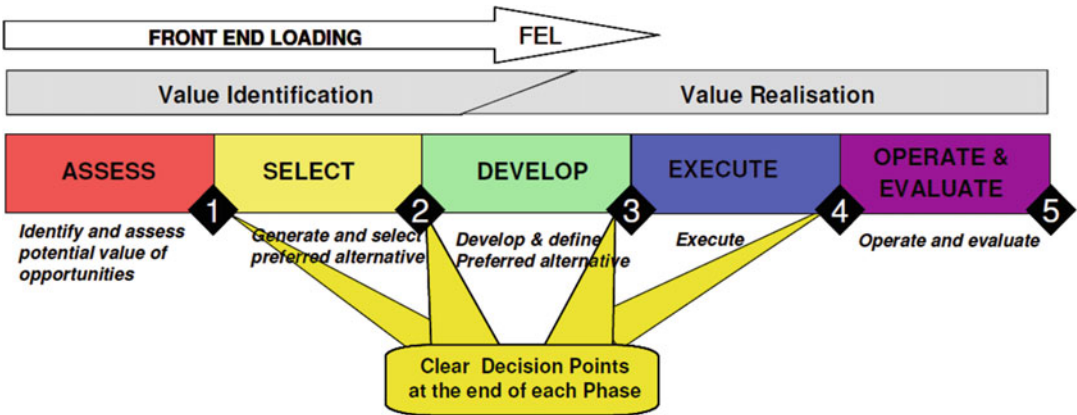
The process of Field Development is staged, with the highest-level summary of the staged process outlined as follows:

- **Discovery:** The hydrocarbon resource (or prospective resource) is identified.
- **Licensing:** The developer is selected through some form of competitive or structured allocation of the prospective area (permit area) as illustrated in Fig. 1.
- **Resource Appraisal Phase:** The hydrocarbon resource is evaluated and confirmed – potentially including activities such as detailed sub-surface surveying the area and/or the drilling of exploration or appraisal wells to confirm the presence and extent of the resource.
- **Project Development Phase:** A process of planning, engineering, and evaluation is undertaken in parallel with this resource quantification culminating in a Final Investment Decision (FID) on whether to proceed with the development, as illustrated in Fig. 2.
- **Construction and Pre-commissioning Phase:** The equipment, structures, and components of the Field Development are then procured, manufactured, transported, installed, and prepared for operation, including drilling and completion of the wells, subsea production system, host facilities, and export system.
- **Operational Phase:** First operation then takes place in a controlled process to commence the production phase of the Field Development.



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Field Development, Fig. 1 Typical permit area map (NOPTA 2021)



Field Development, Fig. 2 Typical Field Development staged process (Brennan 2004)

- Following which, the field equipment and reservoir are monitored, maintained, and repaired for the producing life of the field.
- Brownfield Redevelopment Phase: Often, fields that have been producing for some time

experience a slowdown in the rate of production, which is managed through Brownfield Redevelopment where additional fields or wells are developed and brought on stream to continue production.

- **Decommissioning Phase:** After which time, when the reservoir(s) is(are) depleted, the Field Development is shut in and prepared for decommissioning.

Discovery

National governments tend to be extremely proactive in encouraging the development of their country's natural resources, including offshore hydrocarbon reserves. While these developments may generate environmental, safety, and community consequences, they often also generate significant national wealth and provide energy security through either domestic consumption of the energy produced or export revenue.

Therefore national governments often support the performance of baseline surveys of their exclusive economic territory (both marine and terrestrial) to identify prospective sites where the geology is interpreted to support the presence of (in this case) hydrocarbon fluids, as illustrated below from Geoscience Australia, which is a statutory body created and funded by the Australian Federal Government (Fig. 3).

Licensing

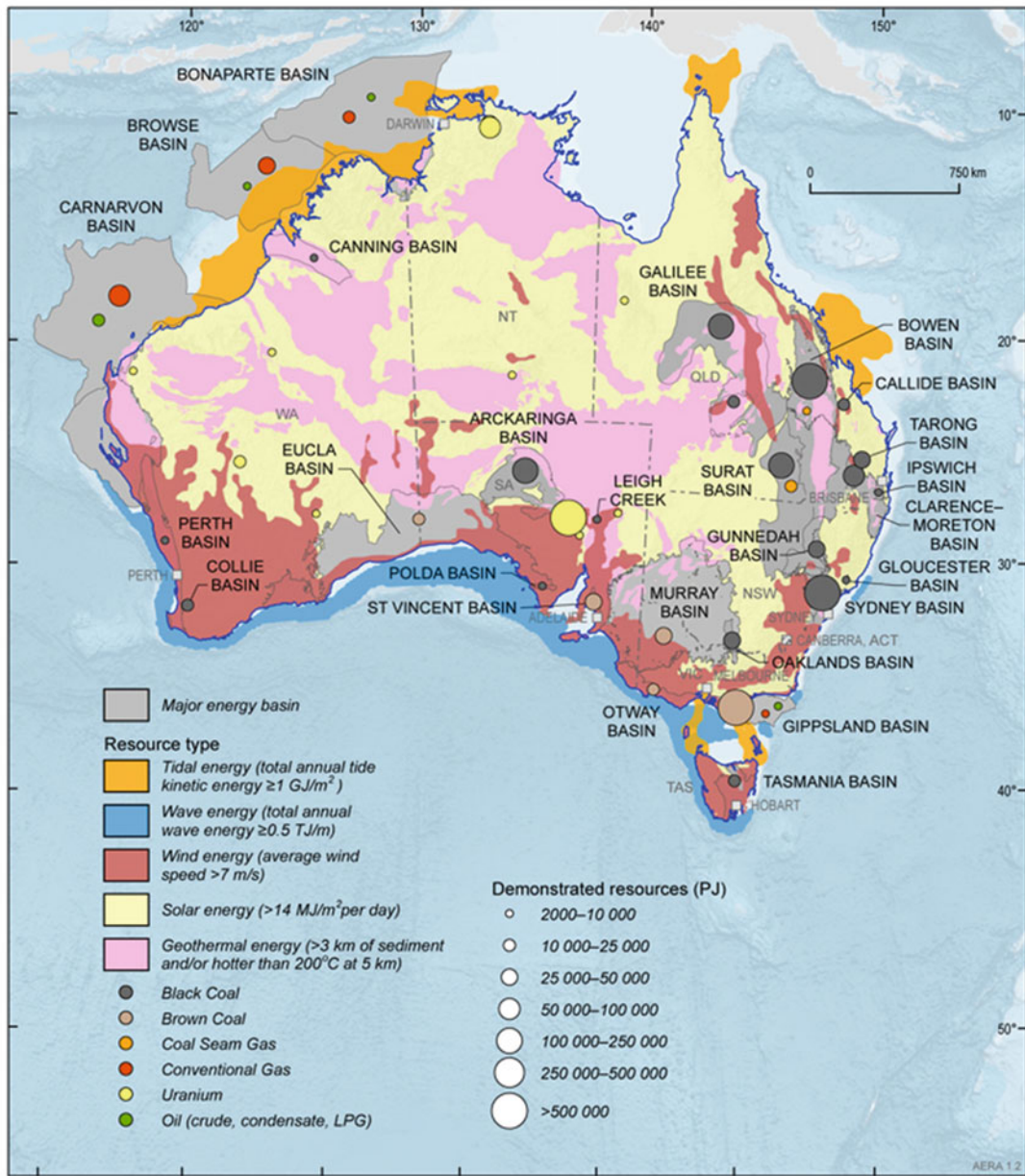
The awarding of locations (blocks as shown in Fig. 1) for exploration and Field Development is typically undertaken by periodic auctions by the National government. The auction typically involves commitments by the prospective Operators to undertake expenditure and work within a given timeframe on further appraisal and evaluation of the potential resources in a block, including detailed sub-bottom seismic surveys, and drilling of exploration or appraisal wells. Where Operators fail to undertake the work committed, or following identification and confirmation of resources being present if the Operator fails to progress towards Field Development (so-called fallow fields), then the license can lapse, resulting in the national government reacquiring the block. It is not uncommon for licenses for blocks to be awarded to consortia of Joint Venture (JV)

partners or for the post-award redistribution of ownership of a block to occur (so called "farm in" agreements) where an incoming participant may "pay" for their share in the future production value of the license through performance of work on the resource appraisal.

Resource Appraisal Phase

The resource appraisal phase consists of a number of distinct parallel activities to define the optimum Field Development concept, including:

- Improved definition of the hydrocarbon resource, including detailed sub-bottom seismic surveys, and drilling of exploration or appraisal wells where the type, quality, and flow properties of the reservoir can be confirmed, as illustrated in Fig. 4.
- Planning and engineering for the required wells, subsea, and surface facilities to support the production from the field. This process has different names and acronyms for the different Operators but in essence consists of a staged process (with an example shown in Fig. 2) of consideration of alternatives, selection of a preferred option, and progressive refinement of the selected concept.
- Field surveys and data acquisition, modelling, and interpretation of site conditions including geophysical, geotechnical, metocean and atmospheric aspects.
- Engineering and design activities sufficient to provide adequate definition of the preferred concept and likely costs, including supporting the key decisions on the Field Development configuration, phasing, production, processing, and export systems – commonly referred to as FEED (Front End Engineering Design).
- Management due diligence and investigation, including evaluation and analysis of environmental and community impacts from the Field Development.
- Commercial negotiations with both energy purchasers where long-term sales contracts are required to support the development



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Field Development, Fig. 3 Typical national resources characterization map (Geoscience Australia 2014)

(typically required for LNG projects) or market modelling to determine the likely commercial performance of the Field Development where the product is anticipated to be sold into a freely traded “spot market” (such as typically the case for oil or condensate projects).

- With vendors for the supply of the equipment and services required to construct the Field Development and commence the Operational Phase.
- Commercial and management negotiations with the joint venture partners to secure agreement on the development concept, often including



Field Development, Fig. 4 Typical appraisal well flow test (TestWells 2021)

negotiations with the partners in adjacent blocks with a view to “collaborative” developments spanning across multiple licenses. This can become necessary where a resource is found to extend beyond the boundary of a single block.

These processes culminate in a proposed Field Development plan being put to the senior management or board of directors of the Operator and associated JV partners for approval, known as a “Final Investment Decision,” to either proceed with the development or recycle the concept for further evaluation, reconfiguration, and reconsideration. It is not uncommon for FID decisions to be deferred or Field Developments abandoned, depending on the global economy and level of business confidence.

Project Development Phase

Following sanctioning of the Field Development (FID), the Operators and JV partners have

committed to the expenditure of the approved budget and for the project to proceed. The initial steps are the awarding of contracts to suppliers and contractors for the longest-lead items, which either require significant resources or time to manufacture. Examples include the production facilities such as platforms (e.g., the jacket shown in Fig. 5), linepipe where long-distance export pipelines are required, or for subsea equipment such as Christmas trees where market congestion with very few qualified suppliers means that order durations significantly exceed the actual manufacturing time.

The Project Development Phase also consists of the detailed engineering design of all of the equipment and facilities for the project, with this phase characterized by the analysis of all elements of the project in a high level of detail, while there exists a very low level of tolerance for any changes which impact on other systems or elements of the development. In this phase of the project, management of change processes, quality assurance, and interface management pose some of the greatest



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Field Development, Fig. 5 Typical fabrication site (Tamar Jacket fabrication, Deltec Group 2017)

challenges to successful delivery on-time and on-budget for the Field Development.

Construction and Pre-commissioning Phase

Careful planning of the in-field construction and commissioning phase is undertaken to ensure that the sequence of activities are performed in a safe and efficient manner, with vessels, crew and equipment mobilized, and activities completed to minimize idle time or weather downtime.

The construction sequence for different elements of the system depends on the nature of the Field Development concept selected, but each development option will result in a clear sequence of activities to ensure success of the project

development. As an example, for a conventional oil or gas platform such as the one illustrated in Fig. 6 below, the jacket must be installed and piled prior to the topsides being installed, prior to the production platform wells being drilled. The export pipelines may be installed before or after these activities, but hookup and pre-commissioning (preparing the facilities for first operation) cannot take place until each of the major elements is in place.

The pre-commissioning phase is last in this sequence and involves testing and preparing each element of the production process for introduction of hydrocarbons and commencement of production. This typically involves cleaning; pressure testing of all connections; deoxygenating of all process pipework, vessels, and pipelines; and preparation of baseline as-built documentation of the entire system ready for handover to the



Field Development, Fig. 6 Typical construction activities (Herema 2020)

operations team. The process also includes verification certification and quality assurance processes.

Operational Phase

On commencement of production, the Field Development enters into the Operational Phase. This phase is often associated with intermittent start-up, short-duration production, and process trips as any residual construction; commissioning or technical difficulties are resolved. Equipment failures or production problems are most frequently encountered either in this initial phase of operation or late in the field life once corrosion, fatigue, and other late-life integrity issues become more prevalent – these two periods of enhanced production disruption are known as the “bathtub failure curve.” The initial set of production disruptions are the most costly in terms of deferred production and net present value for the field.

Once reliable production is established, the Field Development typically enters a relatively static phase of routine production where the

hydrocarbons are produced and exported to market, with only (a) routine planned maintenance shutdowns or (b) exceptional disruptions due to severe storms where the facility has not been designed to remain permanently on station. Unlike some FPSOs which are designed to disconnect and move out of the path of tropical revolving storms (cyclones/typhoons/hurricanes), many facilities are designed to remain in production on station in even the design extreme storm event, for example, the Shell Prelude facility shown in Fig. 7 which has recently entered the operational production phase.

Brownfield Redevelopment Phase

Many Field Developments experience extended operational lives due to their redevelopment. The motivations for this can be to extend the producing life of the host facility in order to maximize the total production and therefore revenue from a facility. Motives can also be to prolong the producing life in order to postpone the considerable financial decommissioning liability. It is relatively common for Brownfield



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Field Development, Fig. 7 Typical operational activities (Shell 2021)

Redevelopment to also coincide with transfer of ownership of the field from the original Operator responsible for the initial development, to a much smaller Operator. These smaller Operators may specialize in the operation of many such fields on a “low-cost” model of minimizing maintenance and operational overheads to extend the profitable lifespan of the Field Development. Redevelopment options commonly employed include:

- Subsea tieback of additional smaller reservoirs located in proximity to the host facility, thus increasing the total reserves accessed by the Field Development
- Reworking of the existing wells in order to enhance the production from the existing reservoir, including redrilling new wells from within the bore of existing wells
- Implementation of enhanced recovery methods, including the installation of additional waterflood wells and gas injection wells or other strategies for increasing the depletion of the existing reservoirs
- Adding subsea compression, subsea pumping, or other enhanced processing facilities to better enable efficient production from the existing

wells as the reservoir experiences late-life production phenomena such as increasing water production, greater gas/oil ratios, or decreased reservoir drive due to lowering reservoir pressure

Each of these Brownfield Redevelopment Phase actions can be considered, in some respects, as a new “Field Development” and follows most of the same steps outlined as for the original Field Development.

Decommissioning Phase

Following permanent cessation of operation/production from the Field Development, the facilities enter into the Decommissioning Phase. This phase can be characterized by a number of distinct sub-phases, which include:

- A nonoperational period while the Operator determines the preferred decommissioning plan, together with negotiation and seeking agreement and approval from statutory authorities for this decommissioning plan. Key elements involve the plans for plugging and

making safe of the wells and removal or disposal or “decommissioning in situ” of the seabed, surface and topside facilities, pipelines, subsea production, and other equipment.

- A deconstruction period where:
 - The wells are typically plugged with concrete, wellheads removed and the casing/well cut off below the mudline.
 - Removal of the surface facilities such as manifolds, platforms, topsides, risers, moorings, and floating facilities such as TLPs, FPSOs, or CALM buoys.
 - While many jurisdictions mandate removal of all installed equipment, some elements may be considered for decommissioning “in situ” such as pipelines and flowlines. The relative merits of this involve balancing potential environmental benefits from the equipment acting as a nucleating site to enhance the natural marine habitat, versus the residual impact of potential ultimate decay of the materials – which can include metal and plastic used in the original manufacture as well as some contamination from production such as mercury, normally radioactive materials and other possible pollutants.
- An indefinite residual liability period where the wells, reservoirs, and any remaining offshore equipment remain in the marine environment. While some jurisdictions transfer this residual liability to the national government, others leave the liability with the Operator in perpetuity.

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TestWells (2021) image courtesy Well Test Support and eLearning – Oil and Gas Well Testing, TestWells Ltd www.testwells.com

Filler and Wrap

- [Umbilical Cable](#)

Finite Element Method (FEM)

- [Aquaculture Structures: Numerical Methods](#)

Fish Cage

- [Traditional Aquaculture Structures](#)

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Fisher

- [Fishing Ships](#)

Fishing Boats

- [Fishing Ships](#)

Fishing Ships

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Synonyms

Fisher; Fishing boats; Fishing vessels

Introduction

Fishing ships usually refer to production vessels that directly catch fish and other fishery resources. With the increase of voyage, the expansion of production scale, and the refinement of labor division, the traditional fishing ships, which catch fish and keep fresh on board and transport them directly back to the base port, have gradually evolved into several specialized ships engaged in fishing, processing, and transporting catches. In order to improve the efficiency of fishery production, strengthen fishery production management, make reasonable use of fishery resources, and train fishery production personnel, fishery survey ships, fishery administration ships, and fishery practice ships have appeared (Fu 1996).

Scientific Fundamentals

History

Fishing has been going on since human beings existed. It is believed that fishing ships were used as fishing tools as early as the Neolithic Age. The canoeing boat for fishing at that time was the prototype of fishing ship. With the development of productivity, the equipment of fishing ships is constantly updated, and the improvement of fishing ships also promotes the development of fishing technology.

In ancient times, the navigation and operation area of fishing ships was considerable, while the large-scale development of fishing ships in the world came after the application of steam engine

on ships. The motorization of fishing ships began with the side trawler. It was first established in Britain in 1882 and later developed in Europe, America, Asia, and other regions. Although the internal combustion engine was applied to fishing ships at the beginning of the twentieth century, the large-scale use of diesel fishing ships, replacing the steam fishing ships, was in the 1950s.

There has been an increase in fishing operations far away from the base port with the emergence of the motorization of fishing ships. Because of this, although the size of the fishing ship has increased, average accidents have also increased in view of the operation in the wind and waves. So the Germanischer Lloyd began to study the stability of fishing ships in the early twentieth century.

The processing base ship for frozen fish catches emerged in the 1930s, for the convenience of long-term offshore operations. This method of base ship fishing was first adopted by the French in the sixteenth century in the Newfoundland fishery, with the aim of extending the working time at sea. The base ship supplied the fishing ship and received the catch for salting.

If the steam engine used for fishing ships is one turning point in the fishing ship history, then another one is the emergence of the stern ramp fishing ship. In 1954, Britain built the world's first new type of stern ramp trawler (Hjue 1970), and the former Soviet Union then began the large-scale construction of it.

The development of the world's fishing ships has been affected to a certain extent with the introduction of the United Nations Convention on the Law of the Sea and the division of the 200 nautical mile exclusive economic zone. One major aspect is that the scale of fishing ships tends to go both ways. On the one hand, in order to strengthen the development and utilization of domestic offshore resources, many countries have vigorously developed small fishing ships suitable for offshore operations. On the other hand, a number of countries with developed marine fisheries have developed large- and middle-sized fishing ships, especially large ones, in order to operate in deep-sea fisheries, because they can operate in the fishery far from the base

port for a long time, adapt to the complex meteorological conditions, and be equipped with more processing equipment.

Longline fishing ships have been gradually developed with the prohibition of drifting on the high seas, because longline fishing ships are good at choosing fishing targets and protecting resources. Besides, they are effective for low resource density and are much better at improving catch quality and protecting the environment than trawlers.

Characteristics

1. Fishing ships, in order to operate at sea, must have good navigational performance, which is ensured by the hull structure and general layout adapted to the operation mode. At the same time, they must also have good fishing performance for carrying out fishing operations, increasing the catch amount and obtaining higher economic benefits (Yonghe XIE and Liping ZHAO 2004). This is mainly ensured by the reliable fishing and refrigeration machines and some fishing aid equipment (Asche et al. 2014).

There are many kinds of fishing machines, which are usually complex and with high power. The winch used by the trawler varies in tension and speed with water depth, fishing gear, ship size, and engine power. Fishing machines on the seiner may reach up to more than 20. Some seiners are equipped with three to five fishing launches to coordinate the operation. Longline fishing boats are equipped with coiling machines, payoff machines, bait hanging machines, etc. Squid fishing boat and pole fishing boat are equipped with automatic jigging machines.

Fresh preservation and processing equipment are different on various types of fishing ships. Coastal and inshore fishing trawlers mainly use ices of 0 °C or so for cold preservation (Hernández et al. 2014). Middle- and small-sized seiners mainly adopt cold seawater of −1 °C or so for refreshment (Himelbloom et al. 1994; Erikson et al. 2011). The trawlers or fishery base ship operating far from the fishing port shall adopt plate

freezing, tunnel air-blast freezing, and other low-temperature freezing (Lugasi et al. 2007). Pole fishing ships and tuna longline fishing ships mainly use cool brine, with temperature of −50–55 °C for freezing (Kamalakanth et al. 2011). In addition, catches can be processed into fillet, fish block, can, surimi, fish meal, etc. on the large- and middle-sized ocean fishing ships, which have to be equipped with a variety of fish processing machinery, such as fish washing machine, filleting machine, cutting machine, peeling machine, meat separator, fish meal machine, etc. (Fet et al. 2010; Bar 2015).

Fishing aid equipment (Asche et al. 2014), navigation, and communication devices are used to detect fish groups or assist fishing operations, such as depth fish-finder, fishing sonar, net sounder, bull trawler rangefinder, trawl warp length tensiometer, radar buoy and its receiver, etc. The modern fish-finder has been equipped with a computer to realize the function of memory, storage, analysis, and so on. Fishing sonar has developed into an electronic scanning type, which can detect fish groups in all directions and display them directly. The combination of fishing aid equipment and navigation devices forms an integrated fishing aid system.

2. Fishing ships consume oil, water, and supplies while fishing at sea, and fishing operations may have a wide and complex impact on fishing ships, which make them become operating ships with variable load and speed. Therefore, sufficient attention should be paid to the stability check.
3. The working condition of the fishing ship is extremely bad, and the operating frequency is high. Hull corrosion and equipment wear are serious, and the maintenance condition is also poor. Therefore, the design margin of the hull structure, machinery, and equipment of the fishing ship is larger.
4. Fishing ships are usually small in size, but in order to increase the catches, a large number of equipment are equipped on board. As a result, fishing ships become intensive complexes with multiple equipment.

Performance

Fishing ships, like other vessels, are engineering structures on water which must have some navigation and operation performance for navigating on water, i.e.:

Navigation Performance

1. **Buoyancy** is the ability to float on the surface of the water under certain loading conditions.
2. **Stability** is the ability to avoid tilting to dangerous angles under external forces and return to the original equilibrium position when the external force is eliminated.
3. **Insubmeribility** is the ability to maintain sufficient buoyancy and stability without sinking or capsizing when one or more cabins are damaged and flooded.
4. **Rapidity** is the ability of a ship to sail at a high speed at a certain engine power.
5. **Maneuverability** is the ability to keep the course and change course rapidly according to the intention of the pilot.
6. **Seakeeping** is the ability of a ship to maintain enough stability and structural strength when generating the swaying, slamming, green water, and stall under the action of wind and waves and other external forces and to maintain a certain speed for safe navigation.

Operation Performance

1. **Deadweight** refers to the deadweight capacity of a ship, which is usually divided into net deadweight and gross deadweight. Net deadweight refers to the quality of goods, passengers, and their luggage carried that are used for profit. Gross deadweight refers to the quality of net deadweight plus the quality of fuel, fresh water, food, fleet and their luggage, nets, and spare parts, i.e., the difference between full-load displacement and light ship displacement.
2. **Tonnage** is an indicator of the volume of the enclosed area of the ship, which is usually used as the basis for paying fees for ship inspection, berthing, water diversion, etc. It is usually divided into gross tonnage and net tonnage. Gross tonnage refers to the total volume within a ship measured in accordance with tonnage measurement specifications. Net tonnage is the remaining effective volume minus the volume

that cannot be used for profit from the gross tonnage.

3. **Speed** is the speed at which a ship sails, measured in kn.
4. **Endurance** refers to the maximum distance a ship can reach at a certain speed without adding fuel, fresh water, food, and spare parts during the voyage, measured in nautical miles.
5. **Habitability** refers to the comfort of living conditions and facilities.
6. **Robustness** means that the hull has sufficient strength and does not consume too much material to increase the weight of the hull in the navigation area.

In addition to the above, the following technical characteristic parameters are available to characterize the capability of a fishing ship to conduct fishery production:

1. **Operating radius** refers to the maximum distance from the base port to the operating fishery when the fishing ship returns to the base port after arriving at the operating fishery for fishing without adding fuel, fresh water, food, and spare parts during the voyage and can guarantee the quality of the catch.
2. **Self-sustaining capacity** refers to the maximum number of days that a fishing ship can navigate and operate at sea and ensure the quality of the catch without adding fuel, fresh water, food, and spare parts during the voyage.
3. **Catch handling capacity** refers to the number of fish that can be processed every day by ships equipped with fishing processing equipment (including freezing, fish meal, fish oil, fish paste, cans, etc.).
4. **Fishing cycle** refers to the number of days when a fishing ship is loaded with fuel, fresh water, food, and spare goods to reach the state of being able to sail again for fishing operations after arriving at the fishery for operation and returning to the base port to unload its catch. Therefore, this cycle includes the number of sailing days from the base port to the operating fishery, the number of working days including the transfer of fisheries and avoiding wind, the number of days required to return from the

fishery to the base port, and the number of days required for unloading catch and loading fuel, fresh water, food, and other spare parts.

Key Applications

According to the different tasks in fishery production, fishery ships can be divided into fishery production ships, productive fishery auxiliary ships, and nonproductive fishery auxiliary ships.

Fishery production ships refer to production ships which are mainly engaged in fishing operations directly using certain fishing gear.

1. **Trawler** is a ship that use trawl to catch fish. The trawl is a filterable fishing gear. Fish and shrimp are collected into the net in the process of trawling. The water can filter through the net, while the catch cannot pass through to achieve the purpose of catching fish. Trawlers can be divided into single boat trawler and pair trawler according to their different modes of operation. There are two types of single boat trawlers: side trawler and stern trawler. Two fishing ships tow a net together during fishing operation of the pair trawler. There is no significant difference between the single boat trawler and pair trawler, except that the trawl should be equipped with the otter board in the single boat trawler while it is not needed in the pair trawler (Fig. 1).
2. **Seiner** is a ship that uses purse seine to catch fish. The purse seine is also a kind of filterable fishing gear which is mainly used to catch pelagic fish. The rapid turning of the ship makes the net expand vertically in the water to form a circular wall around the fish for trapping them during operation. Seiners can be divided into single-boat seiner, two-boat seiner, and multi-boat seiner according to the number of ships used in the operation (Fig. 2).
3. **Angling boat** is a ship that uses fishing tackle to catch fish. Fishing tackle is a good fishing gear for catching scattered fish. It is adapted to operate in multi-reef fishery and catch large fish of good quality, which is conducive to the protection of fishery resources reproduction. Fishing tackle uses the fish's food to lure them to swallow the hook tied to the fishing line which is fitted with a seductive bait. Another way is to lay dense and sharp empty hooks on the fish migration passage for catching. Angling boats can be divided into longline fishing boat, pole fishing boat, and squid fishing boat according to the fishing gears. Longline fishing boats are fishing for large catches. Tuna is the main catch for longline fishing boat in some countries with developed marine fisheries. Pole fishing is an old fishing method. At present, skipjack is the main catch of the pole fishing boat. There is a live bait tank in the pole fishing boat on sail, which is one of the requirements different from other types of fishing



Fishing Ships, Fig. 1 Trawler. (a) Inshore trawler (<http://trawlerphotos.co.uk/gallery/showphoto.php?photo=169399&title=pino-ladra&cat=602>). (b) Ocean trawler (<http://www.sfgc.com.cn/html/yyyy/yycd/gyxtwcd/25717.html>)

ships. Squid fishing boat is a special fishing boat that uses fishing tackle to catch squid. Squid have the habit of phototaxis, which are used by squid fishing boats for light trapping (Fig. 3).

4. **Drifter** is a ship operating with drift net. Drift net is a kind of simple belt net which belongs to passive net. Its fishing principle is to connect tens to hundreds of rectangular meshes into a ribbon shape, and fish are stuck in the mesh or tied to the netting when they try to pass through. In order to protect marine fish resources, drifting on the high seas was totally prohibited on June 30, 1992. Most of the existing drifters are small fishing ships or ships operating combined with trawler or angling boat (Fig. 4).



Fishing Ships, Fig. 2 Seiner (<http://www.sfgc.com.cn/html/yyyy/yycd/jqywwcd/25708.html>)

5. **Hunting boat** is the general name of all fishing ships that use forks, darts, guns, or other tools to hunt large fish and marine mammals, such as sharks, whales, dolphins, seals, walruses, etc. hunting boat is usually small in size (except for whalers) but has a fast speed and flexible operation. It is equipped with a towing device for the catch, which is generally collected by other ships (Fig. 5).

In addition to the abovementioned fishery production ships, there are also catamaran fishing ships, mixed fishing ships, and some other fishery production ships.

Productive fishery auxiliary ships refer to ships directly engaged in auxiliary production operations.

1. **Fishery factory ship** is also known as a factory mother ship. The fishing ship is carried on it to and from the fisheries. The tasks of fishery factory boats include organizing marine fishing production; accepting and processing the catch from fishing ships; storing and transporting the processed fish product; replenishing oil, water, supplies, and other fishery materials for fishing ships and carrying out emergency maintenance, providing medical, cultural, and other services to the fleet; transferring the catch or processed fish product to the transport ship; and accepting oil, water, supplies, and other reserves (Fig. 6).



Fishing Ships, Fig. 3 Angling boat. (a) Tuna longline fishing boat (<http://www.sfgc.com.cn/html/yyyy/yycd/jqyysdcd/56801.html>). (b) Squid fishing boat (<http://www.sfgc.com.cn/html/yyyy/yycd/dxdddcd/25690.html>)



Fishing Ships, Fig. 4 Drifter (<http://nbdjcy.com/content/?833.html>)



Fishing Ships, Fig. 7 Light fishing ship (<http://www.jinquanship.com/productshow.asp?id=36>)



Fishing Ships, Fig. 5 Whaler (<https://jennifermarohasy.com/2006/10/norwegian-whaling-boat-picture-note-from-george/>)



Fishing Ships, Fig. 8 Tuna carrier (<https://3g.china.com/act/news/11038989/20170821/31144632.html>)



Fishing Ships, Fig. 6 Fishery factory ship (http://www.ships.lv/en/vessels/8301175-ocean_fresh.html)

2. **Light fishing ship** is an important auxiliary ship for the light seining. Its main function is to detect fish groups in fisheries and use lights to trap them. Besides, it is usually used to assist fishing ships in operation (Fig. 7).

3. **Catch carrier** is usually equipped with refrigerating equipment. Its job is to bring the catch back from the fishery to the base port. Sometimes it has to transport a certain amount of oil, water, supplies, and fishing supplies to fleets operating in fisheries (Fig. 8).

Nonproductive fishery auxiliary ships refer to ships indirectly engaged in auxiliary operations.

1. **Fishery survey ship** is an important part of the modern fishing fleet. Its mission is to investigate fishery resources, explore new fisheries, conduct experiment of fishing gear and method, and study the method of processing and keeping fresh. There are usually laboratories of resources, biology, chemistry, and processing on board, with a certain number of researchers. It is also equipped with a variety

of fishing equipment and can be engaged in trawling, seining, and angling operations (Fig. 9).

2. **Fishery training ship** is the ship for student practice. It can often carry out a variety of fishing operations, such as trawling, seining, angling, and so on. Therefore, it is equipped with a variety of fishing equipment for students to learn. It requires more cabins to meet the layout requirements of classrooms, training rooms, and living rooms, but the fish tank volume is small (Fig. 10).
3. **Fishery patrol ship** is the ship enforcing fishery laws and regulations to maintain the normal order of fishery production. Its specific tasks include patrolling the fishing grounds and supervising fishing ships to enforce the law on the reproduction of aquatic resources, supervising the implementation of fishery

agreements with the countries concerned, supervising the operation of foreign fishing ships, handling fishery disputes, and so on. It is usually required to have higher speed and better seakeeping qualities, so as to adapt to navigation in bad sea conditions. The ship is equipped with good navigation communication equipment and recording and video machines. In addition to the general sailors, there are fishery inspectors and fishery police on board (Fig. 11).

4. **Fishery rescue ship** is used to rescue fishing ships in distress, so it is required to have fast speed and good seakeeping qualities, and certain first aid equipment is also necessary (Fig. 12).
5. **Fishery supply ship** is used to supply oil, water, and supplies and to carry the rotating sailors of the fishing fleet. It is required to have larger liquid tanks and more living accommodations (Fig. 13).

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Fishing Ships, Fig. 9 Fishery survey ship (http://www.china.org.cn/china/2017-11/02/content_41833427.htm)



Fishing Ships, Fig. 11 Fishery patrol ship (http://m.guancha.cn/Neighbors/2012_10_26_106152)



Fishing Ships, Fig. 10 Fishery training ship (http://www.shuichan.cc/news_view-301343.html)



Fishing Ships, Fig. 12 Fishery rescue ship (http://www.cnss.com.cn/html/2016/yzxc_0817/231345.html)



Fishing Ships, Fig. 13 Fishery supply ship (<http://roll.sohu.com/20120713/n348066132.shtml>)

According to the fishing radius or sea area, fishing ships can be divided into ocean fishing ships, inshore fishing ships, and coastal fishing ships.

1. **Ocean fishing ship** refers to the ship that can fish in unlimited waters, often with a radius of several thousand nautical miles.
2. **Inshore fishing ship** refers to the ship with a radius of about 200n mile.
3. **Coastal fishing ship** refers to the ship operating within a radius of 100n mile.

According to the different fishing gear and methods, fishing ships can be divided into net fishing ships, line fishing ships, hunting ships, and ships using other fishing gear and methods.

1. **Net fishing ship** refers to the fishing ship using nets for fishing operations, such as trawlers, purse seiners, drifters, and set net fish ships.
2. **Angle fishing ship** refers to the fishing ship that uses angle gear for operations, such as longline fishing ships, pole fishing ships, squid angling ships.
3. **Hunting ship** refers to the ship that uses hunting fishing gear for operations, such as whalers.

Besides, there are some other fishing ships with different fishing gear and methods, such as light fishing ships.

According to the material of construction, fishing ships can be divided into wooden fishing ships, steel fishing ships, aluminum fishing

ships, ferro-cement fishing ships, glass fiber-reinforced plastic (GFRP) fishing ships, and polyethylene (plastic) fishing ships.

According to the propulsion methods and power equipment, fishing ships can be divided into fishing motor sailers, diesel-powered fishing ships, diesel-electric propulsion fishing ships, and steam turbine-powered fishing ships.

According to the method of fish catch preservation, fishing ships can be divided into refrigerated fishing ships, cooling seawater fishing ships, partial freezing fishing ships, freezing fishing ships, fish factory ships, and so on.

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Fishing Vessels

► Fishing Ships

Fit Out a Ship

► Ship-Fitting Design

Fitting-Out

► Ship-Fitting Design

Fixed Weight

► AUV/ROV/HOV Hydrostatics

Flexible Connectors

► Connectors of VLFS

Flexible Pipes and Umbilicals

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Flexible Pipes

Definition

Flexible pipes are used to transfer marine oil and gas resources, and the statement of flexible pipes is opposite to traditional steel pipes. Flexible pipes

have good stiffness and other good performance, and they are much more widely used now.

Scientific Fundamentals

Introduction

In traditional concepts, pipes are mainly divided into metal pipes and nonmetal pipes. Metal pipes have high strength and high temperature resistance, but they are vulnerable to corrosion. It has many limitations in marine oil and gas transportation (Chen 2017). Nonmetallic pipes are corrosion-resistant, light in weight, good in hydraulic performance, and easy to install, but have disadvantages, such as poor heat resistance, poor impact resistance, and large coefficient of linear expansion. With the development of modern science and technology, single-material pipes cannot meet the needs of production and social development. People use composite technology to combine pressure-resistant and non-corrosion-resistant metal materials with pressure-resistant and nonmetal materials to make flexible pipes with excellent comprehensive performance.

At present, there are three kinds of composite pipes widely used in ocean engineering: (1) flexible composite pipes; (2) nonmetal reinforced composite pipes; (3) metal reinforced composite pipes.

Flexible Composite Pipe

Flexible composite pipe is a kind of composite material pipe composed of plastic and metal. Compared with metal pipes, flexible composite pipe has the advantages of high strength, high pressure resistance, smooth surface, small friction coefficient, low heat transfer coefficient, good flexibility, long service time, good insulation performance, easy operation, convenient transportation, and good low temperature impact resistance. Therefore, the flexible pipe can allow the floating body to move in a larger range, and the pipe will not withstand large top tension. Flexible pipes are widely used in many fields, even used to transport drinking water, chemical and pharmaceutical supplies (Antal et al. 2002; Chen 2011). With the continuous development of flexible pipes, there are many kinds of flexible pipes, which can be

divided into bonded flexible composite pipes and unbonded flexible composite pipes according to the transfer mode of interlayer force.

Bonded Flexible Composite Pipes Bonded flexible composite pipe is a kind of multilayer composite structure. The layers of the pipe body are closely combined without relative slip. Generally, it consists of inner layer, middle layer, outer layer, and reinforcement layer, as shown in Fig. 1. After extrusion, it should be vulcanized to make the layers bond together so that they have a higher bonding strength (Wei et al. 2003). Bonded flexible pipes are mostly used in offshore floating or underwater oil pipes, mostly in connection with marginal oilfields and floating production systems, or in oil transportation between wharf and tanker. The corresponding standard specifications of bonded flexible composite pipes are API SPEC 17 K and ISO 13628-10. They have the following characteristics in production and use (Zhang 2011):

1. The corrosion resistance of the bonded pipe is strong because the main bearing body of the structure is embedded in the rubber matrix.
2. Under the same size and compressive capacity, the bonded pipe is lighter.
3. The production cost is low, and it has price advantage in short hose assembly.
4. Leakage is a slow leak in the pipe, which is safer than unbonded pipe.

Unbonded Flexible Composite Pipes There is no bonding effect between layers of unbonded flexible composite pipes, which allows relative

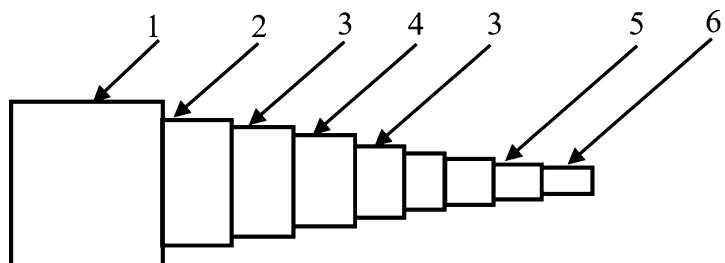
slip between layers. At the same time, the spiral bands of each reinforcing layer are also nonbonding (Zheng 2010). Usually, the reinforcing layer of the pipe is spirally wound with nonmetallic or metal materials (Chen 2005), or consists of a self-locking structure. Compared with the bonded pipe, the unbonded pipe has better performance. Its bending radius is small and curvature is large and it can produce large transverse deformation. At the same time, it can also ensure the axial tensile strength, and the ability to withstand the internal and external circumferential pressure. It can also produce in large length in production, manufacture, and application. In shallow water, unbonded pipes absorb additional stresses caused by temperature difference or vibration at both ends of the pipe through their bending deformation to protect the structural safety of the whole pipe (Feret and Boumazei 1987). Relevant design specifications for unbonded flexible composite pipes are as follows: API SPEC 17B-“Recommended Practice for Flexible Pipe” and API SPEC 17 J-“Specification for Unbonded Flexible Pipe”.

Unbonded pipe is composed of multilayer structure, including insulating layer, anti-wear layer, inner protective layer, outer protective layer, and functional reinforcement layer which plays a major role in load-bearing. According to the different materials of reinforcement layer, it can also be divided into two categories: “Metal unbonded flexible pipe” (using metal materials as reinforcement, mostly carbon steel) and “Nonmetal and unbonded flexible pipe” (using fiber reinforced resin matrix composites as reinforcement).

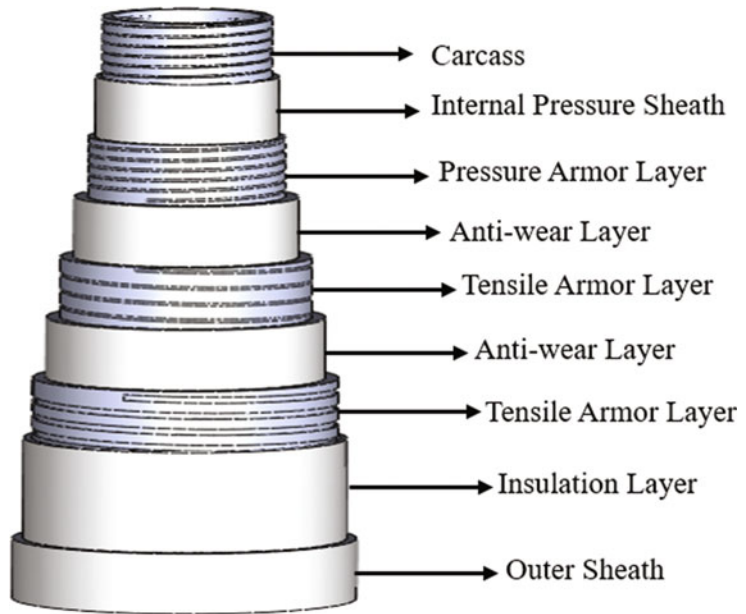
Metal Unbonded Flexible Pipe Figure 2 shows the typical structure of flexible metal pipes, which are carcass, internal pressure sheath, pressure

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Fig. 1 Bonded flexible composite pipe: 1 Outer layer, 2 Outer protective layer, 3 Middle layer, 4 Reinforcement layer, 5 Inner protective layer, 6 Inner layer



Flexible Pipes and Umbilicals, Fig. 2 Metal unbonded flexible pipe (Ren 2015)



armor layer, anti-wear layer, tensile armor layer, insulation layer, outer sheath from inside to outside. They have friction and allow relative slip (Wellstream 2010).

Carcass: A self-locking metal structure with S-shaped cross-section and gaps between layers is used in the innermost layer of pipe. When the pipe is subjected to external pressure or impact load, it can resist the collapse of the lining or the whole pipe and prevent the collapse of the structure. Sometimes it can be placed on the outer layer to protect the outer layer of the pipe. Carbon steel, stainless steel, nickel-based alloy, and other materials are usually used.

Internal pressure sheath: The material is a polymer, which prevents the fluid from leaking into the external structure and guarantees the integrity of the internal fluid. This layer has obvious creep effect.

Pressure armor layer: Self-locking structure or rectangular cross-section spiral winding structure can be used. The winding angle is close to 90° degrees. This layer can enhance the internal and external pressure bearing capacity of pipe and resist mechanical impact load. It has many structural forms (American Petroleum Institute 2012a). Carbon steel is selected as the material and carbon content is determined according to the design

requirements. Sometimes aluminum alloy can be selected (Out and Morgen 1997).

Anti-wear layer: Polymer material located between two metal structural layers which is used to reduce the friction damage between adjacent reinforcement layers and improve the fatigue life of pipes. Generally, the stiffness is not considered. Common materials are high density polyethylene (HDPE), cross-linked polyethylene (XLPE), medium density polyethylene (MDPE), nylon 11 (PA-11), polyvinylidene fluoride (PVDF).

Tensile armor layer: It is made of a pair of flat steel spirals with opposite winding angles. The winding angles are usually between 20 and 55 and mainly bear the effects of tension, torque, and internal and external pressure. The material of this layer is the same as the pressure armor layer. The type of carbon steel can be selected reasonably according to the design requirements. Sometimes, aluminum alloy can also be selected (American Petroleum Institute 2012b).

Insulation layer: This layer is located between the tensile layer and the outer layer. It is used to improve the thermal insulation and thermal insulation performance of pipes. PP/PVC/PU insulation material is used.

Outer sheath: Polymer material is used as the outermost structure of the pipe, which isolates the

internal structure from the external environment, protects the pipe from corrosion and wear by sea water, and keeps the flat strip of the tensile layer spirally wound and formed without dislocation. The materials used are usually high density polyethylene (HDPE) and nylon 11 (PA-11).

Nonmetal Unbonded Flexible Pipe Nonmetallic unbonded flexible pipes are reinforced with fiber-reinforced composite. Compared with metal flexible pipes, this new type of pipe does not have carcass, and both tensile and compressive layers are composite, so the whole pipe is nonmetallic materials.

Figure 3 is a typical structure of a flexible nonmetallic composite pipe. From the inside to the outside, they are: Liner Extrusion, Anti-Extrusion Layer, Hoop Reinforcement, Anti-wear Layer, Membrane Extrusion, Tensile Reinforcement, Anti-wear Layer, Outer Sheath.

Liner Extrusion: The polymer extruded by the extruder can prevent the fluid leakage inside the pipe and ensure the integrity of the internal fluid. It has good wear resistance and smooth inner wall, which is more conducive to the transportation of crude oil. The performance of this layer is mainly affected by temperature and internal flow material. HDPE, XLPE, PA-11, PVDF, PE-RT are often used.

Hoop Reinforcement: It is made of several flat strips spirally wound, and the winding angle is

close to 90° . The flat strip is a composite laminate with rectangular cross section. Friction between the flat strip and the flat strip is allowed and relative slip is allowed. Glass fiber-reinforced polymer, carbon fiber-reinforced polymer, and aramid fiber-reinforced polymer are commonly used in flat strip (American Society for Testing and Materials 2007).

Anti-wear Layer: The main function is to reduce the wear between layers, prevent the leakage of internal gas, insulate, and keep warm.

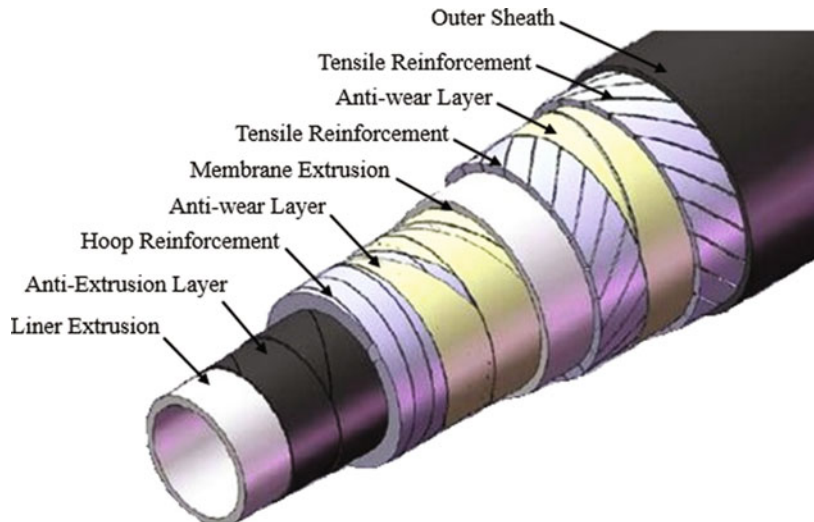
Tensile Reinforcement: The two layers appear in pairs, and the helical angles of the two layers are positive and negative. The material of hoop reinforcement layer is also spirally wound by composite flat strip. The winding angle is generally between 25 and 55 with rectangular section shape. Friction and relative slip are allowed. If necessary, more pairs of tension reinforcement layers can be added to improve the mechanical properties of pipes.

The advantages of nonmetallic flexible pipes are as follows:

1. Light weight and high specific strength, it is 25–50% lighter than metal reinforced flexible pipe, which is more suitable for deep and ultra-deep water environment (DNV 2010).
2. Nonmetallic composite laminates have better fatigue resistance and structural integrity.

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Fig. 3 Nonmetallic composite pipe (Zhang, 2014)



3. The installation process is more convenient, the installation and deployment cost is lower and more economical.
4. Good resistance to high temperature and corrosion and the fatigue life is longer.
5. Good wear resistance, the inner wall is smooth which improves the fluidity of crude oil transportation.
6. The whole pipe structure is made of nonmetallic materials with low thermal conductivity and good thermal insulation performance.

Nonmetallic Reinforced Composite Pipe

Nonmetallic reinforced composite pipe has the advantages of light weight, high specific strength, low hydraulic friction, long service life, and convenient installation. The reinforcing materials used in nonmetallic reinforced composite pipes are mainly glass fiber, carbon fiber, aramid fiber, ultra-high molecular weight polyethylene fiber, and so on (Lu 2005). Compared with glass fibers, carbon fibers and other fibers are expensive and costly to process. In fact, up to 98% of the nonmetallic matrix reinforced composite pipes use glass fibers as reinforcing materials, only 2% use carbon fibers, aramid fibers, ultra-high molecular weight polyethylene fibers, and other fibers (Wang and Jiang. 2003). Therefore, the glass fiber-reinforced composite pipes are only introduced as follows.

Fiberglass-reinforced plastic pipe (GFRP or FRP pipe) is a kind of flexible composite pipe (resin, fiber, sand, etc.) which has been widely used at home and abroad. Glass fiber-reinforced continuous plastic composite pipe is a kind of multilayer structure and high performance

composite pipe. Its inner layer is a medium conveying layer made of special polymer. It not only has the characteristics of corrosion resistance and wear resistance but also has a hydrophilic material on the surface. When transporting waxy crude oil, it can effectively prevent wax condensation in the pipe. The middle layer is reinforced by high performance resin glass fiber composite material. The layer winds at different angles on the outer side of the inner layer, which makes the pipes have higher pressure bearing capacity. The whole pipe is wrapped by polymer materials, which can effectively prevent the damp, thermal, light, and mechanical damage of the external environment, as shown in Fig. 4 (Xia et al. 2013).

Compared with traditional pipes, FRP composite pipes have the following advantages:

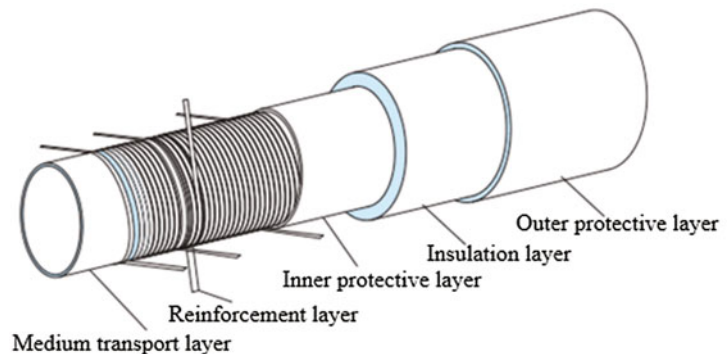
1. Good corrosion resistance and dielectric properties
2. Strong elasticity and toughness, creep resistance and surface scratch resistance
3. Easy to design
4. Good thermal performance, low coefficient of expansion, strong thermal insulation and heat preservation capability

Metal-Reinforced Composite Pipe

According to manufacturing process, metal-reinforced composite pipes can be divided into steel wire wound reinforced plastic composite pipes (PSP pipes), steel wire spot welded into mesh reinforced plastic composite pipes (SPE pipes), and porous steel strip plastic composite pipes (PSSCP pipes).

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Fig. 4 Fiberglass-reinforced plastic pipe



Steel Wire Wound Reinforced Plastic Composite Pipe Steel wire wound reinforced plastic composite pipe (PSP) is a steel-plastic composite structure, which is based on thermoplastic-high density polyethylene (HDPE) and reinforced by reticulated skeleton formed by staggered winding of high strength steel wire. The steel wire and HDPE are bonded by high performance resin, as shown in Fig. 5 (Zhao 2013).

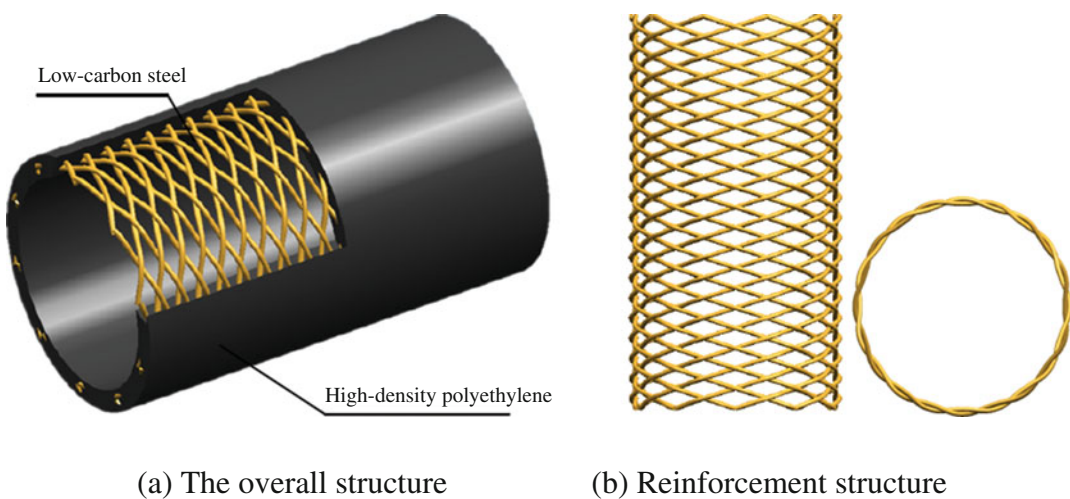
The unique structure of PSP not only integrates the advantages of high strength of steel pipe and corrosion resistance of plastic pipe but also overcomes the defects of metal plate skeleton reinforced plastic composite pipe such as easy delamination. PSP has the advantages of high production efficiency, strong structural design, strong bearing capacity, good corrosion resistance, good wear resistance, high cost performance, light weight, and convenient transportation and installation. It has been widely used in municipal engineering, civil construction, pharmaceutical and chemical industry, agriculture, and coal chemical industry, and has produced remarkable social benefits and huge economic benefits (Zheng et al. 2012).

Steel Wire Spot Welded into Mesh Reinforced Plastic Composite Pipe Steel wire spot welded into mesh reinforced plastic composite pipes (SPE

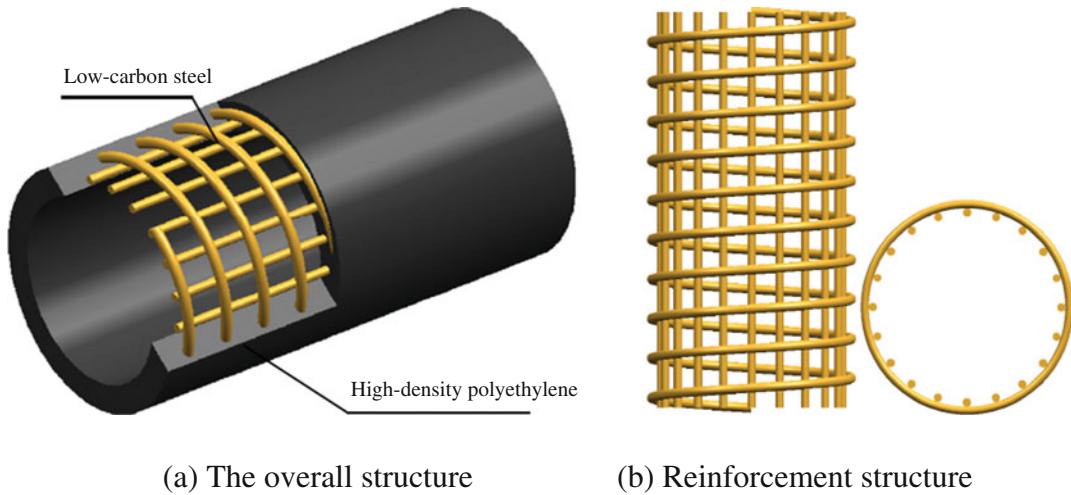
pipe) is a new type of double-sided anticorrosive pressure pipe formed by continuous film drawing on production line, which is based on high-quality low-carbon steel wire as reinforcing phase and high-density polyethylene (HDPE) as matrix. Spot welded steel wire is synchronized with plastic extrusion filling, as shown in Fig. 6 (Zhao 2013).

The tubular wire mesh in SPE pipe is made of warp wire and weft wire by winding and welding. The warp wire is parallel to the center line of SPE pipe, while the weft wire is wound around the outer layer of the warp wire. The diameters of warp and weft wires are 2–4 mm low carbon steel wires. In the production of SPE pipes, steel wires are laid along the axis of the pipe with an interval of 8 mm and wound at an interval of 8–12 mm along the outer ring of the steel wires. Then the steel wires are spot welded into a mesh structure, filled with high density polyethylene on both sides of the mesh structure, and wrapped in the middle of the mesh structure to form a stable steel skeleton polyethylene structure.

Due to the complex winding process, slow production speed and thicker pipe wall structure of wire spot welding reinforced composite pipe, the proportion of composite pipe is large, the cost is high, and the transportation and installation are



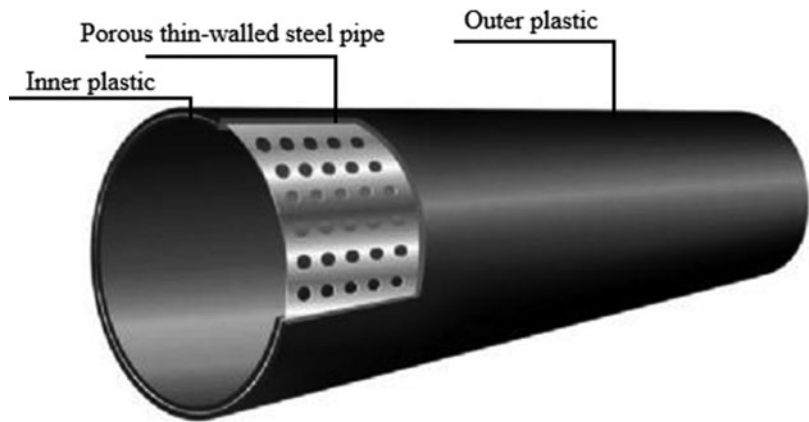
Flexible Pipes and Umbilicals, Fig. 5 Steel wire wound reinforced plastic composite pipe: (a) The overall structure (b) Reinforcement structure



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Flexible Pipes and Umbilicals, Fig. 6 Steel wire spot welded into mesh reinforced plastic composite pipe: (a) The overall structure (b) Reinforcement structure

Flexible Pipes and Umbilicals, Fig. 7 Porous steel strip plastic composite pipe



very inconvenient. Therefore, its application field is greatly limited (Wang et al. 2017).

Porous Steel Strip Plastic Composite Pipe Porous steel strip plastic composite pipes (PSSCP) are made of cold-rolled steel strip and thermoplastic high-density polyethylene, and the porous thin-walled pipe formed by hydrogen arc butt welding is used as a reinforcement. It is a new composite pressure pipe with a double-sided composite thermoplastic in the outer and inner layers, as shown in Fig. 7.

It is made of cold-rolled steel strip after punching, coiled, and welded into mesh steel pipe.

The inner and outer layers are made of thermoplastic resin and continuously compounded by extrusion. By hot melting, the inner and outer plastic layers can be contained through thin steel strip mesh, which can avoid the separation and peeling of plastics and metals, solve the problem of insufficient strength and stiffness of pure plastic pipes, and control the linear expansion coefficient of plastics. It integrates the advantages of steel pipe and HDPE. The pipe is smooth and hard. It can be used in the environment of -40 – 120 °C. It can resist corrosion, pressure, and abrasion on both sides. It has been widely used in petroleum, chemical, metallurgy, construction, and environmental protection (Tian et al. 2008).

Umbilicals

Definition

Umbilical is a key component of subsea control system, which connects the upper facilities and subsea production system (Fig. 8), and transmits hydraulic pressure, electric, control signals, and chemicals between them.

Introduction

Subsea umbilical is installed between the host facility and the subsea facility. In general, the umbilical includes a catenary riser (dynamic segment) transitioning into a static segment along the seabed to the umbilical termination assembly (UTA) at the subsea facility. For shorter lengths, the segments may be identical. The umbilical pull-head will include a split flange (or other) assembly for hang-off of the umbilical at the host facility. A bend stiffener or limiter may be installed at the top of the catenary riser and at the UTA (Yong Bai and Qing Bai 2012).

General requirements for the umbilical system include:

- Electrical power, control, and data signals should, as a base case, be contained on the same pair of conductors.
- Super duplex steel tubes should be used. (Other materials can be considered but substantial documentation should be required to guarantee their applicability.)

- The umbilical is fabricated in one continuous length.
- The umbilical system is designed without any planned change-out over the design life.

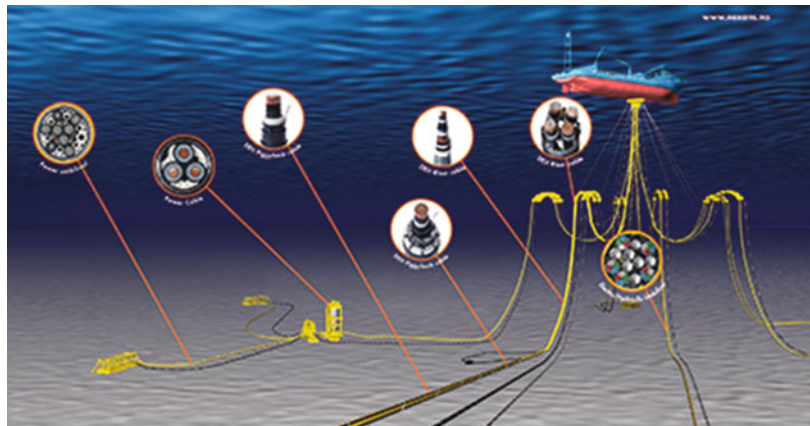
Development History of Umbilical

Umbilical has been used in oceanographic engineering for nearly 60 years. The earliest umbilical appeared in the 1960s. It was used in a 16 m-deep underwater well built by Shell in the Gulf of Mexico in 1961. The umbilical is a direct hydraulic type, which is composed of many hydraulic pipes, each of which is connected with the corresponding tree valve, and the whole cable has a large cross-sectional area (Yijimei 2014).

In the 1970s, due to the wide application of hydraulic directional valve and integrated circuit technology, electrohydraulic composite control system appeared, which greatly reduced the cross-sectional area of umbilical. By this time, umbilical was no longer only composed of pipes, but also power cables and data transmission lines.

The early umbilical pipes used thermoplastic hosepipes (Fig. 9). It was not until 1982 that Sandvik company first replaced thermoplastic hosepipes with steel pipes and successfully applied them in the Frigg oil and gas field development in northeast Norway. This umbilical is 57 km long and is used for remote hydraulic control and methanol injection. Unfortunately, in the next few years, the development of steel umbilical has not got enough attention.

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Fig. 8 Umbilical



However, in the late 1980s, as the depth of water increased, some problems of the hosepipes gradually emerged, so it is getting more and more



Flexible Pipes and Umbilicals, Fig. 9 Hose umbilical

limited. Leakage occurred at the connection and hydraulic fittings, methanol leakage occurred in the hosepipes, the hosepipes was crushed in the deep-water environment, and the cable and connector take in water problem troubled the whole umbilical industry. As a result, since the 1990s, steel pipe and fiber optic umbilical have received increasing attention and have been increasingly used in offshore oil development.

The main materials for steel pipe are carbon steel, duplex stainless steel, super duplex stainless steel, etc. (Fig. 10). However, the price of umbilical with metal tube is high, and economic factors should be taken into account when choosing it.

In the twenty-first century, with the increase of the depth of water, it is necessary to supply high-voltage power to the deep-water subsea production system. Therefore, the requirements for the voltage level of the umbilical have been raised. In 2005, Nexans began developing and producing the world's first umbilical with a high voltage (24 kV) cable for BP's two subsea recovery Wells in the Gulf of Mexico at a depth of 988 m; it is 26 km long. Since then, high-voltage umbilical has been used in offshore deep-water oilfield development.

Flexible Pipes and Umbilicals, Fig. 10 Steel pipe umbilical



At present, umbilical laying depth is increasing, such as the TIGER oil field in the Gulf of Mexico began to install the laying of steel pipe umbilical in 2009, the water depth reached 2743 m.

The laying length of umbilical is also increasing, ranging from 500 m to 130 km. In 2005, Nexans produced steel pipe umbilical for the offshore gas field of Snohvit, the state oil company of Norway, reaching 143 km. The single longest thermoplastic umbilical is 31 km, produced by the Newcastle Facility and used in the Exxon Mobil Arthur project (Guo et al. 2016; Sun et al. 2011).

Types of Umbilical

According to the actual needs of oil and gas field engineering, the reserved facilities, external pipe diameter conditions, and user requirements, there are great differences in the types of units and the number of units in umbilical. Therefore, umbilical products are typical individual customized products.

According to the types of internal units, umbilical can be divided into cable umbilical, hydraulic umbilical, optical fiber umbilical, electro-optical composite umbilical, hydraulic fiber composite umbilical, electro-optical composite umbilical,

electro-optical composite umbilical, electro-optical liquid composite umbilical, and integrated application umbilical (in addition to the photoelectric liquid unit, it also contains oil and gas channels) (Collins 2008).

The umbilical including pipe units can also be divided into steel pipe umbilical and hosepipe umbilical according to the types of pipe units.

Moreover, umbilical can be classified into unarmored umbilical and armored umbilical according to whether or not they contain armored layers. According to different armored materials, it can be divided into metal armored umbilical and nonmetal armored umbilical. The armored umbilical can be divided into single, double, and multi-layer armored umbilical according to the number of armored layers.

According to different working conditions of umbilical, it can be divided into static umbilical and dynamic umbilical. The following is a brief introduction of common umbilical in the world (Guo et al. 2016).

Single-Layer Armored Optoelectronic-Hydraulic Composite Steel Tube Umbilical

Figure 11 shows a typical cross section of single-layer armored optoelectronic-hydraulic

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Fig. 11 Typical cross section of single-layer armored optoelectronic-hydraulic composite steel tube umbilical



composite steel tube umbilical. The umbilical can deliver power, control, electrical and optical signals, hydraulic power, and chemical agents simultaneously.

Two-Layer Armored Optoelectronic-Hydraulic Composite Steel Tube Umbilical

Figure 12 shows a typical cross section of two-layer armored optoelectronic-hydraulic composite steel tube umbilical. Its function is the same as that of single-layer armored optoelectronic-hydraulic composite steel tube umbilical; the difference is that the umbilical is armored with two layers of flat steel wire, only one layer of outer sheath and inner lining. This type of umbilical is also suitable for subsea production systems with electrohydraulic composite control.

Electrohydraulic Composite Hose Umbilical

Electrohydraulic composite hose umbilical is suitable for subsea production systems with electrohydraulic composite control (Fig. 13). Because of the permeability and self-expansion of the hose-pipe, it is suitable for the working condition with short control distance and low demand of hydraulic response.

It uses five layers of rounding steel wire for armoring. Armoring can bear the tensile load, increase the stability of the seabed, and protect the internal structure of the cable body.

Electrohydraulic Composite Soft Steel Pipe Umbilical

Similar to the electrohydraulic composite hose-pipe umbilical, it is suitable for the subsea production system with electrohydraulic composite control mode. The difference is that both hosepipe and steel pipe are used in the pipe unit (Fig. 14).

Carbon Fiber Umbilical

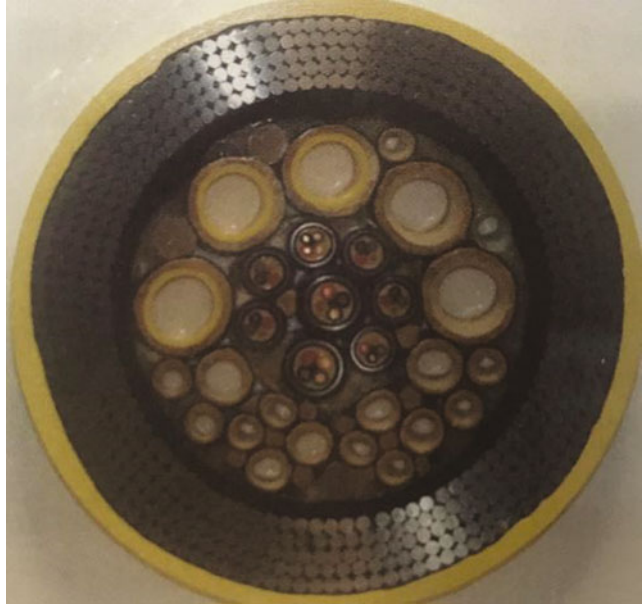
To reduce the weight of the umbilical cable, carbon fiber rods can be added. It has the same stiffness as steel and has a weight loss of 80%. This rod is made of 48 K carbon fiber and vinyl ester resin. As shown in Fig. 15, the carbon fiber pultrusion rods are arranged in the cavity of the outer circumference and the plastic profile inside the umbilical structure. Stainless steel tubes are distributed in the extruded plastic profiles, and the carbon fiber pultrusion rods are placed in the cable production at the same helix angle like other parts (Yu and Su 2008).

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Fig. 12 Typical cross section of two-layer armored optoelectronic-hydraulic composite steel tube umbilical



Flexible Pipes and Umbilicals, Fig. 13 Cross section of electrohydraulic composite hose umbilical



Flexible Pipes and Umbilicals, Fig. 14 Cross section of electrohydraulic composite soft steel pipe umbilical



Umbilical Components and Unit

Umbilical vary widely in structure, it has different structures for different environments, and there is almost no uniform standard and style of it. Generally speaking, it is mainly composed of cables (power

cable, signal cable, or power signal cable), optical cables (single-mode or multi-mode), hydraulic or chemical pipe (steel pipe or hosepipe), polymer sheath, carbon fiber rod or armored steel wire and filler (VSolovev et al. 2017).

Flexible Pipes and Umbilicals,**Fig. 15** Carbon fiber umbilical

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A subsea umbilical consists of electrical cables, fiber optic cables, steel tubes, and thermoplastic hoses. It may also include two or three of these four components for executing specific functions. The umbilical components are designed and manufactured to meet the umbilical functional and technical requirements. Proper materials are chosen to manufacture the components, and verification and acceptance tests are done to demonstrate the conformance to the component functional and technical requirements (ISO 13628-5 2009).

Cable is mainly used for the transmission of power and control electricity, and can also be used for signal transmission when electric communication is adopted. Fiber optic cable is mainly used for transmitting various signals, and in special circumstances, it can also be used for monitoring the status of umbilical itself. Various pipes (including steel pipes and hosepipes) are used to transmit hydraulic fluids, chemicals, and oil gas. Polymer sheath is generally wrapped in cables and various types of pipes outside, which play a role in insulation and protection. Carbon fiber rod or armored steel wire is generally arranged around

the umbilical, which can increase the axial stiffness and strength of the cable body, as well as the mass per unit length of the cable body, which enhances the submarine stability of the umbilical. Fillers are used to fill voids and to fix the relative positions of each component of the cable body (Guo et al. 2016).

Electrical Cable

Electrical cable consists of conductor, insulation, sheath, etc. Some cable units are also shielded and armored. Cables are divided into power cables and signal cables. The insulating layer provides dielectric strength and electrical insulation. Electrical cable is equipped with jacket, which can protect the cable unit, round the cable, and adjust the size and weight. The shielding layer is mainly used to shield the magnetic field. Armor mainly plays a strengthening role.

Fiber Optic Cable

Fiber optic cables are capable of continuous operation when immersed in a seawater environment. The fiber type is of either single-mode or multi-mode design. The design is as given in the

manufacturer's/supplier's specifications. Individual fiber identification is by means of fiber coloring. The fibers are contained within a package that prevents water and minimizes hydrogen contact with each fiber. The carrier package for mechanical protection and its contents are designed to block water ingress in the event that the fiber optic cable in the umbilical is severed.

Pipe Unit

Pipe material is steel pipe or plastic hosepipe. The umbilical used in the deep sea are mostly made of steel tubes, which have better resistance to internal and external pressure and permeability than plastic hoses. Most steel pipes are lined with plastic, which plays a role of cushioning and reducing wear between steel pipes and other components.

Filler

The main functions of fillers are as follows: It makes the umbilical rounded and easy to be processed in the following process, and it is also conducive to strengthening external components to play the role in improving the ability of the umbilical to resist lateral pressure and reducing the friction between functional components and layers. Improve the stability of umbilical structure. When water resistance is required, fill the water resistance material and wrap it around the water resistance belt.

Armor Layer

According to the purpose, environment, and state of use of umbilical, different armored forms can be selected, such as single layer round wire, double or four layered round wire, and double or four layered flat wire. The steel wire mainly plays a strengthening role, providing tensile, torsional, and compressive protection for the umbilical, and at the same time, the steel wire also plays a shielding role, mainly shielding the magnetic field. The armoring method of single layer round steel wire is rarely used, and it is mainly used in places with low tensile requirements such as shallow sea or rivers. The advantages of this armoring method are light weight and softness, and the

disadvantages are poor tensile strength and lateral compression resistance, and the cable is easy to twist when being pulled. In the double round steel wire, the winding direction of the double round steel wire is opposite, so the cable is not easy to twist. Compared with double round steel wire, flat wire has the advantages of reducing the external diameter of the cable and increasing the lateral compression resistance of umbilical cord cable. The disadvantage of the cable is hard, bending performance is poor. The combination of flat steel wire and round steel wire has the advantages of excellent tensile strength and lateral compression resistance, but it has a heavy weight and is suitable for the environment susceptible to high mechanical external forces.

Sheath

The sheath is divided into inner sheath and outer sheath. The function of the inner sheath is to provide mechanical buffering, reduce friction, prevent corrosion, and stabilize the cable structure. The outer sheath is the last line of defense for the umbilical. In addition to playing the role of inner sheath, it also has high anticorrosion performance, ultraviolet resistance, and seawater resistance. Some umbilical cord cables require an outer sheath that is resistant to mold and marine life (Gao et al. 2011; Guo et al. 2011).

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Flexural Strength of Sea Ice

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Synonyms

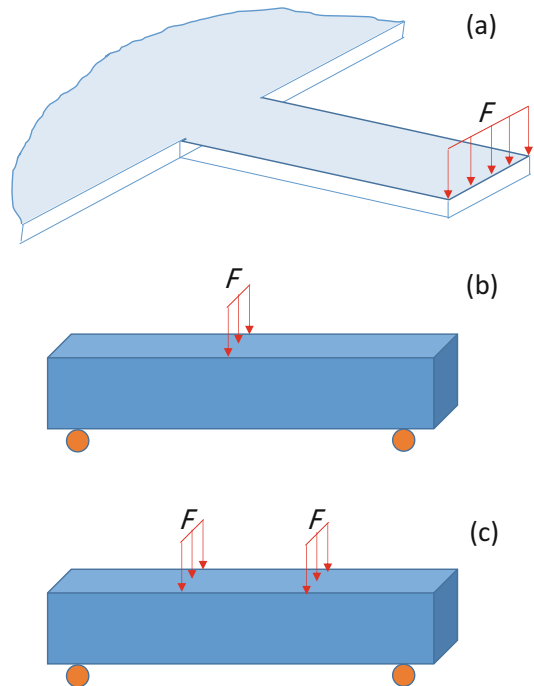
Bending strength of sea ice

Definition

Flexural strength of sea ice is the maximum bending stress that the sea-ice material can take. This definition is drawn from the traditional material mechanics. Once the sea-ice strength is surpassed, the failure and fracture of sea-ice plate are followed. The flexural strength is closely connected with many natural phenomenon and engineering properties of sea ice. For example, the breakup, rafting, and ridge-building are often observed in polar area and seasonal ice-covered region, when the ice covers are under the actions of wave, current, and wind. In the cold region, the jacket platform with ice-breaking

cones, artificial island, slope protection, and ice breaker suffer from the collision of the ice cover. These situations generally cause the bending failure mode of sea ice, and the ice loads of the structures are subsequently determined by the flexural strength.

In essence, the flexural strength of sea ice is induced by the momentum parallel to the ice cover. The momentum generally causes the longitudinal stresses, compressive, and tensile, which distribute linearly with perpendicular distance from the neutral axis in a pure and homogeneous elastic material. Note that the fissure appears at the surface with the zone of critical stress to be very thin, and the fissure propagates in a stress gradient. As we know that sea ice is a complex material with multiphase, anisotropy and not pure elasticity, the effective moduli for tension and compression are not equal, when sea ice takes the bending momentum. Therefore, the sea-ice mechanical properties evidently perform inhomogeneous. Unlikely to the classic material mechanics, the flexural strength of sea ice does not equal to the uniaxial tensile strength.



Flexural Strength of Sea Ice, Fig. 1 Three types of flexural strength tests for sea ice: (a) cantilever beam, (b) three-point beam, and (c) four-point beam

Scientific Fundamentals

Test Types

In general, the flexural strength of sea ice is acquired through the tests in laboratory or in situ. According to the loading types, there exist three test patterns: three-point beam test, four-point beam test, and cantilever beam test, as shown in Fig. 1. Cantilever beam test can only be carried out in situ, and the other two are applicable both in situ and in laboratory.

Cantilever beam test is a full-scale measurement of the strength and the load is applied at the free end of sea-ice sample until the sample breaks at the base of the beam. Three-point beam test and four-point beam test are performed based on the cubical samples cut off from the ice cover, and the loading points are on the center point and middle two of the quantile points, respectively. Cantilever beam test is much better than the other two on measuring the real strength condition of sea ice, because the sea-ice state is maximally preserved in situ without significant movement (Timco and

Weeks 2010). Specifically, cantilever beam test facilitates a relatively easy way to perform on a large beam and can keep the natural temperature gradient in the ice sheet. However, cantilever beam test is highly consuming on many aspects such as time and labor, so it is hard to guarantee enough number of the test for the mechanical property studies of sea ice. Besides, the result of the test is influenced by the variation of physical properties with ice depth, which is brought about by the ice growth history. Stress concentration at the corners of the beam root is another concern for the test. The root of the cantilever may be not rigid enough, especially when the beam is much wider with reference to its depth. The stress concentration sometimes takes place at the corners of the beam root, which could introduce extra error for the measurement of the flexural strength and even fail the measurement.

Otherwise, the three- and four-point beam test is most widely used because of its controllability and feasibility, and its failure position is anticipated much easier than the four-point beam test.

More often, the three-point beam test is preferred to be performed in a laboratory on smaller samples of ice cut from the ice cover. Therefore, most of the studies on the relationship between the sea-ice flexural strength and its influencing factors are based on the three-point beam test. However, the broken location of the three-point beam test is often at the midpoint of the beam, where takes the maximum moment. As a result, the failing location may not be the weakest of the sea-ice sample.

Whereas, four-point bending tests preserve a section of the beam, where the moment is constant and the shear stress is zero. This distribution is able to guarantee the failure happens at the weakest point of the sea-ice sample. Therefore, the four-point test is highly recommended for sea ice, although it is a little more complex than the three-point test.

Stress concentration is another concern to carry out three- and four-point beam test. During the test, the local indentation effects sometimes happen at the loading and supporting points. This has a negative effect on the accuracy of the flexural strength. To avoid stress concentration or local indentations at these points, the test apparatus is suggested to have cylinder supports, which should have enough stiffness comparing with sea-ice sample (ITTC 2014).

The flexural strength of sea ice is not considered as a basic material property. The distribution of the stress field is not uniform inside the sample for the test of flexural strength. The flexural strength of sea ice is more complex than the classic mechanics, and there needs to be extra assumptions to explain the test results. However, because it is a necessary parameter for numerical simulation and ice engineering projects, the flexural strength of sea ice is widely measured.

Influencing Factors of Flexural Strength

Many physical properties of sea ice influence the flexural strength such as salinity, porosity, and temperature. As a deduced parameter from salinity and temperature of sea ice, the brine volume is widely adopted to reflect the physical properties

of sea ice, when the relationship between flexural strength and physical properties of sea ice is investigated. Sea ice is also a rate-dependent material, thus many researchers arguably identify stress rate as another parameter to influence the flexural strength of sea ice. In addition, the scale effect of the flexural strength is observable, so the strength deriving from the small scale test cannot be directly applied to the sea-ice process of the large scale.

Sea-ice temperature is a key element to determine the flexural strength. On microscale, the lower sea-ice temperature is, the denser ice crystal conglomerates. As is embedded on the macro-scale, the flexural strength becomes stronger. Many studies suggest that sea-ice flexural strength is approximately a linear function of its temperature. The flexural strength increases with the decrease of the sea-ice temperature. Contrarily, sea-ice salinity sustains a positive relationship with the flexural strength. The higher salinity of sea ice prone to make more porosity inside of the sea ice, the distribution of these defaults inside the sea ice lower the flexural strength. However, the significance of the linear relationship between the flexural strength and the sea-ice salinity is low, so the influence of the salinity on the flexural strength is more complex. Alternatively, the sea-ice brine volume is computed from the sea-ice temperature and salinity, and it thus can combine these two influencing factor into one variable by (Frankenstein and Garner 1967):

$$v_b = S_i \left(\frac{49.185}{|T_i|} + 0.532 \right)$$

where v_b , S_i , and T_i denotes brine volume, salinity, and temperature of sea ice. This equation is validated only when the ice temperature is $-22.9^\circ\text{C} \leq T_i \leq -0.5^\circ\text{C}$. Therefore, the brine volume tend to be selected to probe into the influence of sea-ice physical properties on the flexural strength.

Many flexural strength tests have been carried out to investigate the relationship between the brine volume and the flexural strength. Most of the researchers reported that the flexural strength of sea ice has a negative exponential relationship

to square root of brine volume. The others concluded that the flexural strength of sea ice is a linear function of square root of brine volume. Although the form of the equation is given, the fitting parameters are a little different from different researchers and different experimental sites. Timco and O'Brien (1994) combined 2500 reported measurements of the flexural strength of freshwater ice (1556 beams) and first-year sea ice (939 beams), as shown Fig. 2. They gave the exponential correlation between the flexural strength and the square root of brine volume as:

$$\sigma_f = ae^{-c\sqrt{v_b}}$$

where σ_f and v_b denotes flexural strength and brine volume of sea ice, respectively. a and c are the parameters through the fitting line.

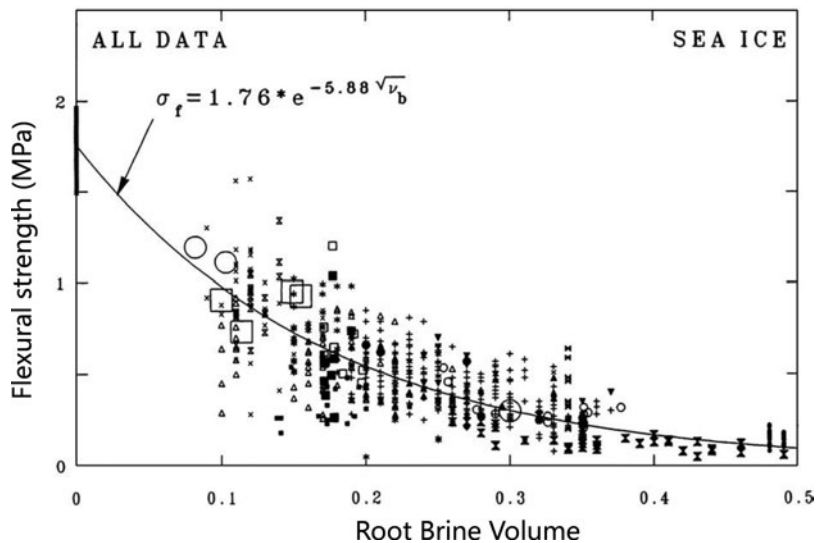
This equation is most representative to calculate sea-ice flexural strength just based on the temperature and salinity of sea ice. Basically, its data comes from a large number of researchers and thus significantly compromises the fitting parameters of a variety of geographic locations, in both polar and seasonal ice-covered region. It is worthwhile to point out that there may be remarkable difference for some certain regions from the fitting parameters (as shown by Fig. 3), but the exponential correlation has been confirmed to be

effective to fit the flexural strength and brine volume of sea ice (Ji et al. 2011).

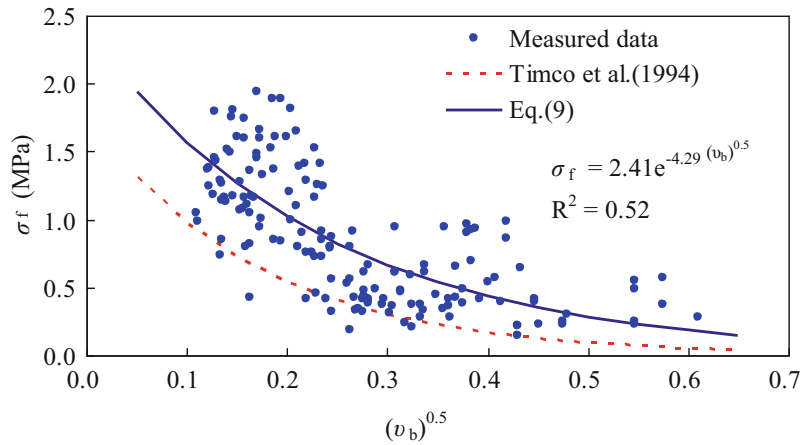
The dependence of the flexural strength on strain (stress) rate is seldom studied and remained as a debatable question (Timco and Weeks 2010; Timco and O'Brien 1994). The dependence on stress rate is confused, and it is not easy to calculate how much degree the observed trends are dominated by the sea-ice physical changes or by the imperfections of test technique. Until now, the agreement with this question has been hardly reached between different researchers. Some researchers insisted that the stress rate has very little effect on flexural strength (Frederking and Timco 1983; Timco and Weeks 2010). The others argued that the flexural strength of sea ice slightly increases with increasing of the stress rate (Ji et al. 2011). One of the main reasons for this inconsistency is inaccuracy of the stress rate. The samples completely crack very soon from the beginning to the end of the loading, approximately within one second (Timco and Weeks 2010). If the sampling frequency is not high enough, the average stress rate is computed less accuracy than that in compression test, where the rate-dependence of compression is clearly observed. However, the flexural strength shows a decreasing trend with the stress rate a little more clearly, when the salinity and temperature of sea ice maximally keep uniform.

Flexural Strength of

Sea Ice, Fig. 2 Flexural strength versus the square root of the brine volume for first-year sea ice (Timco and O'Brien 1994)



Flexural Strength of Sea Ice, Fig. 3 Influence of brine volume on the flexural strength (Ji et al. 2011)



F

This trend enlightens the combination of the brine volume and stress rate to study their coupling effect on the flexural strength of sea ice. Many studies have been focused on the influence of a single factor on the flexural strength of sea ice. In fact, these influential factors are coupled together. Furthermore, the specified factor such as salinity and temperature is hardly controlled to be uniform during the test of sea ice. This causes the inaccuracy of the dependence of the flexural strength on a single factor with the other factors to be constant. Thus, the flexural strength as a multivariate function of temperature, salinity, and stress rate has been proposed. The flexural strength of sea ice can be simulated as a two-parameter exponential equation with the consideration of both brine volume and stress rate (Ji et al. 2011) in Fig. 4. This scheme overcomes the disadvantages of the single factor, but there still need to be more experimental data to determine the fitting equation and its parameters.

$$\sigma_f = ae^{(c+d\dot{\sigma})\sqrt{v_b}}$$

with $a = 2.61$, $c = -5.58$, and $d = 2.09$.

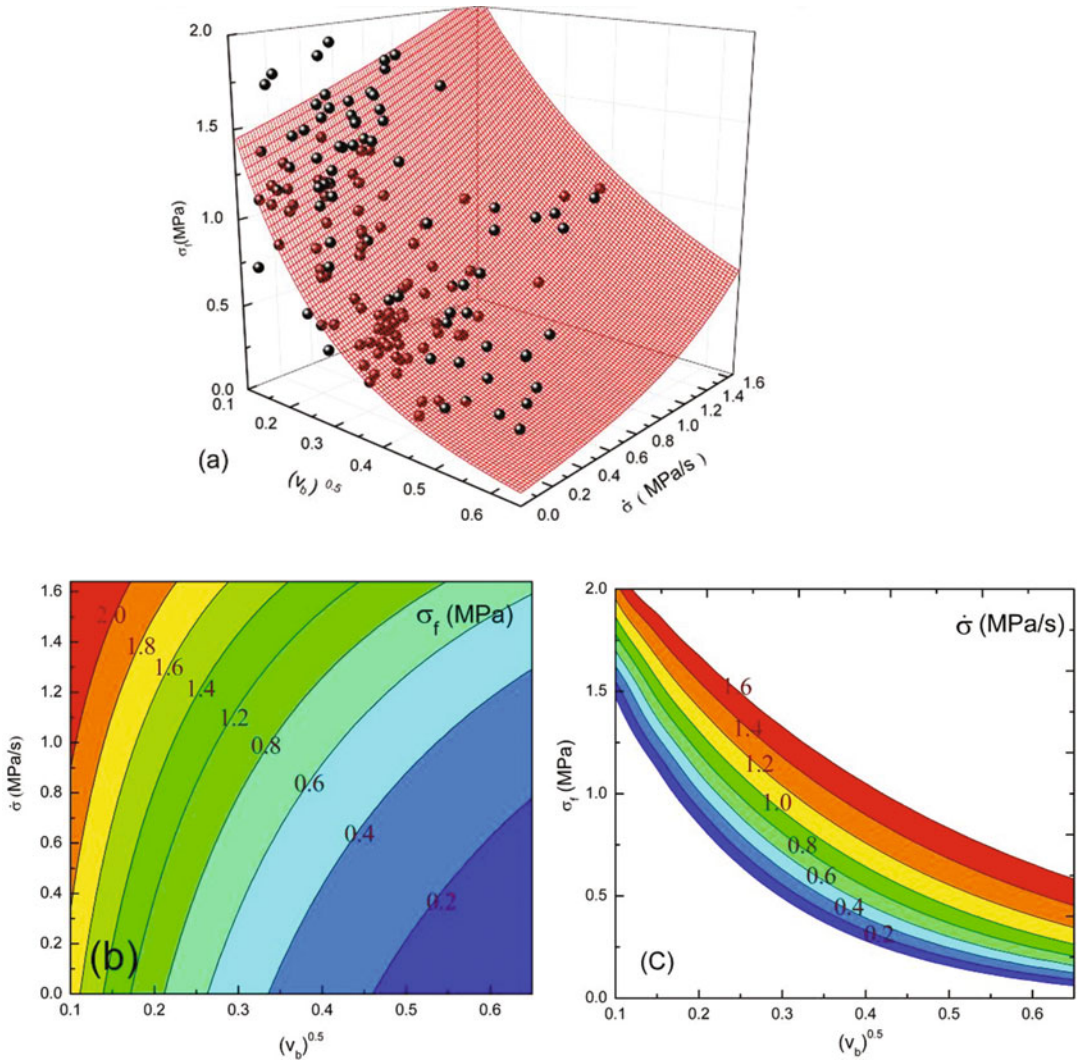
The influence of the scale on the flexural strength is seldom studied, but it is verified by many phenomenon from the large-scale monitor of sea ice such as satellite images (Marsan et al. 2004). For laboratory scale, the sea-ice samples are small volumes, which make the test results hard to be applied into the large-scale condition. Generally, the flexural strength decreases as the

scale become large (Parsonsa et al. 1992; Weiss and Dansereau 2017). For the flexural strength of laboratory, the refined samples could be too small to enclose large quantity of ice crystals. This shortage makes the data not competent to account for grain size effects. Moreover, small samples are likely to reduce the randomness of the distribution of sea-ice flaws. To some degree, the cantilever beam test overcomes this shortage, because it is carried out in situ and maximally preserves the natural flaws of sea ice (Parsonsa et al. 1992). Therefore, we could observe that the flexural strength from the cantilever tests tend to be lower than other measurement methods. However, the conversion equation between different scales has not been given, so much more test data especial from full-scale tests are eagerly to be incorporated to study scale effects.

Loading direction should be another factor influencing the flexural strength. In practice, the direction of the moment is perpendicular to the ice cover, when the interactions between sea-ice covers and engineering structures take place under the current and wave. Thus, the influence of the loading direction on the flexural strength has been paid less attention.

Shortage of Strain Measurement of Sea-Ice Test

As we mentioned, the accurate measurement of strain rate is a key problem to study the



Flexural Strength of Sea Ice, Fig. 4 Curve-fitted flexural strength surface (a), flexural strength contour (b), and stress rate contour (c)

relationship between flexural strength and strain rate. It is not so easy to get the real strain rate, because the length of the time is very short from the beginning of the loading to the break of sea-ice sample. In traditional ice mechanics, displacement actuators, strain gauges, and extensometers are used to measure the deformation of specimen (Timco and Weeks 2010). Those measurements need some operation on the surface of sea-ice sample, and it is likely to bring the local damage and cause the stress concentration. Alternatively, the equivalent displacement is widely adopted to

study the relationship based on the displacement from an indenter (Wang et al. 2019). The loading surface of the specimen and indenter on it move together and thus have the same displacement all the time. Therefore, the strain of the specimens can be deduced from the displacement of indenter without considering the deformation of the rig itself. This method provides only one value to represent the overall deformation of the specimen, even under the assumption that the loading system has enough stiffness compared with that of sea-ice specimen. For sea-ice material, the local variation

in deformation cannot be ignored because of the existence of brine pockets and air bubbles inside of sea ice.

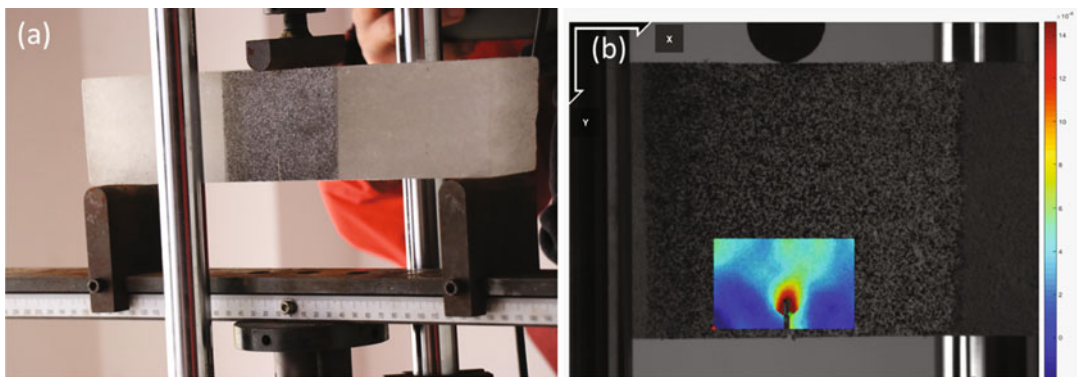
Digital image correlation (DIC) method provides a new concept to obtain strain (rate) from the sequential images. This method has been applied to the deformation measurements of other materials. Based on the DIC, full-field displacement and strain can be accurately obtained by comparing the digital images of specimen surfaces for the initial and deformed states in Fig. 5. Wang et al. (2019) used the DIC to investigate the uniaxial compressive strength and fracture mode for sea ice and lake ice. Moreover, based on the DIC technique, the full-field deformation is accurately deduced, and the damage process of the specimen is clearly obtained at high spatiotemporal resolution.

With the assistance of high-speed camera, the strain can be computed on enough frequency even within a very short time from the loading to the break of sea-ice sample. The DIC technique will facilitate the measurement of sea-ice strain (rate). Then the study of relationship between flexural strength and strain rate will become more accurate. Meanwhile, digital images can be used to capture the local conditions of sea-ice specimens and depict the failure characteristics during the loading process (Ji et al. 2020). This will extend the ability to further explore the complex mechanical behaviors of sea ice.

Discrete Element Method for Simulation of Flexural Strength

Sea ice is composed of solid ice, brine, and gas. This multiphase makes its mechanical properties behave much heterogeneously. Therefore, sea ice is isotropic with granular structures as well as anisotropic with columnar structures. In turn, these differences are reflected to the mechanical properties of the ice (Timco and Weeks 2010). The flexural strength of sea ice is such a mechanical property and depends on many influencing factors such as temperature, salinity, porosity, and grain structure. Consequently, the mechanical properties of sea ice are complex. Experimental investigations in situ or in laboratory do not satisfy the requirement of relationship between flexural strength and its many influencing factors.

Numerical method leverage us another tool to investigate the mechanical properties of sea ice. Considering the multiphase and anisotropic mechanical properties of sea ice, discrete element method (DEM) is being developed to accomplish the goal and has got a better performance. DEM is based on the discontinuous mechanics model, which approaches the real physical composition of sea ice more closely than the traditional numerical methods such as finite element method and finite different method. Several works have been done to look into the failure process of the three-point beam test and to simulate the mechanical



Flexural Strength of Sea Ice, Fig. 5 Digital image correlation applied into the three-point beam test to get the deformation field

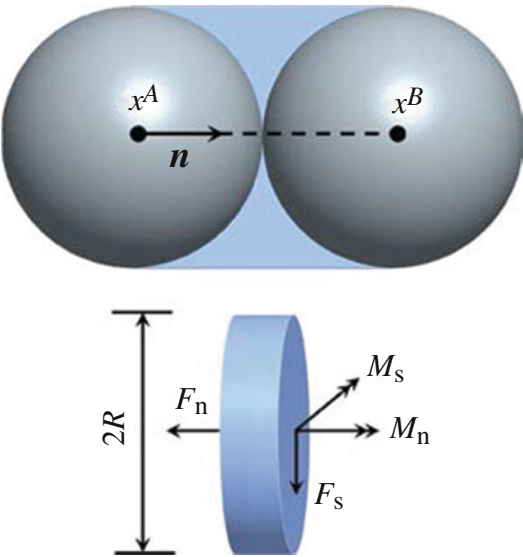
behavior of sea ice in response to ocean waves (Xu et al. 2012).

The principal of the DEM was first proposed in the late of 1970s. Since then, it has been developed to simulate the failure process of brittle material such as sea ice (Long et al. 2018; Bateman et al. 2019). For the DEM, sea ice is approximated as an assemblage of the bonded particles, of which each is dominated by Newton’s second law to simulate the mechanical behavior. On microscale, the main parameters of the particle model include normal and shear stiffness, normal and shear bonding strength, particle diameter, sample size, and interparticle friction coefficient, as shown in Fig. 6. When the interparticle stress surpasses a critical threshold, the bonds between particles break. The flexural strength of sea ice can be obtained depending on these micro-parameters in the DEM.

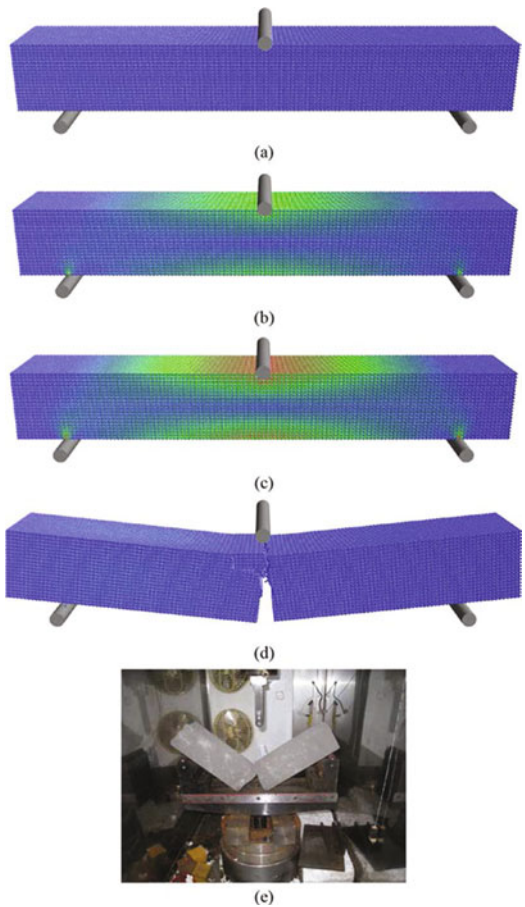
During the simulation using DEM, the particle size, bonding strength, and friction coefficient are the major parameters to influence the flexural strength of sea ice. In order to simulate complex mechanical behavior during the experiment of flexural strength, it is essential to develop an effective method to bridge the gap between the micro-parameters and the macro-strength in the

DEM. In general, these parameters are calibrated and validated according to that of sea-ice strength, as measured in a series of flexural and compressive tests. The assemblage of particles is subjected to the load and then the adjustment of bending stress accomplishes the block fractures in Fig. 7. The interparticle bond strength limits are adjusted until the critical failure stress determined for the numerical sample falls within the ranges confirming to the data from the test of flexural strength.

After filling the gap, the relationship between the flexural strength and physical properties (salinity and temperature) can be checked out. With the development of parallel computation of GPU, the number of the particles can easily reach the million level. As a result, the influencing of



Flexural Strength of Sea Ice, Fig. 6 Parallel bond model between two spherical particles



Flexural Strength of Sea Ice, Fig. 7 Failure processes of three-point bending test (Long et al. 2018)

strain rate on the strength is able to be figured thoroughly, which is not so easy to be obtained for tests in situ or in laboratory. However, there need to be more developments to improve the theoretical basis of DEM numerical approach. Furthermore, the mechanism of more micro-parameter effects on macro-behaviors of mechanics should be analyzed in detail in future works.

Cross-References

- [Ductile-Brittle Transition of Sea Ice Under Uniaxial Compression and Its Engineering Applications](#)

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FLNG

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Synonyms

FLNG (floating liquefied natural gas); LNG (liquid natural gas); LNGC (LNG carrier)

Definition

Floating liquefied natural gas (FLNG) refers to water-based *liquefied natural gas (LNG)* operations employing technologies designed to enable the development of offshore natural gas resources. Floating above an offshore natural gas field, the *FLNG* facility produces, liquefies, stores, and transfers liquefied natural gas at sea before *LNG* carriers ship it directly to markets (https://en.wikipedia.org/wiki/Floating_liquefied_natural_gas).

Introduction

Due to the growing demand for clean energy, natural gas reserves stranded offshore, which were once considered to be quite challenging, are becoming more and more attractive. Those stranded gas fields are commonly located in deep water and are isolated from onshore infrastructures or offshore pipelines, which challenges their exploitation (Zhao et al. 2014). To overcome

those difficulties, floating liquefied natural gas (FLNG), a concept of offshore unit, has been proposed. One of the most famous FLNG projects, Prelude FLNG, located at western coastline of Australia was put into use in 2018. It is illustrated in Fig. 1.

Floating liquefied natural gas (FLNG) refers to water-based liquefied natural gas (LNG) operations employing technologies designed to enable the development of offshore natural gas resources. Floating above an offshore natural gas field, the FLNG facility produces, liquefies, stores, and transfers liquefied natural gas at sea before LNG carriers ship it directly to markets.

The free floating FLNG can be moored directly above the natural gas field. It will route gas from the field to the facility via risers. When the gas reaches the facility, it will be processed to produce natural gas, LPG, and natural gas condensate. The processed feed gas will be treated to remove impurities, and liquefied through freezing, before being stored in the tanks. LNG carriers will offload the LNG, as well as the other liquid by-products, for delivery to markets worldwide (Fig. 2).

FLNG has its unique hydrodynamic characteristics. Due to the large oil and gas processing topside module, wind loads have considerable influence in motion responses of FLNG. FLNG with single-point mooring system is dominated by the weathervane effect to minimize the

environmental loads, including wind, wave, and current loads, on the vessels. Besides, the coupling effect between ship motions and inner-tank sloshing is strong due to the fluidity of LNG (Fig. 3). Tremendous impact forces of inner-tank sloshing threaten the safety of LNG storage systems and greatly change the hydrodynamic performance of FLNG.

A number of major gas and oil companies are still researching and considering FLNG developments, with several initiatives planned for the future. However, the world's first development of FLNG is Shell's Au\$12bn *Prelude FLNG* project, 200 km offshore Western Australia. Royal Dutch Shell announced their investment in FLNG on 20 May 2011, and construction began in October 2012. Numerous other FLNG projects are also underway in regions like North America, West Africa, Brazil, and Southeast Asia. The following shows the distribution of some ongoing FLNG projects worldwide (Fig. 4) (Lovatt 2017).

Development History

Studies into offshore LNG production have been conducted since the early 1970s, but it was not until the mid-1990s that significant research backed by experimental and numerical development began.

The concept design of FLNG was firstly initiated by Shell in 1969. Then, in 1978, FLNG was firstly proposed as an engineering project in

FLNG, Fig. 1 Snapshot of prelude FLNG

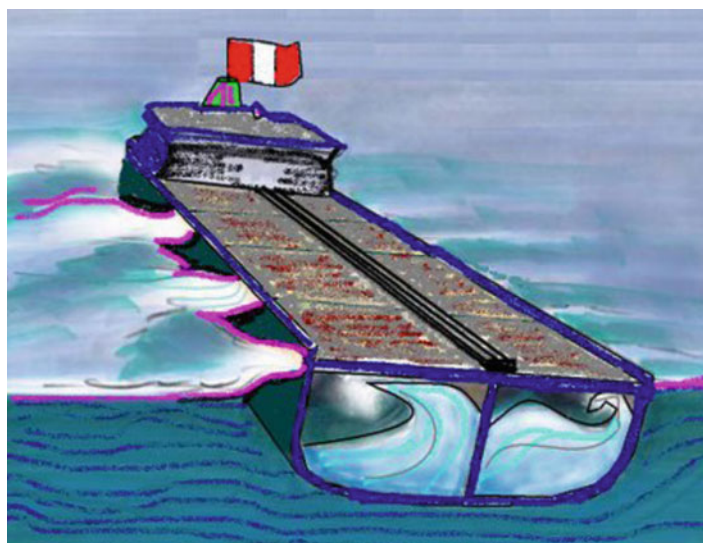




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FLNG, Fig. 2 Offloading process between the FLNG and an LNG carrier

FLNG, Fig. 3 Couplings between inner-tank sloshing and ship motions

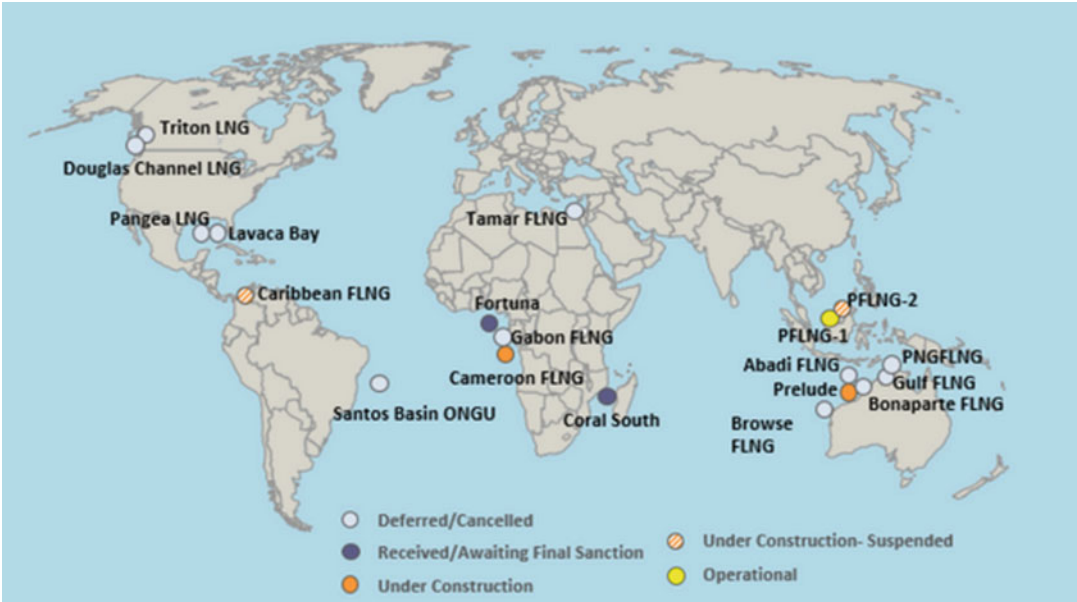


exploitation of gas and oil fields of Kangan, Iran. The first generation of FLNG, Kangan FLNG, was converted from two tankers that are combined side by side to enlarge the deck area (Howard and Andersen 1979) (Fig. 5).

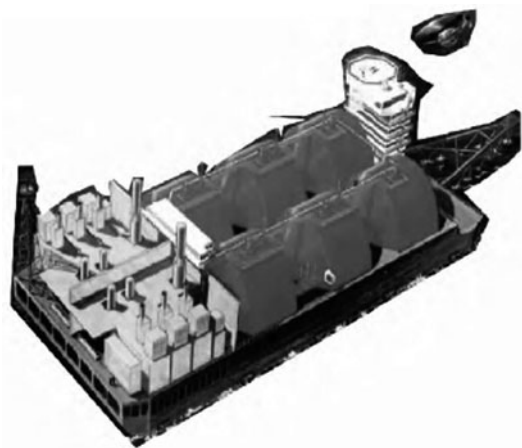
In 1997, Mobil developed a FLNG production concept based on a large, square structure (160 m $\times$ 160 m) with a moonpool in the center. The Mobil proposal was sized to produce 6 million tons LNG per year, with storage provided on the

structure for 250,000 m³ of LNG and 103,000 m³ of condensate.

In 1999, a major study was commissioned as a joint project by Chevron Corporation and several other oil and gas companies. This was closely followed by the so-called “Azure” research project, conducted by the EU and several oil and gas companies. Both projects made great progress in steel concrete hull design, topside development, and LNG transfer systems.



FLNG, Fig. 4 Distribution of FLNG projects worldwide



FLNG, Fig. 5 Kangan FLNG

Since the mid-1990s, Royal Dutch Shell has been working on its own FLNG technology. This includes engineering and the optimization of its concept related to specific potential project developments in Namibia, Timor-Leste/Australia, and Nigeria. In July 2009, Royal Dutch Shell signed an agreement with Technip and Samsung allowing for the design, construction, and installation of multiple Shell FLNG facilities.

As of November 2010, Chevron Corporation was considering an FLNG facility to develop offshore discoveries in the Exmouth Plateau of Western Australia, while in 2011, ExxonMobil was also waiting for an appropriate project to launch its FLNG development.

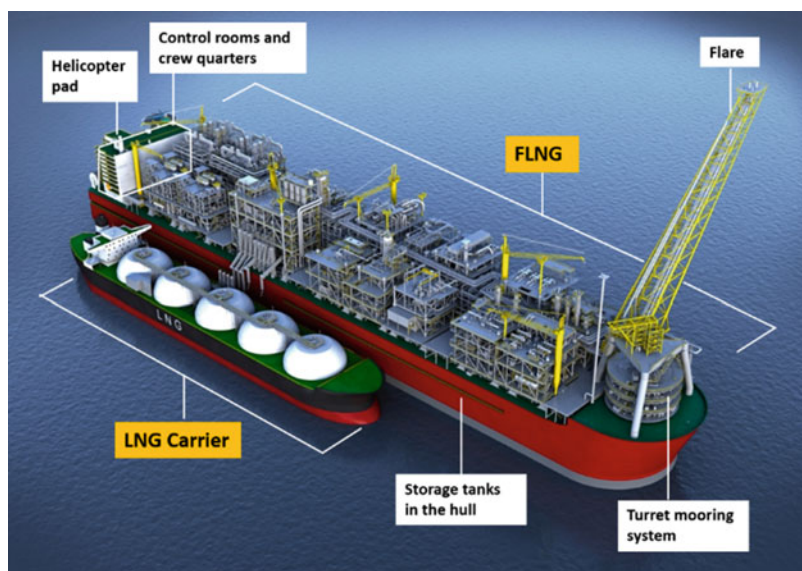
As a result, it is found that the progresses of FLNG development were mainly contributed by large global energy companies due to the complexity of the FLNG systems and the great amount of investment. Meanwhile, a number of shipyards, classification societies, and equipment suppliers have fierce competition and formed a number of research and development alliance to carry out engineering project.

Main Structure of FLNG

FLNG is equivalent to placing an offshore gas liquefaction plant on the deck of a ship hull. However, the deck area is only 1/4 of that of the onshore natural gas liquefaction plant, so that the liquefaction system process should be very compact (Fig. 6).

The process mainly includes preparation of refrigerant nitrogen expander cycle system, liquefaction and condensation extraction system, and gas processing system. The whole system should

FLNG, Fig. 6 Layout of FLNG structures



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be compact, safe, and have low sensitivity to hull movement. Therefore, the main structure of FLNG includes LNG processing equipment, LNG storage system, single-point mooring system, LNG offloading system, and living building, helicopter platform, and flare tower. Figure 6 shows the layout of the main structures of FLNG.

Main Characteristics of FLNG

Advantages

With the maturity of FLNG technology, this engineering concept has been accepted by many energy companies with considerable advantages.

Economic – Utilizing FLNG to develop offshore gas fields has put an end to a single transportation mode where offshore gas fields can only be transported ashore by pipelines, saving large transportation costs and occupying no land space. Besides, FLNG can also be reused twice after the end of gas field mining and be installed in other natural gas fields with higher economic performance.

Environmental – The device can be installed far away from populated areas and is safe and environmentally friendly. Avoiding construction also helps preserve marine and coastal environments. Additionally, environmental

disturbance would be minimized during the later decommissioning of the facility, because it could be disconnected easily and removed before being refurbished and redeployed elsewhere.

The above advantages of FLNG make it the preferred equipment for developing small and mid-sized marginal gas fields offshore.

Technical Challenges

Moving LNG production to an offshore platform presents a set of challenges. In terms of the design and construction of the FLNG facility, every element of a conventional LNG facility needs to fit into an area roughly one quarter the size while maintaining appropriate levels of safety and giving increased flexibility to LNG production. Main challenges include the liquefaction of natural gas, LNG storage, and LNG offloading between FLNG and carriers.

At present, the natural gas liquefaction technology onshore is relatively mature, but offshore operations make the design standard of liquefaction process much different. Liquefaction process of FLNG systems more emphasizes safety, simplicity, compactness, and adaptability to different gas fields and ocean environment. Three methods of liquefaction are normally used in FLNG, including cascade cycle, mixed refrigerant

cycle (MRC), and expander cycle, while they barely satisfy the harsh operation conditions in open seas.

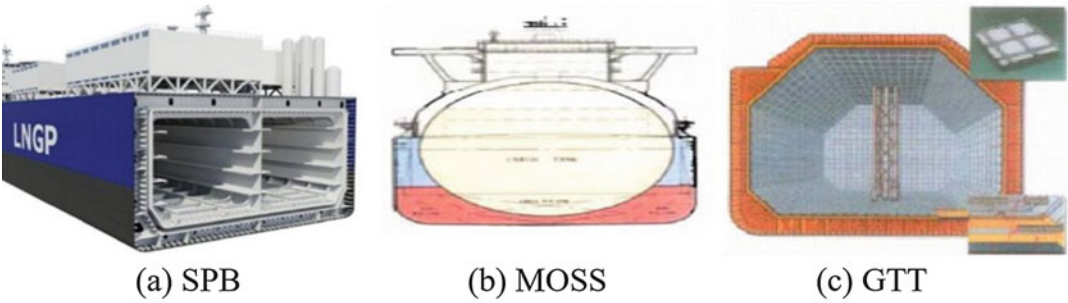
For the storage of LNG, the LNG containment systems need to be capable of withstanding the damage that can occur when the sea’s wave and current motions cause sloshing in the partly filled tanks. Also, LNG should be kept in the low temperature of minus 162 °C during storage process, and the wall of inner tanks will bear a certain vapor pressure. In order to avoid the dangerous situation, the material of the storage tanks and its insulation are very important. LNG tanks that have been successfully put to use can generally be divided as MOSS type, SPB type, and GTT type (Fig. 7).

Side-by-Side Offloading Operation

FLNG (floating liquefied natural gas) system has great safety, economic, and environmental

advantages in the development of marginal gas fields by means of liquefying the natural gas on the production barge and offloading it to a LNG carrier (LNGC). Instead of a conventional method of gas transportation through long pipelines, the cryogenic nature of liquefied natural gas (LNG) determines a small offloading distance between FLNG and LNGC with a side-by-side configuration in offloading operation (see Fig. 8). LNG loading arms (see Fig. 9), as a substitute for hoses, also require the side-by-side offloading configuration, while the relative motions during side-by-side offloading will induce large angles in the offloading arms.

With the increasing transportation amount of LNG, side-by-side offloading operation is a proven methodology for ship lightering and is successfully implemented in the first FLNG project, Prelude FLNG. Many approaches have been developed to investigate and ensure the safety and reliability of offloading operations.



FLNG, Fig. 7 LNG tanks of different types

FLNG, Fig. 8 Narrow gap between side-by-side FLNG and LNG carrier





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FLNG, Fig. 9 LNG loading arms on the FLNG broadside

Nevertheless, there are still challenges that hinder the side-by-side offloading operations. Due to the small gap width (about 4–6 m) between side-by-side moored FLNG and LNG carrier, hydrodynamic interactions are stronger and complex. Relative motions should be strictly controlled because of high risks of collisions (Soares et al. 2015). As a result, the side-by-side offloading operations can only be implemented in relatively benign waters or weather conditions. Special mechanical connection systems, such as fenders and hawsers, are compulsorily deployed to constrain the relative motions for enhancing operational safety.

Besides, some special hydrodynamic phenomenon can also be observed in the gap region that might cause damage to the offloading equipment. For instance, strong wave resonances can be induced at certain wave frequencies in the confined zone between side-by-side vessels. The amplified waves have negative influence on the hydrodynamic performances of both vessels in close proximity and potentially threaten the safety of transport pipelines above the waves. Extensive works have been carried out to study the resonant modes and frequencies of the gap wave field (Yeung and Seah 2007; Molin 2001).

Hydrodynamic Performances of Side-by-Side Moored Vessels

The hydrodynamics of two side-by-side vessels is more complex than that for a single vessel. The difficulty lays in the hydrodynamic interactions, the shielding effects, and the wave resonance observed in the gap between the two vessels.

Shielding Effect

When two vessels locate in close proximity, the shielding effect of parallel vessels in oblique waves is quite obvious. Figure 10 shows the layout of two side-by-side vessels under different wave directions. For the vessel at lee side, as the LNGC in 135deg waves, the wave loads and motion responses are found to be suppressed due to the shielding of vessel at weather side.

Shielding effect of each vessel is different due to its size, displacement, and wave directions. Comparisons of motion responses under different wave directions show that motion responses of FLNG are insusceptible to the change of wave directions because of its large inertia, while motion responses of LNGC at the lee side (135deg) are more stifled due to the stronger shielding effect of FLNG. Interestingly, either LNGC at lee or weather side has smaller amplitude of sway or yaw motion than that free floating solely, owing to the suppression effect of the FLNG aside.

Gap Wave Resonance

Gap wave resonances are induced by incoming waves with frequency close to the natural frequencies of the gap water. Wave energy is gradually accumulated in the gap zone, and resonances finally get developed. Hence, gap wave elevations can be amplified even in mild weather conditions.

Divided by the types of restoring forces, two resonant modes can be defined, namely, the piston mode and sloshing modes (see Fig. 11).

For the piston mode, gap water (gray area between two hulls in Fig. 11a) acts like a piston moving up and down periodically dominated by the gravity. For the sloshing modes at higher frequencies, a complex pattern of wave amplification and cancelation can be seen in the gap region. Standing waves are triggered by the interference between waves reflected back and forth at the gap exits into the open ocean (Fig. 11b). The gray area shows the parallel part of the gap. The blue line is the incoming wave, the dotted line the reflected wave.

Many numerical studies have been implemented to unravel the physics of resonant gap waves. In addition to computational fluid

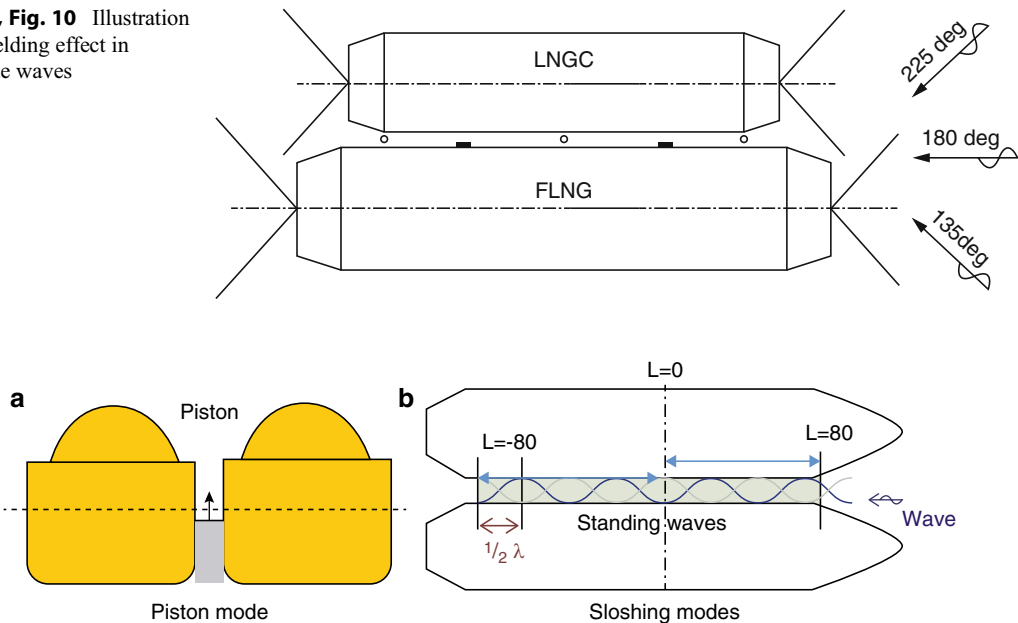
mechanics (CFD) methods, diffraction computation based on potential flow theory is also widely used. Nevertheless, resonance behavior of the wave in the gap always tends to be overestimated using standard diffraction programs. Viscous effects, neglected in diffraction theory, may be more dominantly present at the bilges of the keel due to resonant flow between the bodies.

Mechanical Connections

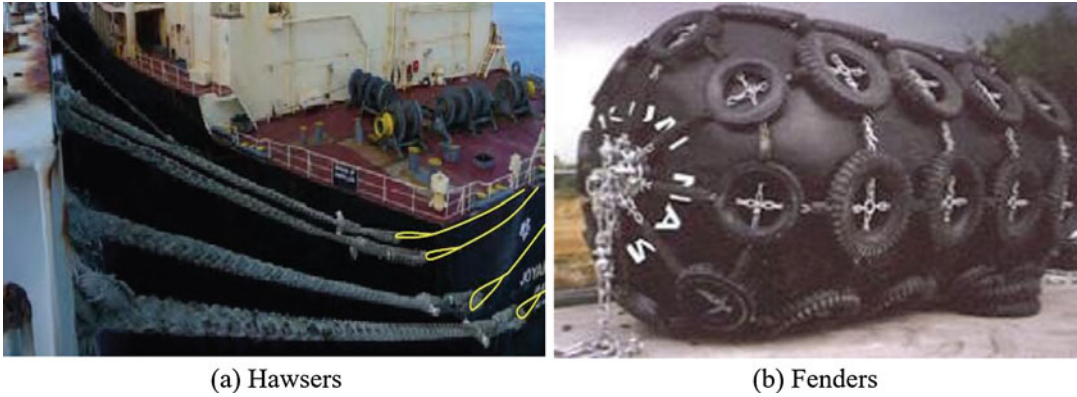
During offloading operations, mechanical connections between FLNG and LNGC, like hawsers and fenders (as shown in Fig. 12), are used to restrict fierce relative motions between vessels and lower the intensity of collisions. The LNGC are moored to the FLNG in side-by-side configuration with hawsers having the segments of nylon stretchers and steel wires (yellow lines in Fig. 12a). To avoid hard collisions of two adjacent vessels, fenders are also used to soften the contacts between two huge floaters (Voogt and Brugts 2010).

Normally, the fenders are floating on the free surfaces in the gap zone, and the hawsers are located obliquely at the bow, stern, and middle

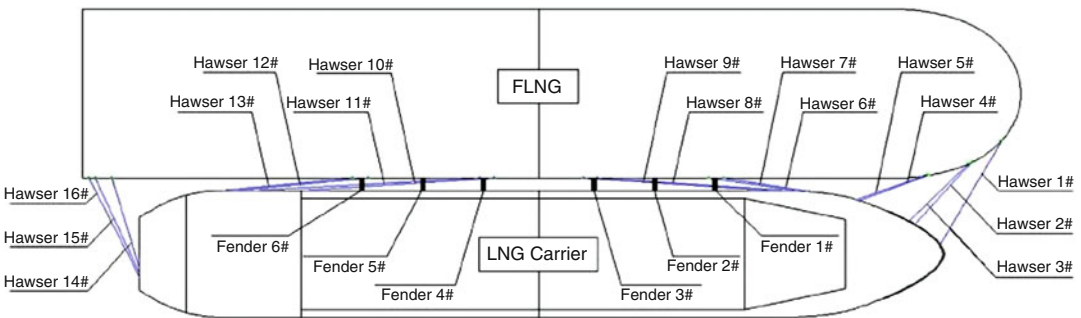
FLNG, Fig. 10 Illustration of shielding effect in oblique waves



FLNG, Fig. 11 Different resonant modes of gap waves



FLNG, Fig. 12 Illustration of hawsers and fenders



FLNG, Fig. 13 Layout of hawsers and fenders

areas (see Fig. 13). Hawsers are pretensioned to restrain the relative motions in the horizontal plane, mainly the surge and yaw motions. The time series of the forces acting on the fenders are in the form of pulses, which indicates that the two vessels are separated in some cases and there are also many collisions between the two vessels (Zhao et al. 2014).

Both the pretensions and the stiffness of the hawsers can significantly affect the hydrodynamic performance of an FLNG system in side-by-side offloading operation, such as the trajectories of the vessels, the relative motion responses between the two vessels, and the loads acting on the hawsers and the fenders. The decrease of the pretension and the stiffness of the hawsers will lead to a reduction of collisions between the two vessels, but an increase of the relative motions between the two vessels in horizontal plane.

Technical Challenges

Offloading Arm

For transportation of liquid natural gas from FLNG to LNGC, novel offloading arms have been developed to overcome the relative motions between two vessels. The company FMC Technologies in France is famous for the offloading arm design capability and has been taken the job for Prelude FLNG of Shell offloading arm design (see Fig. 14). The arm can swivel, rotate, and follow the motion of LNGC for over 15 h during offloading operation. Besides, the arm should perform in extreme cold for the LNG should be stored in -162°C . Once the movement of arm exceeds the safe limit in extreme sea conditions, it should automatically break away from the LNGC and release the LNG left between two valves in the connection head (like the white gas in Fig. 14).

FLNG, Fig. 14 LNG offloading arms



FLNG, Fig. 15 Deployment of offloading arms on FLNG



In FLNG offloading operation, a group with six to eight offloading arms can be deployed on the starboard of the vessel to guide the pipelines to LNGC (Fig. 15). In such circumstance, LNG can be transferred efficiently and safely in side-by-side configuration.

Offloading Pipe

Offloading pipe should withstand extreme low temperature and the fatigue risks due to relative motions between two vessels. Hence, the pipe should have outstanding heat-insulating property using low-temperature-resistant materials. Vacuum annular is adopted to cut off heat transfer between LNG and the air outside. Details of the pipe structure can be obtained

FLNG, Fig. 16 Offloading pipe cross section



in Fig. 16, with illustration of each layer on the right side.

- 1. Protection sheet
- 2. Flexible stainless steel outer pipe

3. Vacuum annular with superinsulation for best insulation property and leak detection
4. Stainless steel armor
5. Flexible stainless steel inner pipe

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FLNG (Floating Liquefied Natural Gas)

► FLNG

Floating Bridge

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Synonyms

Floating curved bridge; Floating suspension bridge; Pontoon bridge; Ribbon bridge

Definition

A floating bridge is a continuous bridge supported by the buoyancy of a required number of pontoons or floating rafts instead of fixed piles and is a means for pedestrians and vehicles to cross rivers or streams. It is mainly suitable for two scenarios. One is a temporary bridge that can be used in emergency rescue or warfare for relief supplies, wartime troops, weapons, and equipment to quickly pass river barriers. The other one is a permanent civil bridge to cross rivers or seas where the width and the water depth are so large or the bottom is covered by soft silt so thick that it is technologically easier and more economical to utilize floating foundations.

Scientific Fundamentals

Historical Development

The term “floating bridge” refers to a bridge floating on the water supported by a required number of boats or pontoons instead of bridge piers. Before the twentieth century, floating bridges were mostly used for military warfare, so they were mostly used temporarily. The earliest historical record of floating bridges is in China. It is recorded in the *Shi Jing* (Book of Odes) that King Wen of Zhou was the first to create a pontoon bridge in the eleventh century BC. The first permanent floating bridge was constructed in the Qin Dynasty (221–207 BC). Then, during the Eastern Han Dynasty (25–220 AD) and Song Dynasty (960–1279 AD), various pontoon bridges spanning the Yellow River and Yangtze River were constructed (<https://en.wikipedia.org/>). Among them, the Guangji Bridge was constructed in 1171 and renovated in 2007, as shown in Fig. 1. It is a combination of fixed and floating bridges with a total length of 518 m. The floating part is 97.3 m long and is supported by 18 wooden boats.

In Europe, the first floating bridge was built by the Persian Emperor Darius to cross the Bosphorus as he invaded Greece approximately in 500 BC, as recorded by the Greek writer Herodotus. Then, in 480 BC, Xerxes I built a pontoon bridge crossing the Hellespont to transport a large army to Europe (<https://en.wikipedia.org/>), as illustrated in Fig. 2.



Floating Bridge, Fig. 1 Guangji Bridge in China. (Photograph by Luis Evers; from Wikimedia)



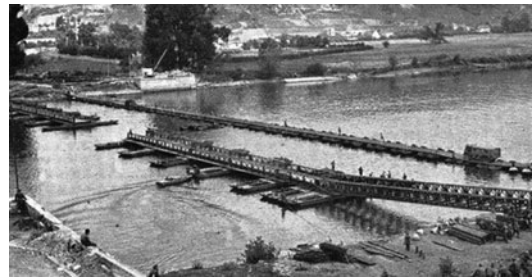
Floating Bridge, Fig. 2 Construction of Xerxes' bridge of boats. (Photograph by A. C. Weatherstone)

In the Middle Ages and early modern period, pontoon bridges were widely used in wars. During the Fifth Crusade, the Crusaders built two pontoon bridges across the Nile. One of them was supported by 38 ships. This kind of bridge was also used in the Battle of Rio Bueno by the Spanish army in 1654, the Battle of Montebello by French troops in 1800, and the American Civil War.

During the twentieth century, floating bridges were still used for cross-river operations in the war, such as the First and Second World Wars, as shown in Figs. 3 and 4. Meanwhile, with the development of bridge construction technology and the special terrain requirements of crossing rivers and seas, more permanent civil floating bridges have appeared, especially in the United States and Norway, and have received widespread attention.



Floating Bridge, Fig. 3 Floating bridge of the US during World War II (www.wikiwand.com)



Floating Bridge, Fig. 4 UK Military Bridging – World War II (thinkdefence.co.uk)

Classification

Pontoon Road Bridge

From ancient days to the present, the design of the pontoon road bridge has not undergone major changes either for war or civil use. The roadways and the supported floating pontoons are normally separated. In war, for temporary use, the pontoons are usually designed as smaller ships or rafts. Meanwhile, for the modern permanent civil floating bridges, the pontoons are mostly made of reinforced concrete or steel structures.

Ten of the longest pontoon bridges for civil use are listed in Table 1 (Kvåle 2017). These have been constructed because either the bottom topography or the length and depth are not appropriate for bridges supported by fixed foundations. The pontoon bridges in the USA and Japan listed in Table 1 are all due to the bottom topography. For instance, at the location of the Evergreen Point Floating Bridge, the lake has a water depth of 61 m atop 61 m of soft silt. Constructing fixed foundations would be technically more

Floating Bridge, Table 1 Ten of the longest pontoon bridges for civil use (<https://www.enr.com/>)

Name	Location	Length	In operation	Pontoon design
Evergreen Point Floating Bridge	U.S.A, Seattle, Lake Washington	2350 m	2016-	Continuous with side anchor
Lacey V. Murrow Memorial Bridge	U.S.A, Seattle, Lake Washington	2020 m	1940-	Continuous with side anchor
Hood Canal Bridge	U.S.A, Hood Canal	1988 m	1961-	Continuous with side anchor
Demerara Harbour Bridge	Guyana, Georgetown, Demerara River	1851 m	1978-	Discrete steel pontoons with side anchor
Homer M. Hadley Memorial Bridge	U.S.A., Seattle, Lake Washington	1772 m	1989-	Continuous with side anchor
Berbice Bridge	Guyana, Berbice River	1571 m	2008-	Discrete pontoons with side anchor
Nordhordland Bridge	Norway, Bergen, Salhusfjorder	1246 m	1994-	Discrete concrete pontoons without side anchor
Bergsoysund Bridge	Norway, Møre og Romsdal county, Bergsoysundet fjord	933 m	1992-	Discrete concrete pontoons without side anchor
William R. Bennett Bridge	Canada, British Columbia	690 m	2008-	Continuous with side anchor
Yumemai Bridge	Japan, Osaka	410 m	2000-	Two discrete pontoons

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challenging as well as too expensive. In contrast, in Norway, floating bridges are built mainly for the crossing of fjords with a large width and water depth. Among them, the Salhusfjorden spanned by the Nordhordland Bridge has a water depth of 320 m and width of 1246 m. Norway is currently in the process of designing a new pontoon bridge for the Bjørnafjorden, which has a span and a depth of more than 5,000 m and 550 m, respectively. In addition, the design of arch-shaped bridge girders and pontoons without side anchors is more popular in Norway, as shown in Fig. 5.

From the perspective of the shape of the pontoon, the floating bridges in the USA listed in Table 1 are mainly supported by continuous pontoons. The bridges in Norway, Guyana, and Japan in the table are all designed using discrete pontoons. This is partly attributed to the environmental conditions in the different locations. In Norway, the bridges are normally located in open seas where the wind, waves, and current are very strong. The use of discrete pontoons with the long sides of the buoy parallel to the main wave and current directions can effectively reduce the environmental load and ensure safety. In the USA, the Evergreen Point Floating Bridge

and Lacey V. Murrow Memorial Bridge are in the inner lake and suffer relatively more gentle environmental conditions. The continuous design yields better dynamic performance.

Military Ribbon Bridge

In a military ribbon bridge, the foot boat and the roadway are normally merged into a whole. It is different from the pontoon road bridge in which the roadway and the floating pontoons or small supported ships are constructed separately. The advantage is that it can greatly reduce the number of components to ensure more rapid erection and disassembly with fewer operators. Due to its outstanding features in rapidity and mobility, the ribbon bridge is widely used in emergency rescue or warfare for relief supplies, wartime troops, weapons, and equipment to quickly pass river barriers.

The military ribbon bridge was developed earlier in Europe and America. The first ribbon bridge was designed by the Soviet Union after World War II and named the PMP ribbon bridge, as shown in Fig. 6. It was deployed in 1962 and subsequently adopted by Warsaw Pact countries and other states employing Soviet military



Floating Bridge, Fig. 5 Several permanent pontoon bridges for civil use. **(a)** Evergreen Point Floating Bridge. (Photograph by Jelson25; from Wikimedia). **(b)** Demerara Harbour Bridge. (Photograph by Dan Lundberg; from Wikimedia). **(c)** Yumemai bridge. (Photograph by Ignis;

from Wikimedia). **(d)** Homer M. Hadley Memorial Bridge (left) and the Lacey V. Murrow Memorial Bridge (right). (Photograph by SounderBruce; from Wikimedia). **(e)** Nordhordland Bridge. (Photograph by Aas-Jakobsen AS)

equipment. In 1969, the US military began to develop a similar type of the PMP ribbon bridge, which was officially equipped with troops in 1976. In 1974, Chinese troops designed and manufactured the first-generation “74-type” ribbon bridge. Since the 1980s, countries have begun to compete with ribbon bridge equipment upgrades, such as the US Improved Ribbon Bridge (IRB), the German M3 self-propelled ribbon bridge, and the French EFA frontline crossing equipment, as shown in Fig. 7.

Floating Suspension Bridge

The concept of floating suspension bridges is a combination of offshore ocean engineering and bridge engineering proposed by the Norwegian Public Road Administration. It mainly consists of two components, i.e., a traditional suspension bridge above the water and several tension leg platforms (TLP) in the water. The design of three-span suspension bridges for the crossing of Bjørnafjorden along the E39 highway of Norway is shown in Fig. 8. The fjord has a length of over



Floating Bridge, Fig. 6 Hungarian PMP Pontoon Bridge that crosses the Sava river at Slavonski Brod, Croatia, 1996. (Photograph by SSGT Edward W. Nino - <http://www.defenseimagery.mil>)



Floating Bridge, Fig. 7 US and German Army units that cross the river Elbe using German M3 amphibious bridging vehicles, 2015. (Photograph by Sgt. Jesiah Dixon, U.S. Army)

5 km and a depth of approximately 550 m. For such great water depths, bridges with foundations fixed on the seabed are not advantageous. The performance or the feasibility of the floating suspension bridge has been evaluated by several consultancies in Norway since 2014, and the work is still continuing.

In these new concepts, the main cables are supported by two fixed pylons at each end and one or two floating pylons in the middle of the bridge. The bottom part of the floating pylon is similar to tension leg platforms moored by four groups of tethers, providing a high stiffness in heave, pitch, and roll. The TLP, originally developed for the oil and gas industry, is the main point that enables the bridge to be built on greater depths than what had been possible for the fixed type (Veie and Holtberget 2015). In the design of the bridge across the Bjørnafjorden, each of the main girder spans has a length of 1385 m. The pylons are 198 m high and have a draft of 65 m. The water depth is 550 m and 450 m at the left and right floating pylons, respectively.

Key Technology

Environmental Loads

For a bridge with a fixed foundation, wind, traffic, and earthquakes are usually considered. In addition to such traditional actions, wave-, current- and tide-induced load effects must also be included in the pontoon bridges, which bring more challenges into the design.



Floating Bridge, Fig. 8 Three-span suspension bridge with two floating pylons. (Illustrated by Arne Jørgen Myhre, Statens Vegvesen)

The wind action on the slender structures, such as the steel column, tower, and cable, is usually modeled by viscous drag forces. For bridge girders, it can be more complicated. In principle, it is composed of the mean and buffeting forces due to the mean and turbulent wind velocities and the self-excited forces induced by the motion of the structure (Xu et al. 2017). The mean and buffeting forces are normally calculated based on quasi-steady theory using the frequency-dependent drag, lift, and torsion moment force coefficients and their derivatives with respect to the angle of attack (Strømmen 2010), which can be achieved from wind tunnel experiments or CFD simulations (Helgedagsrud et al. 2019; Siedziako and Øiseth 2018). The self-excited force is relatively more complicated in the simulation since it is related to the oscillation frequency of the structure. Theodorsen's and Wagner's functions are the renowned expressions representing the aerodynamic motion-induced forces on an airfoil in the frequency and time domains, respectively (Theodorsen 1935; Wagner 1925). The theories are generalized to describe the motion-induced forces acting on a bridge deck by other researchers. The commonly used expression for a bridge deck includes the aerodynamic derivatives, indicial function, rational function, and state-space model (Øiseth et al. 2012; Scanlan and Tomo 1971).

The hydrodynamic actions consist of hydrostatic forces, wave excitation forces, and radiation forces. The hydrostatic forces are frequency independent and can provide a restoring force for the structures, usually modeled as a spring in the static and dynamic analyses. The wave excitation forces related to the fluctuating pressure on the submerged surface of the pontoons are caused by incident waves assuming that the structure is fixed, which consist of first-order, second-order difference and sum-frequency excitation forces (Faltinsen 1993). The magnitude of the first-order forces is always larger than that of the second-order forces and is defined by a transfer function that can be obtained by potential theory. The natural frequencies of the low-order modes of the long and slender floating bridge are well below the energy content in the wave spectra such that

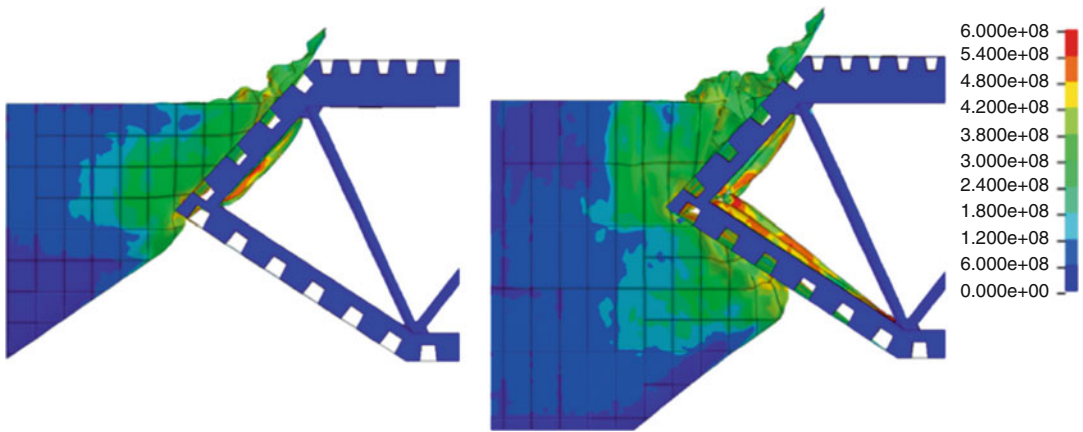
these modes will not be excited by first-order excitation forces. However, the difference-frequency forces can cause wave excitations in this frequency range even though they are of small amplitude. The sum-frequency forces introduce excitation at higher frequencies compared to the first-order excitation forces, which are unlikely to be an important factor for the floating bridges. When a floating structure oscillates in waves or still water, it generates outgoing waves, resulting in oscillating fluid pressures on the surface of the body. The integrated hydrodynamic pressures give rise to radiation forces, which are a combination of frequency-dependent added mass and potential damping.

The current-induced loads on the pontoons are also normally modeled by viscous drag forces using the modified Morison equation (Morison et al. 1950). The effect of the tide is also quite important for floating bridges, which is modeled as a constant force equal to the product of the tide level and the hydrostatic restoring stiffness.

Hydro- and Aeroelastic Response Analysis

Different from the oil platforms that are normally considered as rigid bodies, floating bridges are super slender structures, and elastic deformation is of the same magnitude or even much larger than rigid-body motion. The coupling between the elastic deformation and the wave-induced or aerodynamic loads increases the complexity of understanding the dynamic characteristics of such structures.

For ribbon bridges, only hydroelastic response should be considered. Research on this topic has been ongoing since the 1970s. Two-dimensional hydroelastic theory, originating from strip theory, was widely used and studied at that time. In 1984, Wu first proposed a three-dimensional hydroelastic theory in the frequency domain (Wu 1984). It can be applied to very large floating structures with any complex geometry. Subsequently, a great amount of research on time-domain hydroelastic theory was performed in order to take into consideration moving loads or different kinds of nonlinear features (Chen et al. 2003; Wei et al. 2018).



Floating Bridge, Fig. 9 Structural deformations of ship-girder collision (Sha et al. 2019)

For bridges supported by discrete floating pontoons, such as the Nordhordland Bridge (in operation) and the floating suspension bridge (still in design), they can be considered a multi-body system (floating pontoons) connected by a simplified beam (bridge girder) in the hydro- and aeroelastic response analysis. For the aeroelastic theory, it is basically the same as for traditional fixed-type bridges. The main difference is that the multibody hydrodynamics and the aeroelastic response have to be considered simultaneously (Xu et al. 2018a, b).

For the bridge supported by continuous pontoons, such as the Lacey V. Murrow Memorial Bridge, both hydro- and aeroelastic responses should carefully be taken into consideration.

Ship Collisions

Ship collision is a great threat to bridges across navigation waterways. Possibilities exist for collisions between various types of surface ships with different sizes and speeds and bridge girders and pontoons. Such a collision is a complex nonlinear dynamic process occurring in a very short time but with large impact loads. A variety of nonlinear properties of materials, geometry, and contact introduce tremendous challenges into the accurate numerical simulation of the collision process. To reduce collision occurrence, various measures such as specifying ship lanes (vertical and horizontal as well as keel clearance at the bridge), speed limits, safety zones, traffic control,

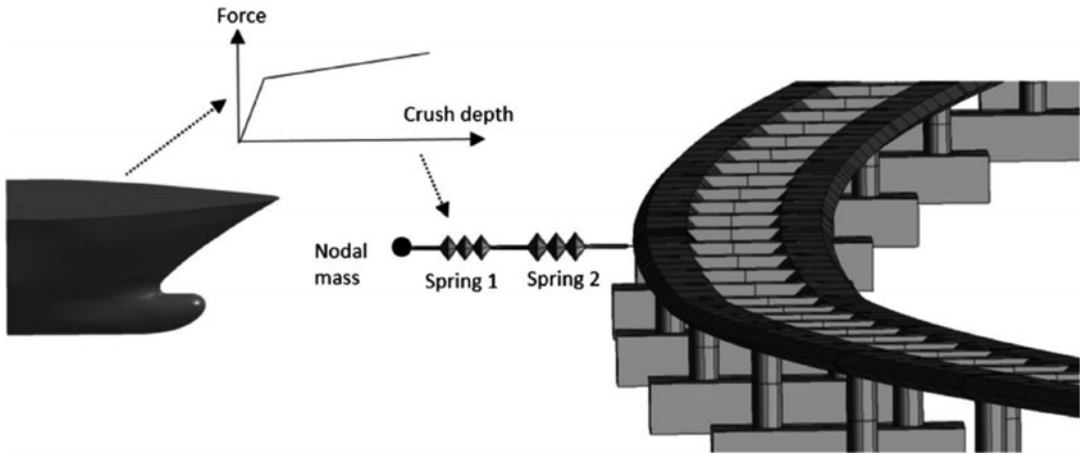
and surveillance should be put into effect (Moan and Eidem 2020).

Compared to bridges with fixed foundations, the floating pontoon bridges are more flexible and are likely to experience a large deformation under ship collision loads. Therefore, if the collision energy was assumed to be absorbed locally, as is normally done on fixed type bridges, the local deformation would be significantly overestimated for floating bridges since the global motion can also contribute to the energy dissipation (Sha et al. 2019). Local and global responses of a new concept of floating bridge in Norway under collisions between ships and bridge girders have been investigated. A traditional local collision load estimation approach was first employed to achieve the relation between the force and local deformation, as shown in Fig. 9. Then, in the global collision analysis, the ship is modeled by a mass-spring system connected to the bridge, as shown in Fig. 10. The stiffness of the spring is determined by the force-deformation curve achieved in local analysis. It is found that approximately 40% of the collision energy can be dissipated by the global motion of the bridge.

Key Application

Lacey V. Murrow Memorial Bridge

The Lacey V. Murrow Memorial Bridge is the first bridge to cross Lake Washington in Seattle, USA.



Floating Bridge, Fig. 10 Global collision analysis (Sha et al. 2019)

It was opened in 1940. Since the bottom of Lake Washington is covered by very thick mud deposits, it was challenging to build fixed foundations. As an alternative and feasible design, the bridge deck is supported by 22 floating pontoons made of reinforced concrete with dimensions of $106.7 \times 18 \times 4.3$ m. Each one is divided into 96 cells with four across the width and 24 along the length (Dusenberry et al. 1995). It is the world's first floating bridge to be built using concrete pontoons. Adjacent pontoons are bolted together end-to-end and kept in position by side anchors. At the ends, the roadway is elevated on beams and columns and is sloped upward to the transition spans connected to the fixed spans. Fixed spans connected the floating bridge to the approach highways.

The bridge was closed for refurbishment on June 23, 1989. The Washington State Department of Transportation decided to use hydrodemolition to remove the unwanted materials on the bridge and pontoons. The water from hydrodemolition was considered contaminated and was not allowed to flow into the lake. By calculation, it was found that the floating pontoons had the potential to store the contaminated water temporarily. During the Thanksgiving Day holiday (November 22–24, 1990), the bridge was attended intermittently by only a few people. Simultaneously, a strong storm passed by the lake. On the last day of the holiday, it was found that one pontoon was



Floating Bridge, Fig. 11 The Lacey V. Murrow Floating Bridge sank to the bottom of Lake Washington on Nov. 25, 1990 (<https://mynorthwest.com/>)

beginning to sink, as shown in Fig. 11. Although a significant effort of pumping the water out of the pontoon had been done, it was too late to stop the disaster (Dusenberry et al. 1995). In the end, an 850-m-long section of the bridge sank to the bottom of the lake. The disaster postponed the reopening of the bridge for 14 months and resulted in a large financial loss.

Bergsøysund Bridge in Norway

The Bergsøysund Bridge is a floating pontoon bridge on the western coast of Norway with a length of 930 m, operated by the Norwegian Public Road Administration, as shown in Fig. 12. It spans the Bergsøysund fjord with a water depth of 320 m, providing a permanent road link between the islands of Aspøya and Bergsøya. It consists of



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Floating Bridge, Fig. 12 The Bergsøysund Bridge. (Photograph by NTNU/K. A. Kvåle)

a continuous steel truss superstructure and seven floating pontoons. The pontoons are shell structures that are made of lightweight aggregate concrete, providing buoyance force or restoring stiffness in the vertical and torsional directions. The bridge has no side-mooring or anchoring and relies fully on a flexible rod connected to an abutment on the land at each end, resulting in a construction method that does not involve excavation of the seabed (<http://www.eng.mortemor.com/>). To ensure sufficient lateral stiffness, the bridge is designed in an arc shape with a radius of curvature of 1300 m.

The Bergsøysund Bridge has been instrumented with an extensive monitoring system since it was opened in 1992. Six wave height sensors and five anemometers are installed to collect wave and wind data at the site in real time (Kvåle and Øiseth 2017). The dynamic response of each pontoon is monitored online through tri-axial accelerometers and the Global Navigation Satellite System. Due to the success of the Bergsøysund Bridge, another similar but longer floating pontoon bridge, the Nordhordland Bridge, was opened in 1994.

Improved Ribbon Bridge (IRB)

The Improved Ribbon Bridge (IRB) is a modular and transportable bridge system developed by General Dynamics European Land Systems. It is designed as an efficient way for heavy military



Floating Bridge, Fig. 13 Improved Ribbon Bridge (<https://militaryleak.com/>)

equipment such as loaded heavy equipment transporters, weapons systems, wheeled/tracked combat vehicles, trucks, suppliers, and troops to cross wet-gap obstacles, such as rivers or streams.

A complete ribbon bridge is normally composed of two ramp bays at the end connected to the land and a number of interior bays between them, as shown in Fig. 13. The number depends on the width of the specific wet-gap obstacle. The bays can be disassembled and transported on aircraft, helicopters, trucks, and railway cars in a folded condition.

The IRB ramp and interior bay is a four-pontoon module with two inner pontoons that form a wide roadway and two outer pontoons that provide walkways on both sides, as shown in Fig. 14. The outer pontoons are designed in a



Floating Bridge, Fig. 14 The IRB ramp and interior bay (Trim and Bennett 2020)

bow shape in order to decrease the current-induced drag loads acting on the bridges in the perpendicular direction. The four pontoons are joined by hinges and pins along their adjacent edges that allow the bay to fold and unfold. It unfolds automatically by the force of gravity and buoyancy generated as the bay is released and begins to float (Trim and Bennett 2020). Since they have no power system, the modules have to be allied and connected manually.

Cross-References

- [Experimental Techniques for VLFS](#)
- [Hydroelasticity Theory](#)
- [Station-Keeping System for VLFS](#)
- [Structural Analysis and Design of VLFS](#)
- [Very Large Floating Structures \(VLFS\): Overview](#)

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Floating Curved Bridge

- [Floating Bridge](#)

Floating Production Storage and Offloading (FPSO)

- [Polar Offshore Engineering](#)

Floating Production Storage and Transportation Systems (FPSO)

- [Steel Pipelines and Risers](#)

Floating Production Storage Offloading (FPSO)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Floating Structure

- [Design of Mooring System](#)

Floating Suspension Bridge

- [Floating Bridge](#)

Floe Ice

- [Numerical Simulation Floe Ice–Sloping Structure Interactions](#)
- [Sloping Structure: Floe Ice Interactions](#)

FLS – Fatigue Limit States

- [Design of Renewable Energy Devices](#)

Fluid Dynamics of Net Structure

- [Net Structures: Hydrodynamics](#)

Fluid Effects During Iceberg-Structure Interactions

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Introduction

The ice impact forces on a fixed or floating offshore structure are an important structural design consideration in the Arctic region. To predict the impact forces, a knowledge of velocity, acceleration, and location of both ice mass and structure before impact is necessary. These are heavily influenced by the ice conditions, the structure geometry, the hydrodynamic forces, as well as the location of the ice mass relative to the structure.

The hydrodynamic effects of the surrounding water play an important role in the analysis of collisions between ice mass and fixed or movable structure. For instance, hydrodynamic forces cause an ice mass or floating structure to move

before the actual collision and consequently affect the impact bodies' velocities and trajectories. Especially, for small ice masses and bergy bits, these effects are more obvious (Sayeed et al. 2017). This will result in inaccurate predications of absorbed energy and impact forces. Therefore, it is necessary to take into account the hydrodynamic effects when dealing with ice-structure interactions.

The hydrodynamic effects such as negative wave drift force, fluid cushioning, shadowing, reduction in impact velocity, and hydrodynamic damping are very complex. A number of studies have investigated fluid effects during ice-structure interaction. According to the locations of the ice mass relative to the structure, the studies in this area can be classified into three phases, namely, far-field, near-field, and contact phases.

Far-Field Phase

The far-field phase focuses on ice mass drift in open water, where the location of the ice mass is far from the structure and their interaction can be neglected. The motion of the ice mass is heavily dependent on the ice properties, the size and shape of the ice, and environmental conditions.

Hsiung and Aboul-Azm (1982) developed a mathematical model to predict iceberg drift pattern. The model included the wave drift force and the other conventional force components such as forces due to wind, current, Coriolis effect, and geostrophic effect. Trajectories of two icebergs with the wave effect and without the wave effect were calculated and compared with field results. It was shown that the wave effect had a significant improvement in the correlation.

Pavec and Coche (2007) proposed a model to predict the short-term drift of an iceberg by a prediction of the forces that act on the object. In the model, the wave effect was included, which improved significantly the estimation in some cases. To deal with the uncertainties associated with the parameterization of waves, a stochastic calculation was also included into the model. The calculated drifts were generally in good agreements with the test data. The results emphasized

the need of precise current forecast and the importance of the wave-induced force.

Near-Field Phase

In the near-field phase, the hydrodynamic effects are complex since the surrounding flow fields are modified for different distances between the ice mass and structure. The presence and the motion of the other body have significant effects on the drift motion of the ice during the approach.

Isaacson (1987) investigated the motions of an ice mass due to waves and a current near an offshore structure, in which the nonlinear interactions between waves and current were neglected. The drift trajectory which takes into account the current drag, the wave drift forces, and the added mass effects was obtained through a time-stepping procedure. However, this three-dimensional model was not applicable to situations where the separation distance between the two objects was very small compared to the size of the bodies due to the limitations on facet size. In order to achieve accurate results for very small separation distances, Isaacson and Cheung (1988a) proposed a two-dimensional model for two cylinders of arbitrary shape on the basis of potential flow theory and the boundary-integral method. The results showed that the added mass coefficients and convective force coefficients varied significantly when the separation distance was less than twice the diameter of the drifting cylinder. This model was unsatisfactory when flow separation has a significant effect on the interactions due to the limitations of the potential flow theory.

Gagnon (2004) investigated the hydrodynamic interaction between floating glacial ice masses and a transiting tanker through physical model experiments. Figure 1 shows the small sphere and tanker during a typical test. In contrast to the case where an iceberg was moving in current toward a larger fixed structure and considerable reductions of impact velocity occur due to hydrodynamic effects (Isaacson and McTaggart 1990a), the experimental data indicated that the magnitude of surge speed and sway speed were very low and would not significantly reduce impact speed in the

Fluid Effects During Iceberg-Structure Interactions, Fig. 1

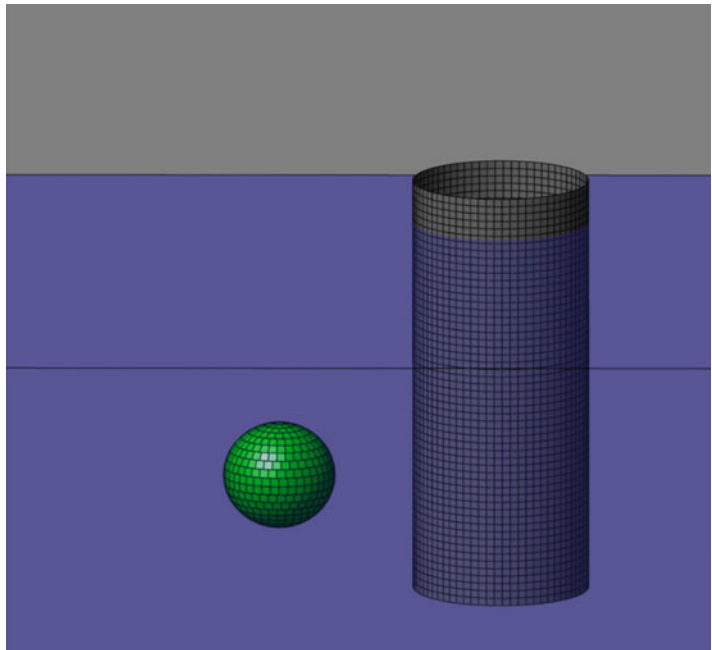
A photograph of a typical test using the small spherical ice mass model (Gagnon 2004)



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Fluid Effects During Iceberg-Structure Interactions, Fig. 2

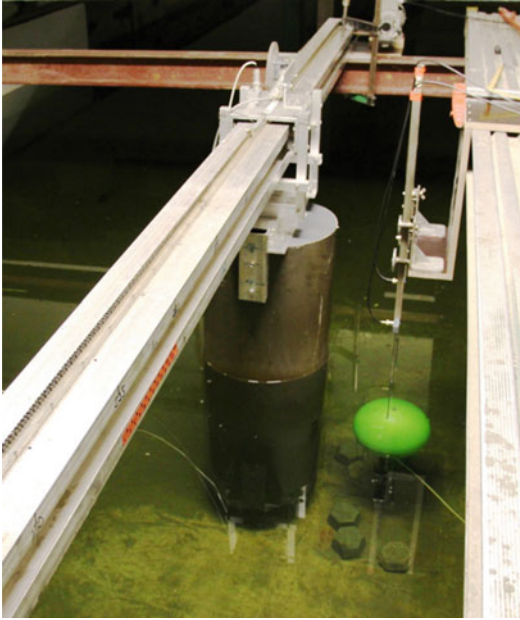
The setup of the numerical model (Tsarau et al. 2014)



scenarios tested. In addition, the results showed that waves tended to increase the degree of sway values. Due to the degree of complexity of the hydrodynamic interaction, numerical method would be more applicable than analytical modeling.

Tsarau et al. (2014) proposed a numerical model to analyze the hydrodynamic interactions between a floating structure and fragmented sea ice, in which rigid-body dynamics and fluid flow around the structure and ice were included (see

Fig. 2). In order to validate this numerical model, a number of physical model tests were conducted (see Fig. 3). It was found that the model was applicable for estimating the hydrodynamic forces on ice upstream of a structure, but it was unsatisfactory when the potential flow assumptions fail. Besides, the experimental data showed that the hydrodynamic forces on the submerged ice model that occur in response to the approaching floater were rather complicated and change values between positive (ice pushing) and negative



Fluid Effects During Iceberg-Structure Interactions, Fig. 3 The setup of the experimental model (Tsarau et al. 2014)

(suction). It was also found that the magnitudes of the forces had a linear relationship with the square of the transit speed as long as the ice was in front of the structure and the motion was steady.

Lee and Nguyen (2011) studied the bow shoulder and side collision responses between a liquefied natural gas carrier (LNGC) and a cubic iceberg using the fluid-structure interaction (FSI) analysis technique of LS-DYNA code. Several collision scenario types were considered, such as moving forward of LNGC in the cases of no detection or no evasion of iceberg, evasion of iceberg by LNGC, and iceberg in the stationary of current condition. The results showed that the iceberg was affected by the wave-making of the LNGC, especially the relative drift velocity of the iceberg that was induced by the pressure from squeezing water layer entrapped between the iceberg and the LNGC. It was also found the significance of bank effect in the flow between very close floating bodies.

Ahmed et al. (2016) have investigated the most relevant parameters affecting the multi-body hydrodynamic behavior of level ice and a single

floater. It was observed that a near-field ice floe significantly effects on the hydrodynamic RAOs of the floater. The results indicated that the most important parameters were wave direction, ice floe distance, and ice floe orientation, and the most enhanced responses were the sway and pitch.

Contact Phase

The contact phase concerns the impact energy and the impact force which are heavily dependent on the impact velocity and the hydrodynamic coefficients. The hydrodynamic forces acting on the model can be classified as drag, buoyancy, and added mass forces. An important parameter in collision is the added mass of ice and floating structure.

A number of studies have used constant added mass (CAM) terms to take hydrodynamic effects into account in analyses of ice-structure or ship-ship interactions. Minorsky (1959) has proposed a method, which assumes that the hydrodynamic effects are represented by an increased inertia force due to added mass effect. He used a constant value of 0.4 for the sway added mass coefficient of the struck ship. Swamidas et al. (1986) conducted experimental and numerical studies to investigate a semisubmersible and a GBS subjected to iceberg and bergy bit impact. The added mass coefficient of the bergy bit was assumed as 0.5, and this value has been normally used for the impacting ice feature (Cammaert and Tsinker 1981; Johnson and Nevel 1985; Matskevitch 1997; Liu and Amdahl 2010). In the study of Isaacson and McTaggart (1990b), it was concluded that the infinite frequency added mass can be used for collisions of short duration, and zero frequency added mass can instead be used as a conservative estimation of the effective added mass for longer duration collisions.

However, this method with assumption of constant added mass neglects the effects of (1) presence and motion of the other body during the approach phase and the collision process and (2) free-surface wave generation. The first effects can be relevant for the relative motion of the ice

and the structure, and the second ones will lead to time-varying wetted surfaces of the two bodies during the impact. This may have consequences on the accuracy of the fluid-ice-structure interaction depending on the time scale of the impact event and the involved geometric and kinematic conditions.

In reality, the added mass coefficient depends on the acceleration and the oscillation frequency and possibly on the proximity of a neighboring body. The acceleration in term depends on the collision force level and the temporal variation and duration. It has been shown in several studies that the added mass varies significantly during the approach and the contact.

Motora et al. (1971) conducted a series of model tests and concluded that Minorsky's assumption of constant added mass is only reasonable when the duration of the collision was very short. For collisions with longer durations, the value of the added mass increases and can reach a value that is equal to or even greater than the ship's own mass. Isaacson and Cheung (1988b) investigated the added mass of an ice mass and found that it varied dramatically as the separation distance between the ice and the large offshore structure decreased. For instance, the added mass coefficient of the ice increased from its far-field value of 1.0 to a value of 1.6 at contact, in which the structure diameter was twice the iceberg diameter and both of them were assumed as circular cylinders.

To get more accurate impact energy and impact load, several researchers have turned to use advanced fluid-structure interaction (FSI) methods such as computational fluid dynamics (CFD), arbitrary Lagrangian-Eulerian (ALE) method, smoothed-particle hydrodynamics (SPH), and other simplified fluid dynamical simulation methods, in which the hydrodynamic effects are directly estimated within the solution algorithm.

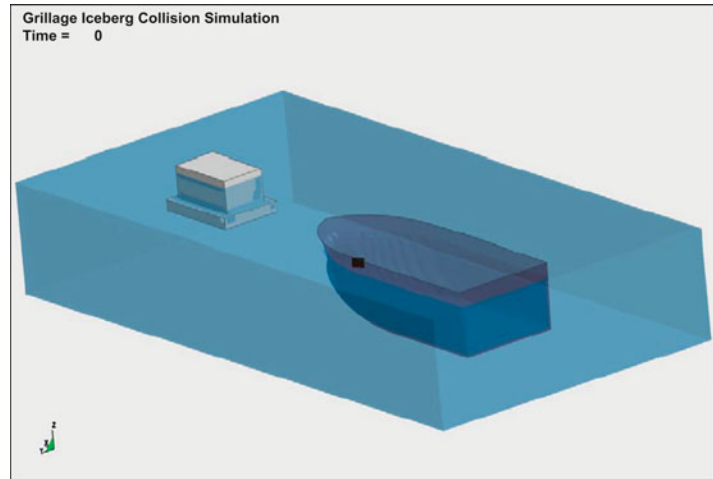
Chander et al. (2015) carried out experimental and numerical studies to investigate the mechanics of ice loads caused by submerged ice blocks colliding. In the numerical study, the CFD model was used to include the hydrodynamic effects. The displacement and velocity of the numerical simulations agreed well with the experimental data. Different hydrodynamic forces acting during

the collision were investigated. The results indicated that added mass plays an important role during the underwater ice collision. The added mass increased suddenly prior to impact and reached the maximum at the time of impact. Tsarau et al. (2015) developed an approach to analyzing ice motions on upstream and downstream of the structure, in which a potential flow model was employed to predict the ice motion upstream of the structure, and a vortex element method was applied to the other situation. The results indicated the importance of including hydrodynamic models in simulation for analyzing ice-structure interactions.

Currently, the ALE method is most frequently used to analyze ship-ship and ice-structure interactions in which the FSI is explicitly considered. To solve the water-structure interaction problem, a Lagrangian formulation is adopted for the structural materials, and an ALE formulation is adopted for the water. In addition, with both Lagrangian and ALE formulations, a contact type algorithm is used to handle the coupling between the water and the structure's materials. Several research articles have presented results of FSI-based simulations that use LS-DYNA's ALE formulation. Wang and Derradji-Aouat (2010) calculated hydrodynamic loads and ship-ice interaction loads based on the FSI method, in which the wave maker simulation showed the good FSI results with regard to wavelength comparison. However, the ship and the ice were treated as rigid bodies in the collision model, which decreases the reality and accuracy with respect to prediction of structural damage. Gagnon and Derradji-Aouat (2006) conducted an ALE simulation of a ship colliding with bergy bits and got good agreement with the experiment with regard to the sway motion. Gagnon and Wang (2012) conducted numerical simulations of a collision between a loaded tanker and a bergy bit, in which hydrodynamics through LS-DYNA's ALE formulation and a validated crushable foam ice model were incorporated. Figure 4 shows the ship-iceberg collision model. It was observed that the movement of the bergy bit due to the vessel's bow wave prior to contact with the ice was realistic. Loads obtained by the simulation

Fluid Effects During Iceberg-Structure Interactions, Fig. 4

Full view of collision simulation components (Gagnon and Wang 2012)



compared reasonably well with measurements from the lab tests. All of these works were conducted in calm water. There is a lack of knowledge in wave effects on ice-structure interactions. In the near future, this problem may get more attentions due to the increasing interest in shipping in the Arctic.

Conclusions

When ice masses impact against fixed or floating structures, pronounced hydrodynamic forces are induced and cannot be ignored in the analysis of ice-structure interactions. It is clear that the hydrodynamic forces have significant effects on the impact velocity and the added mass of the impact bodies or impact trajectory during the approach and the collision. Thus, the hydrodynamic effects should be taken into account when estimating the collision response with regard to collision energy and impact load for ice-structure interactions. Although several studies have focused on the hydrodynamic effects in the far-field, the near-field, and the contact phase, the understanding of wave effects during ice-structure interactions, changing added mass coefficients, reduction of impact velocity, fluid boundary layer effects, viscosity effects, damping effects, and wetted surface effects, is still incomplete. Due to these limitations, further studies in this area through analytical, numerical, and experimental methods are needed.

Cross-References

- [Numerical Simulation Floe Ice–Sloping Structure Interactions](#)
- [Offshore Structure Design Under Ice Loads](#)
- [Ship–Iceberg Interactions](#)
- [Wave–Ice Interactions](#)

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Fluid Mechanics of Net Structure

- [Net Structures: Hydrodynamics](#)

Flux Copper Backing (FCB)

- [Block Assembly Line](#)

Footing

- [Bearing Capacity of Spudcans](#)

Force Oscillation Experiment

- [Aquaculture Structures: Experimental Techniques](#)

Form Resistance

- [AUV/ROV/HOV Resistance](#)

Forming Technology of Profiles

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Definition

Stiffeners, such as longitudinals and frames, are important structural members to support the hull plates in ship structures. They are mainly fabricated by flat, angle, bulb flat and T-shaped bars, etc. Among them, the bending work of frames is

the largest, so this section takes the bending process of frames as an example to discuss.

Introduction

Frame Bending Method and Its Classification

There are many methods to bend the frame, and the typical methods are listed as follows.

(1) Hot Bending by Hand

The heat bending procedure is a widely used technique to obtain curvature in steel frame and plates for shipbuilding purposes. It is a process that relied on highly skilled personnel and did not guarantee constant accuracy of shapes formed. Heat is applied in a line to the web surface of a frame by a flame torch, and then immediate cooling is obtained by air or water. Heat bending can be more time-consuming than using presses.

(2) Cool Bending with Equipment

Its principle is that load is applied on the frame by equipment, such as frame bender. When the load exceeds the yield strength of the material, the frame will be plastic deformed. By controlling the plastic deformation, the profile can obtain the desired shape.

The common equipment in shipbuilding production is cold frame bender, and some enterprises use profile straightening machine to bend the frame.

(3) Hot Bending with Equipment

In production, the frames are preheated and then bent by equipment. Sometimes, heat treatment is carried out simultaneously according to the characteristics of the frame materials. This method is often used in the processing of frame of special material.

The bending methods are usually related to the following factors: temperature, feeding mode, stress characteristics of frame in the process, etc.

(1) Temperature

According to whether the frame is preheated or not in frame processing, it can be divided into cold bending and hot bending.

Cold bending is the processing of bending frame at normal temperature. Hot bending process is to heat the profile above the recrystallization temperature of the frame material, and then apply external force to bend it into the desired shape.

(2) Frame-Feeding Method

The feeding mode of profile during processing usually includes continuous feeding and step-by-step feeding. Continuous feeding method is more suitable for parts whose curvature is constant, such as parts with circle shape.

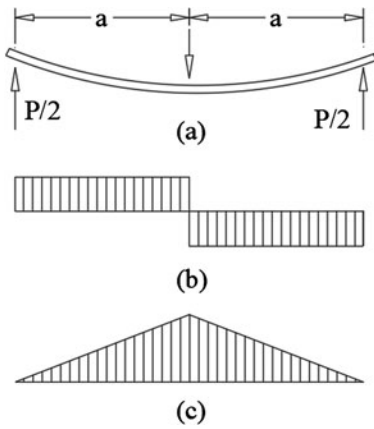
(3) Stress State of Frame in Bending Process

According to the stress state of profile processing, it can be divided into three categories, including tension-bending principle, concentrated force bending principle, and pure bending principle.

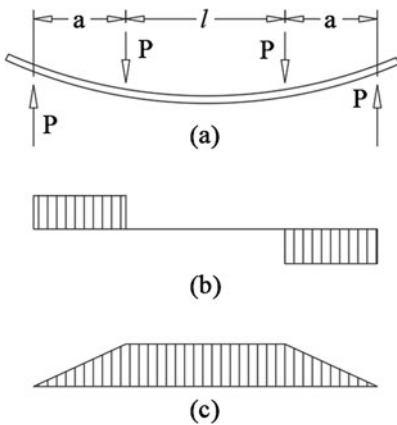
The cold frame bender based on the principle of tension-bending principle is relatively rare. Its main advantage is that there is no compressive stress in the process of bending, so the possibility of wrinkling in the frame is small.

It is a traditional method to bend frames with the principle of concentrated force bending method. At present, most cold frame benders apply this bending principle. Because there are three stress points, it is also called three-fulcrum bending method. As shown in Fig. 1, shear force Q and bending moment M are generated by concentrated load P . At present, most of the cold frame bender used in shipbuilding are designed and manufactured according to the principle of concentrated force bending.

The principle of pure bending method is different from that of concentrated force bending method. The pure bending in mechanics refers to the plane bending state of a section of beam which is only subjected to bending moment but not to shear force. In order to obtain the pure bending condition of the beam, the method of material mechanics is to make the simply supported beam bear two equal concentrated loads symmetrically, and make the beam bending in plane. As shown in Fig. 2, the force acting on the beam under pure bending is shown. The moment at any section of



Forming Technology of Profiles, Fig. 1 Force situation of the method of concentrated force bending method. (a) Force diagram; (b) shear force diagram; (c) bending moment

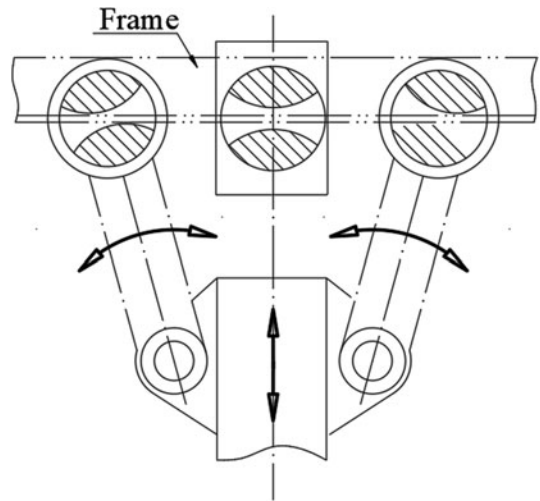


Forming Technology of Profiles, Fig. 2 Force situation of the principle of pure bending method i. (a) Force diagram; (b) shear force diagram; (c) bending moment

the beam between the two concentrated loads is $M=Pa$, while the shear force $Q=0$, so the beam is in pure bending state.

Cold Bending of Frame Parts

At present, curved frame parts in commercial ships are fabricated generally by concentrated force bending principle, and the frames are bent along the longitudinal direction step by step. Following is a brief introduction to the most typical



Forming Technology of Profiles, Fig. 3 Diagram of the working part of the bender

cold frame bender and several technical problems in cold bending process.

Cold Frame Bender with Three Fulcrum

Figure 3 shows a diagram of the working part of the bender. In the bender, the central and two side molds are used as clampers in the bending process. In the working process, the central mold moves forward or backward so the load is applied on the frame, and both side clampers revolve around their respective axes. The frames are bent along the longitudinal direction step by step, i.e., the work is moved intermittently past the clamp which is held in place and actuated by a press. During the process, the central mold and the frame web are in the state of “clip and not tight” to avoid the buckling for concave shape forming. The side clampers clip the frame web tightly when the load is applied. This kind of bender has many advantages and is widely used.

Several Technical Problems in Cold Bending Process

Detection and Control of Shaping

In order to bend the frame to the web edge in accordance with the required curvature, the cold frame bender needs to be checked and measured along the forming line repeatedly in the process, so a certain

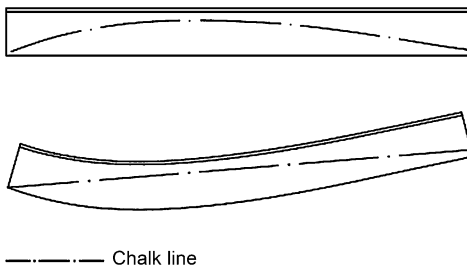
method of detecting and controlling the forming process is needed. For cold frame bender without NC control function, there are several methods to control the bending process.

(1) Artificial Sampling with Iron Samples (or Templates)

In the process of frame bending, the iron sample or template must be used repeatedly to check the web edge shape of the bended frame to see if it is consistent with the profile of the iron sample or template. This method not only needs to make iron samples or templates and labor-consuming material, but also is quite complicated and requires high technical level of workers.

(2) Inverse Curve

Obtaining the correct frame curvature can be achieved by the “inverse curve” method. The inverse line in frame bending refers to a special line, as shown in Fig. 4. The inverse curve is drawn in chalk on the web of straight frame, and the frame is bent until the curve becomes straight on the curved frame. In this way, in the process of frame bending, the formation of frame can be controlled by continuously detecting whether the curve is straight or not. Accurate calculation of this curve on the straight frame before bending is the key to control frame bending by this method. Now in shipbuilding, the inverse curve information can be determined for each frame by the loftsman using a CAD/CAM system.



Forming Technology of Profiles, Fig. 4 Diagram of the inverse curve bending principle. (a) Chalk line on the straight frame; (b) chalk line becomes straight on the curved frame

For the frames with longer length or larger curvature, it is sometimes necessary to divide the profile into several parts along the length, each part drawing an inverse line separately, and each inverse line should intersect a certain length. This bending control method is mainly suitable for frame with high web and small curvature radius.

The main advantage of this method is that it can save the material and time to make the template, but it needs to calculate and draw the inverse line, and the detection of frame processing is also complicated.

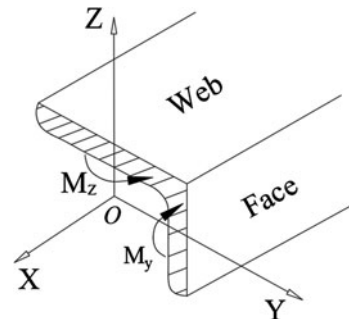
Lateral Deformation and Its Elimination Measures

The lateral deformation refers to the deformation in the direction of web thickness. As shown in Fig. 5, because the section of the frame (angle steel or bulb) is not symmetrical to the XOY plane, when the frame is bent by the moment M_z in the XOY plane, moments of stresses to Y-axis are unbalanced in the cross section; thus, another bending moment M_y on the XOZ plane is generated, which will cause the frame to deform on the XOZ plane (i.e., in the direction of web thickness), which is called lateral deformation.

In order to produce structural parts that meet the quality requirements, methods must be adopted to eliminate lateral deformation in the process of bending.

Springback and Its Treatment in Frame Cold Bending

Plastic deformation occurs in the cold bending process of frame, and it is accompanied by elastic



Forming Technology of Profiles, Fig. 5 Causes of lateral deformation

deformation. The recovery of elastic deformation after removal of external forces is called springback, which is the inherent characteristic of elastic-plastic materials. Springback has a negative impact on the cold bending process. How to deal with this factor correctly is very important to improve the efficiency and quality of cold bending of frames.

There are two ways to deal with it. One is to calculate the springback value in advance by theoretical calculation, and then superimpose it on the bending control quantity to bend the frame excessively, so that the frame can get the correct shape after springback. However, due to the inconsistency of material properties and the approximate simplification of mechanical model in calculation, the calculation results will be different from the actual situation, which makes the shape of frame formed difficult to achieve the desired results.

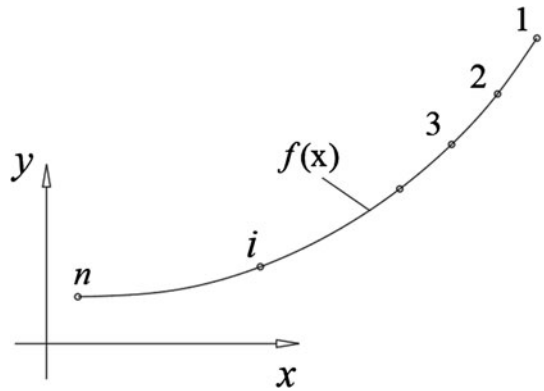
Another method is to measure the actual springback value after each bending action, and then bend the frame excessively according to the measure data. After repeating several times, the shape of the frame bended meets the design requirements.

Control Method of NC Cold Frame Bending

At present, the cold frame bender used in ship-building has realized numerical control (Li et al. 2008), and the control of frame bending is no longer dependent on iron sample, template, or inverse line. The main methods of forming control are endpoint measurement and chord measurement.

Principle of the Forming Control Based on Measurement of Locus of End Point

The key of this method (Shama and Miller 1967) is to control the end point of the frame and make it move in a specific trajectory in the bending process, so as to bend the straight frame into the part with the desired shape. The trajectory is determined by the shape of frame part uniquely.



Forming Technology of Profiles, Fig. 6 Diagram of series of points on the shape of the frame

As shown in Fig. 6, the shape of the frame will be specified as a series of coordinates at discrete intervals on the frame flange according to feed length. Ideally, these coordinates, together with the correct slope and curvature at each point, should be calculated exactly.

The position of the end point with respect to point C, at any bending operation (see Fig. 7), will be called the locus of the end point. Point C corresponds to the midspan of the current processing frame section. The end point position could be defined by the distance X along the tangent at point C and the distance Y along the normal from the end point. In bending process, as long as the X and Y of each processing section are controlled so that they can meet the required values, the straight frame can be bent into the required shape of the frame parts.

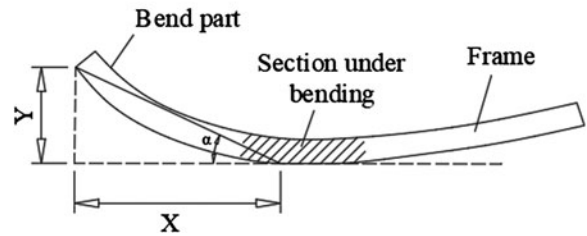
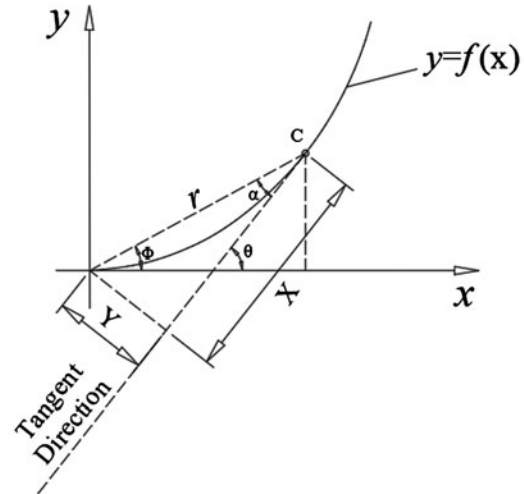
Principle of the Bowstring Measuring Method

The control model, called “bowstring measuring method” (Wang and Xu 1988, Li et al. 2008) that was originated by WHUT (Wuhan University of Technology), has been successfully applied in full-scale CNC frame benders.

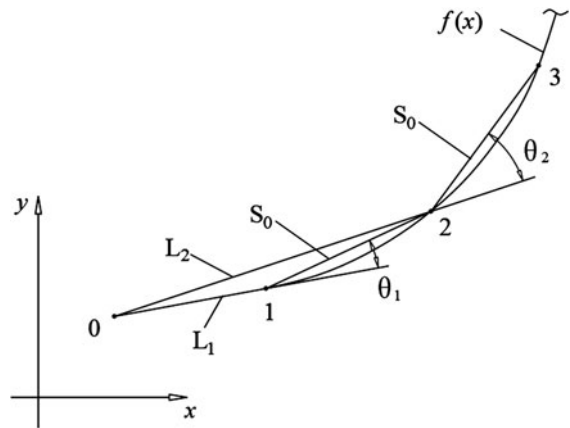
Principle of Bend Control

As shown in Fig. 8, the shape of the frame will be specified as a series of points $P_i (i = 1, 2, \dots, n)$ at discrete intervals on the frame flange according to feed length. Point P_0 is a reference point for forming control in process. In this method, P_0 is on the reverse direction along the tangent to the

Forming Technology of Profiles, Fig. 7 Diagram of principle of measurement of locus of end point



Forming Technology of Profiles, Fig. 8 Diagram of principle of bowstring measuring method



frame at the first point P_i of the shape line of the frame part, and the distance between P_0 and P_i is L_0 which is determined by the structural dimension of the bender. The frame section between P_i and P_{i+1} is the current section in bending. The control principle is to use three elements of triangle $P_0P_iP_{i+1}$ as variables to control the profile forming. As shown in Fig. 8, the segment, which

will be formed between point 2 and point 3, is illustrated. Point 2 and point 3 are the two vertices of the triangle.

The first side length, L_i , is the distance between two vertices P_0 and P_i . The second length, S_0 , is the value of the distance between the two vertices P_i and P_{i+1} . The angle, θ_i , is that between the radial line and the bowstring, in respect to the

current forming section. L_i is used as the control parameter for feeding, and θ_i is used for shape-forming control. L_i and θ_i can be calculated with the formula as follows:

$$L_i = \sqrt{(x_0 - x_i)^2 + (y_0 - y_i)^2} \quad (1)$$

$$\theta_i = \cos^{-1} \frac{L_{i+1}^2 - L_i^2 - S_o^2}{2L_i S_o}$$

where (x_0, y_0) and (x_i, y_i) are the coordinate of point 0 and point i.

Control Method of Springback

The results of theoretical calculation by E.Н.Мошнин's formula for springback show that there is linear relationship between the bending curvature and curvature of springback. Although the calculation result was approximate, the change tendency of springback showed by the result is close to the right law. The control model of stepwise approximation control principle is proposed based on the rules (Wang 1984, Li et al. 2009). Stepwise approximation control is actually a trial and error method.

In the working condition of frame bending, the curvature of the frame shape formed is difficult to measure. So the amount of angle is measured and used as the control parameter in the control model of bowstring measuring method. When the trial

and error method was applied, the amount of angle of spring-backing in the bending process is added to the forming control parameter for the next forming operation for the same frame section. So, iterative operations are needed to achieve the correct shape. Of course the iterative operation will affect the efficiency of frame forming.

In the iterative operation, some times of the forming operation are needed for each working section of frame (see Fig. 9). In the first operation, the control parameter is θ_1 , and the amount of springback is T_1 after the load is released. The next bending operation is needed to be applied on the same working section because of the springback. And the amount of springback T_{j-1} ($j = 2, 3, 4, \dots$) is added to θ_1 for the next forming operation. By this iterative way, bending is applied until an acceptable shape is achieved. The control angle in the iterative process is:

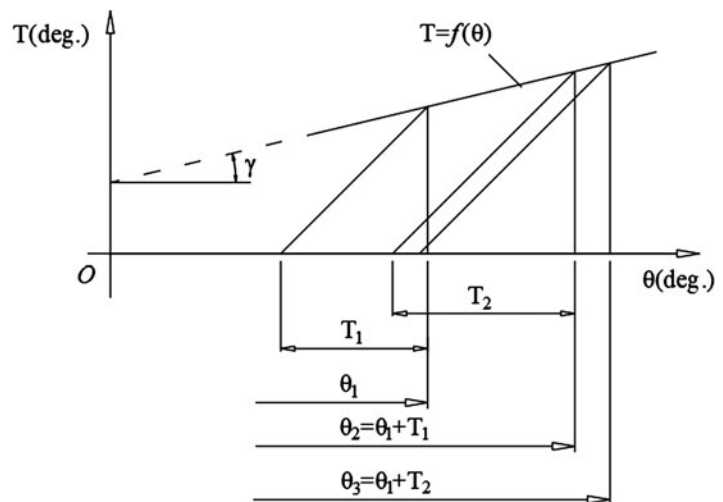
$$\theta_j = \theta_1 + T_{j-1} \quad (2)$$

After the load is released in the first operation, the difference between the plastic deformation and the control parameter is the amount of springback, which is:

$$\Delta\theta_1 = T_1 \quad (3)$$

In each of the next bending operation, the difference between the plastic forming and

Forming Technology of Profiles, Fig. 9 Diagram of control model of trial and error method



the control parameter is $\Delta\theta_j$ for each operation $\Delta\theta_j$ is

$$\begin{aligned}\Delta\theta_j &= (\theta_j - T_j) - \theta_1 \\ &= (\theta_1 + T_{j-1} - T_j) - \theta_1 \\ &= T_{j-1} - T_j\end{aligned}\quad (4)$$

When $|\Delta\theta_j| < \varepsilon$, an acceptable shape is achieved for the working portion. ε is the allowable deviation.

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Forming Technology of Steel Hull Plates

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Synonyms

Ishikawajima-Harima Heavy Industries (IHI); Numerical control (NC)

Definition

The steel hull plates for ship building are categorized into single curved plates and complex curved plates, among which the single curved plates are bent only in one direction, with a simple shape and a small bending degree, while the complex curved plates have a large bending degree and a complex line, including sail-shaped plates (Fig. 1), saddle-shaped plates, etc (Huang, 2013).

The forming technique of hull plate components mainly involves mechanical cold bending and line heating bending. Generally, single curved plates are processed through mechanical cold bending. Complex double curved plates are usually mechanically processed to form the curve in one direction by cold bending, and then processed to form the curve in the other direction by line heating. Besides, a few shipyards also process double curved plates through cold bending by hydraulic press machines or numerical-controlled bending press machines (Ji, 2005).

Introduction

Roll Bending Forming

Roll bending is the most commonly used method for plate cold bending, and the main equipment of roll bending includes three-roll bending machines (Fig. 2), four-roll bending machines (Fig. 3), etc., among which the three-roll bending machine is

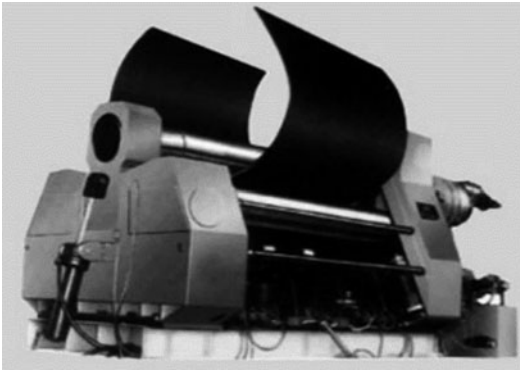


Forming Technology of Steel Hull Plates, Fig. 1 Sail-shaped plate

the most widely used. The single curved plates (e.g., bilge strakes in parallel middle body) among the outer hull plates are usually formed by three-roll bending machines.



Forming Technology of Steel Hull Plates, Fig. 2 Three-roll bending machine

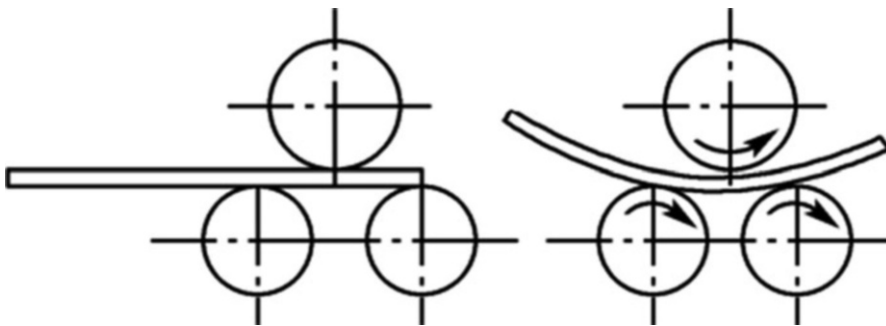


Forming Technology of Steel Hull Plates, Fig. 3 Four-roll bending machine

The lower shaft roller of ordinary three-roll plate bending machines is an active roller, installed in a fixed bearing and driven by an electric motor through a speed reducer to rotate, while the upper shaft roller is a driven roller, which can be adjusted up and down. When the shaft roller rotates, the plate is bent into a certain curved shape between the three rotating shaft rollers, as shown in Fig. 4.

The three-roll plate bending machine is categorized into open type and closed type. The frame on one end of the upper shaft roller of the open three-roll plate bending machine can be disassembled so as to take out the enclosed cylindrical component after bending, while the frame of the closed three-roll plate bending machine cannot be disassembled, so the enclosed cylindrical plate cannot be bent with it.

In order to improve the performance of bending plates, a three-roll bending machine, the three rollers of which are respectively adjustable up and down, could also be used and is suitable for the forming of simple curved plates. Moreover, a three-roll plate bending machine, the shaft roller of which is adjustable side to side, could be used to make closed cylindrical barrel and cone. For the four-roll plate bending machine, its two side rollers could form two pairs of asymmetric three-core rollers with the upper and lower rollers respectively, which can not only minimize the remaining right-angle sides of the workpieces, thus skipping the prebending procedure, but also eliminate the need to turn the work-piece around. The four-roll plate bending machine is of excellent performance and high efficiency but has complex structure, complex control, large weight, and high cost (Liu et al. 2011).



Forming Technology of Steel Hull Plates, Fig. 4 Working diagram of three-roll plate bending machine

In addition, the three-dimensional curved surface flexible roll forming technology is a new plate roll bending forming technology developed in recent years. Its basic principle is as follows: First, the required curve is composed of several short rollers with adjustable heights, which can rotate and swing on their own; second, the three-dimensional curved surface continuous local plastic forming is realized through the pressure of the short rollers and the friction between the short rollers and the plate.

According to the different shapes of the workpieces, the position of the short rollers can be adjusted to form the curvature required for forming the workpieces, so as to realize multipurposes by one machine. Flexible roll forming technology is a combination of multipoint forming technology and traditional roll bending forming technology, which has the characteristics of high efficiency, low cost, flexibility, and continuity. At present, the research on this technology gradually becomes mature, and the experimental device has been developed, as shown in Fig. 5.

Buckling Forming

Buckling forming is another plate bending method based mainly on cold processing. The main equipment used in this method is the

hydraulic press machine and the multiple-use plate bending machine, the latter of which is used by some shipyards. In recent years, with the development of numerical control technology, numerical-controlled bending press has already been developed (Xu, 2005).

Hydraulic Press Machine

Hydraulic press machine is a cold bending equipment which bends and forms the plate using the liquid pressure. The machine uses liquid as the working medium and achieves a variety of processes according to its mechanism of energy transmission based on the principle of Pascal. The hydraulic press machines are generally composed of the machine (main engine), the power system, and the hydraulic control system. According to different liquid media used, there are oil hydraulic press and water hydraulic press, among which the oil hydraulic press is more widely used. The pressure of the oil hydraulic press is produced by the oil pump, and since there is generally no pressure storage device, the oil hydraulic press is relatively light and simple structured, while the water hydraulic press is usually equipped with pressure storage device, so it can produce greater pressure, but is huge and complex structured.

When the hydraulic press machine is performing buckling, the compression module must be installed on the press head to ensure the shape of the plate. The compression module can be divided into general type and special type according to its application scope. The general compression module can bend plates with different surface shapes. If the number of components with the same shape is large, the special compression module could be produced.

Hydraulic press machines can not only bend hull components of different shapes, but also perform plate flanging, angle pressing, prebending, leveling, etc.

Compared with the traditional technique, this hydraulic forming process has obvious technical and economic advantages in reducing weight, reducing the number of parts and molding dies, and reducing production costs and has been

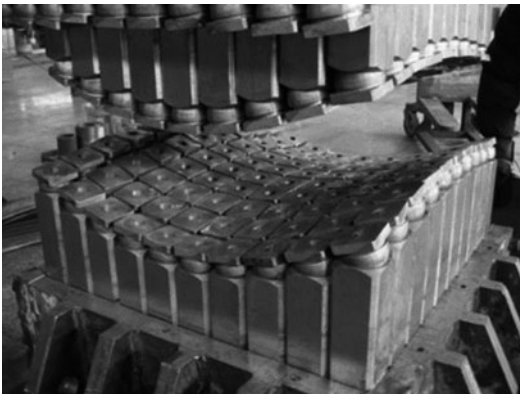


Forming Technology of Steel Hull Plates,
Fig. 5 Experimental device of 3D curved plate flexible roll forming

more and more widely applied in the industrial field.

Numerical-Controlled Bending Press

The application of numerical control (NC) automatic processing technology to plate forming can not only greatly improve the production efficiency of plate bending and reduce labor intensity, but also improve the accuracy of bending, thereby reducing the workload of assembly and rectification. The use of this technology with other NC processing devices could realize the comprehensive mechanization and automation of hull processing workshop, and improve the comprehensive production capacity. The research on numerical-controlled bending press has been paid great attention to both in China and abroad for many years. Figure 6 shows the prototype of the multihead 3D numerical-controlled bending press, jointly developed by Wuhan University of Technology and Shandong Shuoli Machinery Manufacturing Co., Ltd. Before bending, the press heads of the former block (or force plug) are automatically adjusted one by one by NC program, so as to change the heights of them and form a surface of the same shape as the required component (springback within consideration). When the plate to be bent is well positioned, the heads of the upper molds (or lower molds) fall (or rise), and the plate is bent into various shapes required (Qiyang et al. 2017).



Forming Technology of Steel Hull Plates,
Fig. 6 Multihead 3D numerical-controlled bending press

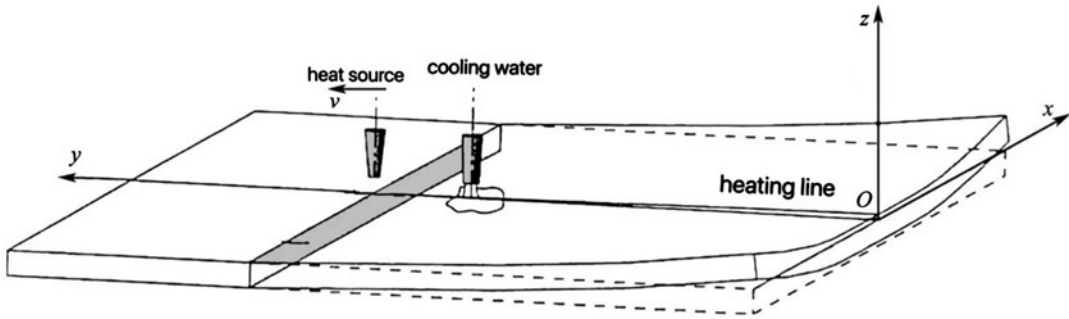
Line Heating Bending

The line heating bending is currently the most widely used double curved shell plate processing technology in shipyards at home and abroad. More than 90% of the hull shell plates with complicated curvature can be processed by this technique (Song et al. 2016).

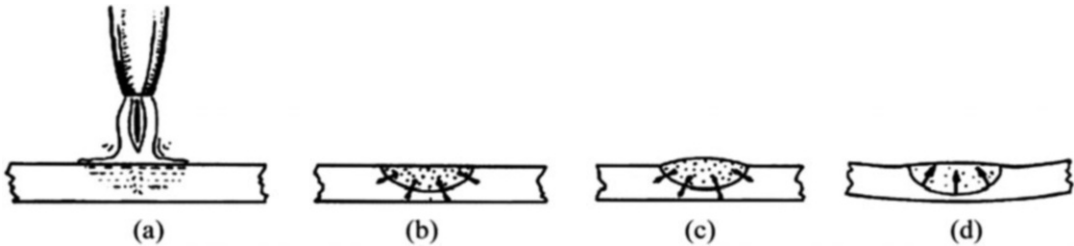
The line heating bending is a thermal forming process, which uses a heat source (such as gas flame, high-frequency inductance, laser heat source, etc.) to move along a predetermined heating line at a uniform or varying speed on the surface of the metal plate, and to perform the local line heating. As shown in Fig. 7, a certain temperature distribution is generated on the surface and the thickness direction of the plate. A certain part of the metal plate is heated and expanded and is hindered by the surrounding cold metal, so the plastic deformation is generated in the process of such mutual interaction. Then water is used to cool the metal in a certain distance behind the heat source. The plastic deformation could cause the heated and expanded metal to begin to shrink and stretch when facing coldness, thus resulting in local deformation of the metal plate, as shown in Fig. 8. This local deformation is due to the angular deformation on the surface and along the thickness of the plate, which is the basis for the forming of the whole metal plate (Li et al. 2006).

Experienced workers often estimate (based on their previous experience) the required shrinkage needed for each part (often divided by the frame space), by observing the overall shape of the curved plate (taking the template guideline as a reference), and then further consider the way to arrange the heating lines and determine other process parameters. At the same time, during processing, it is also necessary for them to continuously examine the differences between the generated shape and the target shape, and timely modify the layout of the heating lines and other processing parameters.

The line heating bending process is a complex thermo-elastic-plastic deformation process, which is affected by many factors. The key to



Forming Technology of Steel Hull Plates, Fig. 7 Schematic diagram of line heating bending process



Forming Technology of Steel Hull Plates, Fig. 8 Schematic diagram of line heating bending deformation

the application research of line heating bending is to find out the influence law of various factors that influence steel plate forming effect during the line heating process (Zhang et al. 2011). First, it is necessary to determine the main process parameters of line heating deformation. The influencing parameters, according to their influence weight, can be generally divided into two categories: The first category is the most important influencing factors, such as the layout of the heating lines, heating speed, heating line length, heating line distance, heat source, distance from the nozzle to the plate surface, the initial shape of the plate (length, width, thickness, curvature, etc.) and initial stress, cooling rate, steel plate support mode (free ends, supported ends, and supported four corners), etc.; the second category is the secondary important influencing factors, such as air temperature, pressure and flow rate of oxygen, pressure and flow rate of acetylene, water flow rate, and the texture of the steel plate. The influence of main process parameters on forming effect generally includes the following:

The Influence of Heating Line Layout on Forming Effect

During the line heating bending, the correct layout of the heating wire is directly related to the correct deformation of the steel plate. The line heating at different positions on the same steel plate will cause completely different forming shapes. Most hull shell plates can be divided into sail-shaped plates and saddle-shaped plates, and the forming of both plates involves first forming the transverse curve through cold bending (due to its large curvature) and then forming the longitudinal curve through line heating bending. There are also differences in forming the two plates. Because the direction of the longitudinal curve of the sail-shaped plate is the same as that of the transverse curve (i.e., both are bent toward one side of the plate), the heating lines are positioned on both sides of the cross section of the plate (Fig. 9). During its bending, the method of line heating closing is used to shorten the longitudinal edges on both sides of the component to obtain the longitudinal strength. However, because the direction of the longitudinal strength of the



Forming Technology of Steel Hull Plates, Fig. 9 Sail-shaped plate and heating line layout



Forming Technology of Steel Hull Plates, Fig. 10 Saddle-shaped plate and heating line layout

saddle-shaped plate is opposite to its transverse strength (i.e., the transverse curve is bent toward one side of the plate, while the longitudinal curve is bent toward the other side), the heating lines are positioned in the middle of the cross section of the plate (Fig. 10). During bending, the middle part of the back of the component should be heated to shorten the middle part longitudinally to obtain the longitudinal strength.

The Influence of Heating Speed on Forming Effect

With all the other processing conditions remaining the same, the lower the heating speed is, the greater the deformation of the steel plate near the heating line will be. It should be noted that the lower the heating speed is, the higher the surface temperature of the steel plate will be. Too high temperature could easily damage the texture of the steel plate, especially that of the thin steel plate, and the plate could be even burnt through. On the contrary, if the heating speed is too high, the surface temperature of the steel plate will be

low, and it is often unable to deform the steel plate, especially the thick steel plate which requires a higher temperature. Therefore, steel plates of different thickness require different heating speed. Generally, for a thin steel plate, a higher heating speed could form a greater deformation, while for a thick steel plate, only a lower heating speed could form a certain deformation (Bae et al. 2013).

The Influence of Heating Line Length on Forming Effect

Generally, the longer the heating line is, the greater the deformation of the steel plate will be. In the real processing, it should be noted that the length of the heating line is also related to the plate width. For the sail-shaped plate, the heating lines are arranged on the edge of the plate, the length of which should not surpass the neutral axis of the cross section, so as to guarantee the forming, and the line could not be too short, either. The heating lines of the saddle-shaped plate are arranged in the middle of the plate, and multiple heating lines could be arranged on the same flame path, and the length of the heating lines should be moderate.

The Influence of Steel Plate Thickness on Forming Effect

With all the other heating conditions remaining the same, the thicker the steel plate is, the more difficult it will be to form steel plate deformation, the greater will be the energy required for deformation of steel plate, and the smaller will be the deformation generated near the heating line (Zayed et al. 2018).

The Influence of Heating Sequence on Forming Effect

For the saddle-shaped plate, the heating lines are arranged in the middle of the plate, and multiple heating lines are often arranged on the same flame path. The heating sequence of these heating lines has a certain effect on the local shrinkage. However, for the sail-shaped plate, such problems do not exist. The influence of the heating sequence on the deformation of the steel plate is mainly manifested in the obvious distinction between

deformations generated by the initial heating line and those by the other lines on the same flame path. With all the other heating conditions remaining the same, the deformation generated by the initial heating line is smaller than those by the other heating lines (Iranmanesh et al. 2014).

In a word, the line heating bending technique is one of the most widely used bending techniques in shipyards. In addition, the line heating bending technique can also be used to level and straighten the welding deformation, such as T-beam welding deformation, plate frame welding deformation, etc. which is generally known as the rectification by flame heating and water cooling, with its principle, influencing factors and process parameters similar to the line heating plate bending.

However, the line heating bending process also has serious defects. First, there are many factors that could influence the forming, thus causing the forming laws difficult to master. At present, the forming mainly relies on manual operation by workers based on experience, and it is difficult to achieve mechanization and automation. Second, its low efficiency could not meet the needs of modern shipbuilding. Therefore, numerical control line heating bending is being studied at home and abroad, in order to achieve the automation of plate forming, improve labor productivity, reduce labor intensity, and completely reform the manual operation. Figure 11 shows the line heating automation system IHI- α , jointly developed by Ishikawajima-Harima Heavy Industries (IHI) and the Welding Research Institute of Osaka University. It is an automatic line heating forming system for the hull shell plates using high frequency heating as the heat source. Fig. 12 shows the prototype of the large and complex curved surface steel plate line heating forming robot, successfully developed through the cooperation of Dalian University of Technology with Dalian Shipbuilding Industry co., ltd. The robot could be integrated with the process parameter prediction software system to achieve the automatic forming of sail-shaped plates and saddle-shaped plates as hull shell plates.



Forming Technology of Steel Hull Plates, Fig. 11 IHI- α system of Japan



Forming Technology of Steel Hull Plates, Fig. 12 The prototype of the large and complex curved surface steel plate line heating forming robot

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Foundation

► Bearing Capacity of Spudcans

FPSO

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Synonyms

FPSO (floating production storage and offloading); SPM (single-point mooring)

Definition

A *floating production storage and offloading (FPSO)* unit is a floating vessel used by the offshore oil and gas industry for the production and processing of hydrocarbons and for the storage of oil. A *FPSO* vessel is designed to receive hydrocarbons produced by itself or from nearby platforms or subsea template, process them, and store oil until it can be offloaded onto a tanker or, less frequently, transported through a pipeline. *FPSOs* are preferred in frontier offshore regions as they are easy to install and do not require a local pipeline infrastructure to export oil. *FPSOs* can be a conversion of an oil tanker or can be a vessel built specially for the application. A vessel used only to store oil (without processing it) is referred to as a floating storage and offloading (*FSO*)

vessel (https://en.wikipedia.org/wiki/Floating_production_storage_and_offloading).

Introduction

A floating production storage and offloading (FPSO) is a floating facility, usually equipped with hydrocarbon processing equipment for separation and treatment of crude oil, water, and gases, arriving on board from subsea oil wells through flexible pipelines. FPSOs have storage capacity for the treated crude oil produced and are equipped with the offloading system to transfer the crude oil to shuttle tankers for shipment to refineries.

FPSOs are also called as floating oil processing factories on seas. Their displacements cover a range from 50 thousand to 300 thousand tons, with the deck area as large as 2–3 football fields. FPSOs can be a conversion of an oil tanker or can be a vessel built specially for the application, as Fig. 1. One of the advantages of FPSOs is that they are particularly effective in remote or deepwater locations, where seabed pipelines are not cost-effective. FPSOs eliminate the need to lay expensive long-distance pipelines from the processing facility to an onshore terminal. Once the field is depleted, the FPSO can be flexibly moved to a new location.

FPSOs have been widely used in oil exploitation around the world with a concentration in West Africa and South America, as shown in Fig. 2 (OilMooc 2011). As of August 2015, there are more than 277 floating production units in use around the world, of which 62% are FPSOs (“Available FPU count at all-time high,” 2013).

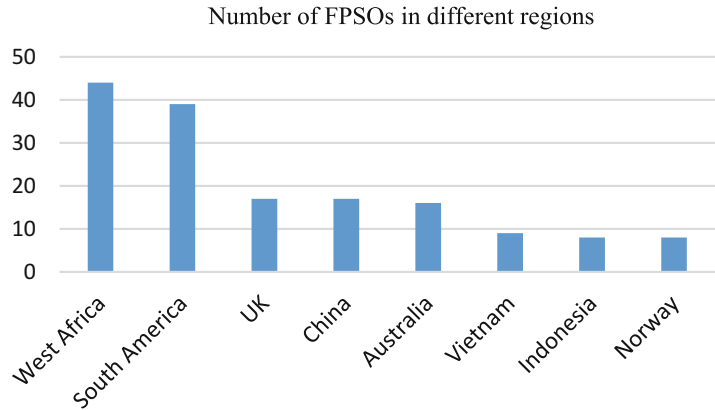
History

The first oil FPSO was the *Shell Castellon*, built in Spain in 1977 (Mohamed A. El-Reedy, 2012). Today, over 270 vessels are deployed worldwide as oil FPSOs, as shown in Fig. 3 (Offshore Magazine 2016). For the production ability, most FPSOs are rated for 7000 bbl/d to 250,000 bbl/d, with a storage of 24,000 to 2,200,000 bbl. They can operate with up to 163 subsea wells and 80 risers. Now, the FPSO operating in the deepest

FPSO, Fig. 1 Illustration of a FPSO in operation



FPSO, Fig. 2 Distribution of FPSOs worldwide

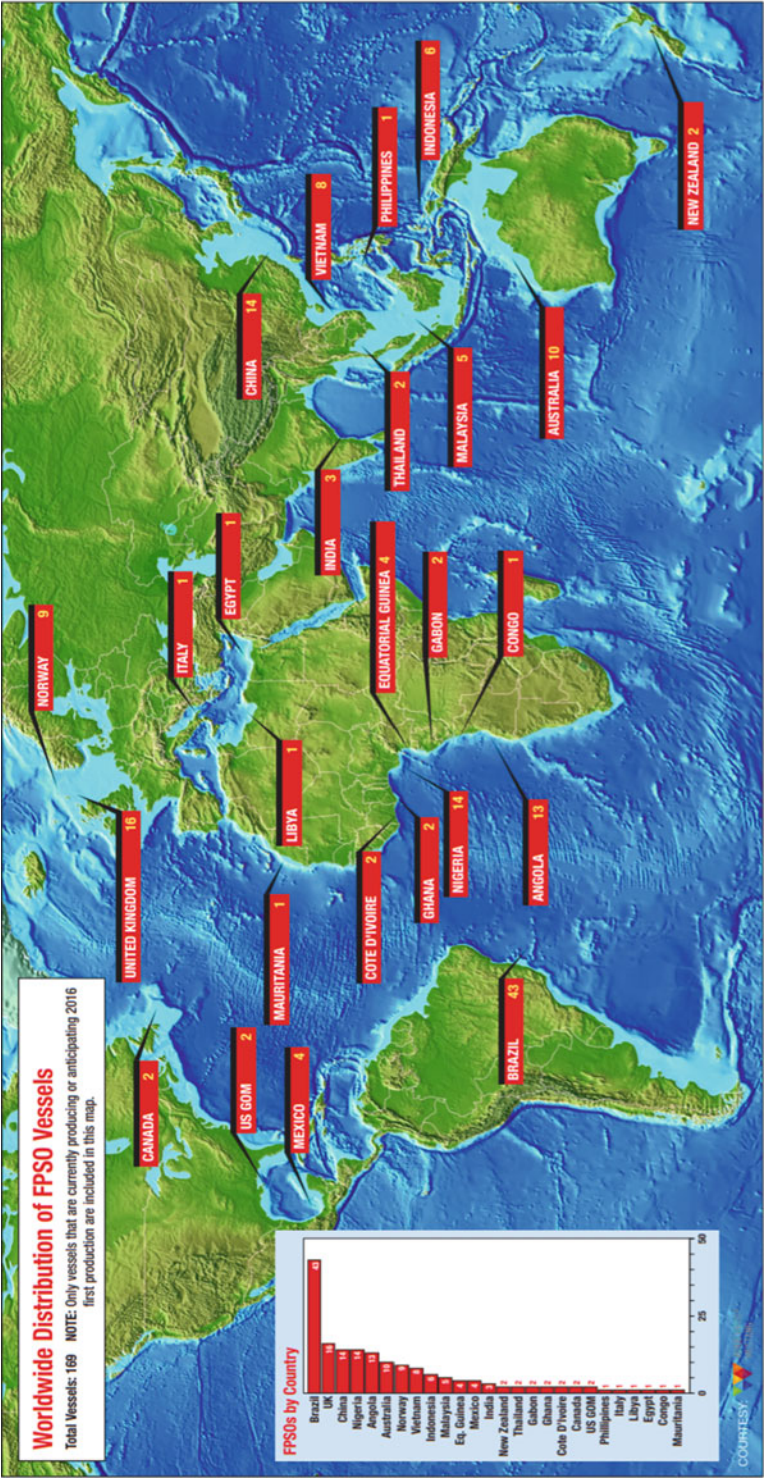


waters is the FPSO *Turritella* designed by SBM, which is owned by Shell and moored at a depth of 2896 m in the US Gulf of Mexico (First Oil for *Turritella* 2016).

The development of FPSO worldwide generally has experienced four periods:

1. 1976–1985 was the early stage of development, during which the single-point mooring system enabled FPSOs to be an effective offshore oil and gas exploitation facility.
2. 1986–1994 was the growth period of this technique, during which FPSOs were widely adopted in stranded oil fields worldwide, with an increase of more than two vessels per year on average.
3. 1995–1998 was the period of expansion, during which the number of FPSOs exploded rapidly, with an increase of more than eight vessels per year on average.
4. 1999–now is the flourishing period, during which some outstanding breakthroughs have been achieved to increase the operation water depth from several hundred meters to over one thousand meters. Especially after 2002, different facilities of offshore oil and gas exploitation have been developed in multiple forms, including LNGFPSO, FPDSON, and LPGFPSO.

Over the 40 years, the first 10 years witnessed the birth of this novel concept and its industrial application. In the following 20 years, this



FPSO, Fig. 3 Distribution of FPSOs worldwide (by 2015, cited from Offshore Magazine)

technique experienced a steady development and wide application. In the recent 10 years, the number of FPSO increased sharply, with a broad application in different fields of engineering.

FPSO Structure

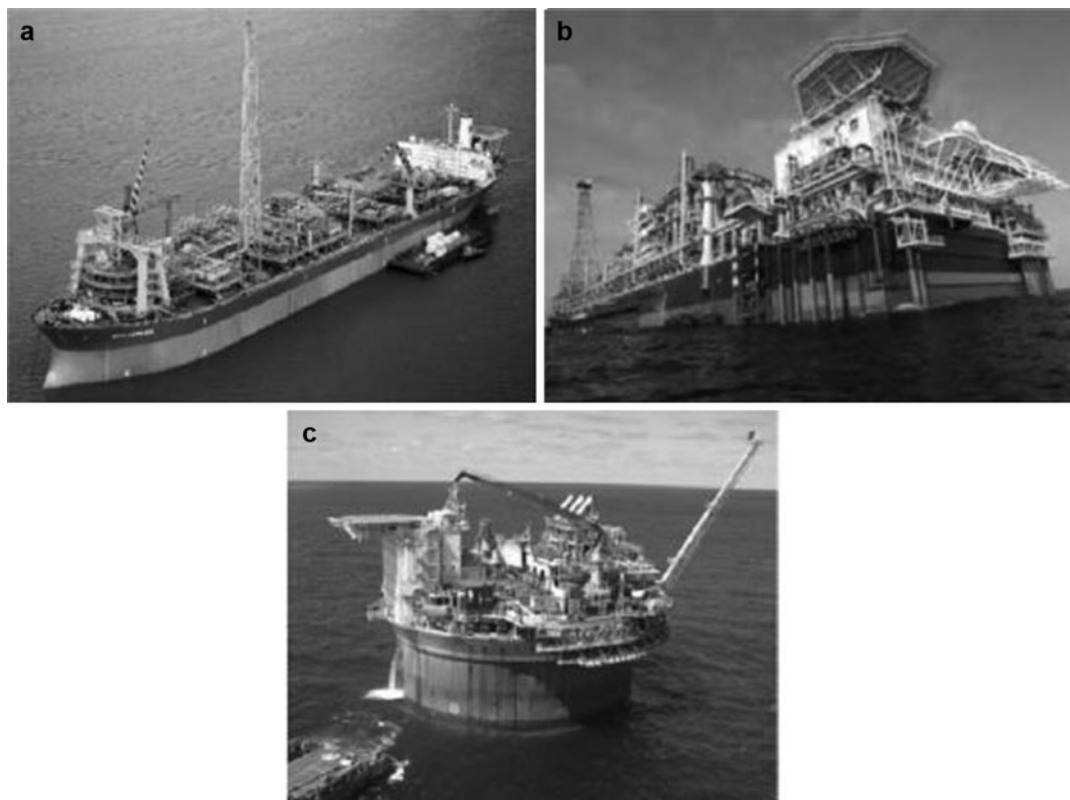
Normally, a FPSO system consists of five components, including the hull, topside modulus, turret, mooring system, and turret.

FPSOs can be divided into three types according to their hull shapes, which are ship-type, multi-body-type, and cylinder-type, as shown in Fig. 4. At present, ship-type FPSOs are the most dominating type used in exploitation that were either newly built or converted from available oil tankers (Fig. 5).

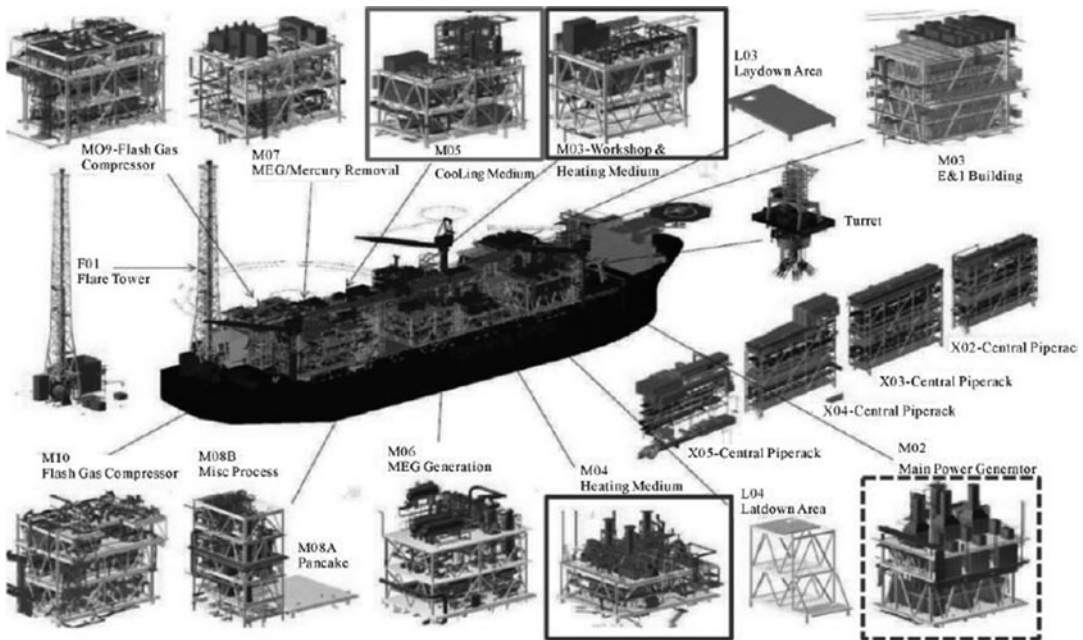
Topside modulus is the core part of FPSOs. Compared with conventional oil tankers, FPSOs'

topsides include some additional modulus, such as production and process module, flare tower, special offloading system, and helicopter platform. The appropriate arrangement of all the modulus in the limited area is of great significance. The following picture shows the layout of the topsides of a FPSO.

Mooring systems are used to constrain the motion of large FPSOs in open seas, through connecting the subsea anchor piles and the vessels with mooring lines made of steel or polyester fiber. There are normally two types of mooring systems used prevalingly, which are single-point mooring and spreading mooring. Single-point mooring system is employed by most ship-type FPSOs in the world due to its weather vane function which can alleviate the environmental loads effectively. Spreading mooring system is employed mostly on cylinder-type FPSO and those ship-type FPSOs in some special ocean



FPSO, Fig. 4 Three types of FPSO. (a) Ship-type. (b) Multi-body-type. (c) Cylinder-type



FPSO, Fig. 5 Topside structure of FPSO

areas whose environmental loads are kept at low-level or have dominating directions all over the year, for instance, Western Africa and oceans near equator.

One of the most important and cost parts of FPSO system is the turret. Turret is a compact and multifunctional structure that integrates the mooring lines, risers, and FPSO hull. FPSO's turret has two forms as inner and external turret. Internal turret enables the hull protecting the turret well, so newly built FPSOs often adopt internal turret design. On the contrary, external turret has no hull protection so the wave slamming problem has to be taken into consideration, but external turret design is often used for converted FPSO because external turret can be added directly at the vessel bow (Zhao et al. 2013).

FPSO Hydrodynamic Performances

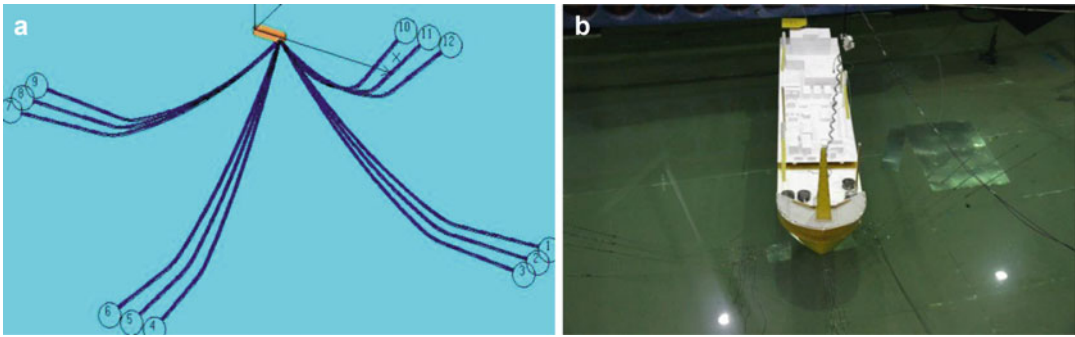
To study the hydrodynamic performances of FPSO systems coupled with mooring and riser systems, both numerical and experimental methods have been adopted by the engineers

(Fig. 6). In addition, model tests in wave basin are also implemented to validate and modify the numerical models.

Numerically, the hydrodynamic information of FPSO can be predicted based on the three-dimensional potential flow theory. The added mass coefficients, radiation damping coefficients, and the first-order response amplitude operators (RAO) of the wave forces and motion responses can be calculated in frequency domain. Further, time domain analysis is applied to calculate the integrated six-degree of freedom motion (Fig. 7) of the FPSO coupled with its mooring system.

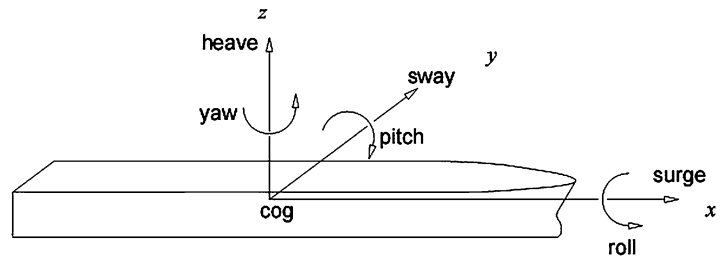
FPSOs of ship-type have special hydrodynamic performances due to the coupling effect between hull shape and mooring system. Firstly, strong vertical plane motion would be induced, namely, the heave, roll, and pitch motions, due to the superposition of their natural frequencies with the wave-dominating frequencies. In addition, FPSOs with single-point mooring (SPM) might suffer from large wave slamming loads on bows because of the weather vane effects (Fig. 8).

Secondly, FPSOs also experiences considerable horizontal plane motions, such as surge,



FPSO, Fig. 6 Topside structure of FPSO. (a) Numerical simulation. (b) Model test

FPSO, Fig. 7 Six-degree of freedom ship motion



FPSO, Fig. 8 Weather vane effect of SPM



sway, and yaw. Apart from the first-order motions under irregular waves, slow drift motions on the horizontal plane are also highly concerned because long natural periods of the mooring system could give rise to the motion resonances induced by second-order wave forces. Hence, the safety of mooring lines needs to consider second-order wave forces.

Thirdly, the ship-shaped FPSOs have small roll damping. Thus, strong roll motions would be simulated under oblique waves which will inflict

discomfort of the crew members. Equipment-like bilge keel and anti-rolling tanks have been developed to mitigate violent roll motions of FPSOs.

FPSO Mooring System and Risers

As an important part of FPSO system, mooring system serves to prevent vessels being drifted away by low-frequency components of irregular waves and keep the vessel in a certain heading

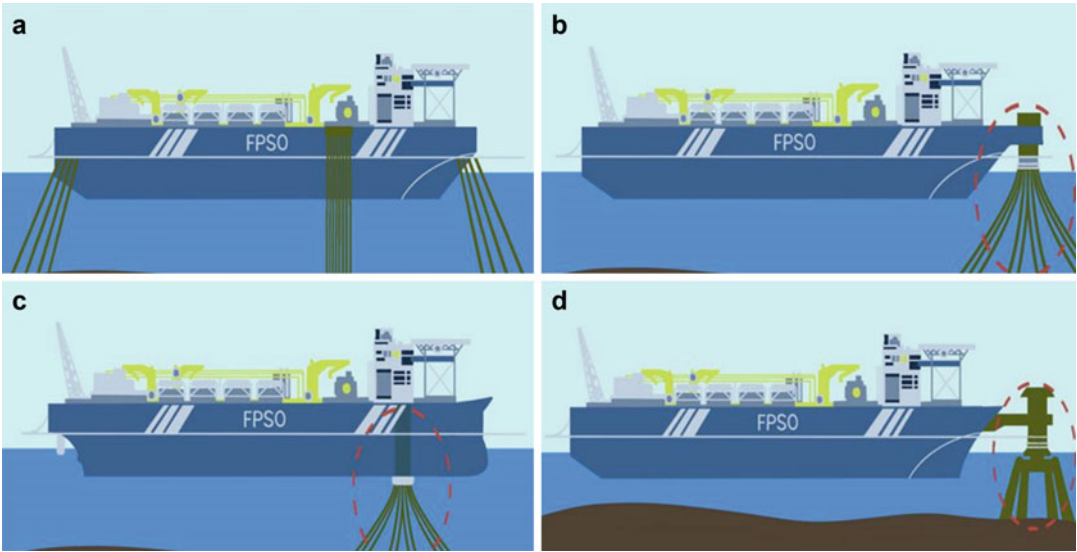
against the dominant environment. Besides, the mooring system can also be combined with a fluid transfer system that enables connection of (subsea) pipelines to the tanker. Generally, mooring systems can be divided into two types—spread mooring system and single-point mooring systems (SPM). SPMs can be further divided as the internal turret, the external turret, and the soft yoke wishbone configuration. Figure 9 demonstrates all four kinds of mooring systems.

Risers are important parts of FPSO systems that connect floaters and subsea production

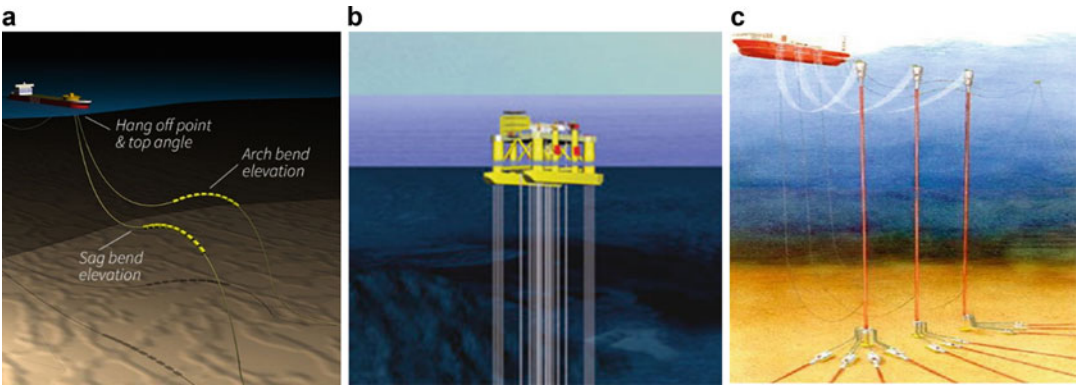
units. As a kind of ship-type floating structure, FPSOs have poor hydrodynamic performances in heave and pitch modes due to the large water-plane area, which brings great challenges to the stiffness and fatigue analysis of risers attached. Besides, the large tension of upper part of risers connected with the floating terminal calls for a higher design standard of FPSO hull structures.

According to their structures and applications, three kinds of deepwater risers (see Fig. 10) are quite applicable for FPSOs, namely, steel catenary

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FPSO, Fig. 9 Illustration of different kinds of mooring systems. (a) Spread mooring system. (b) External turret SPM system. (c) Internal turret SPM system. (d) Soft yoke wishbone SPM tower system



FPSO, Fig. 10 Deepwater riser systems. (a) SCR. (b) TTR. (c) HTR

riser (SCR), top tension riser (TTR), and hybrid tower riser (HTR).

Green Water Effect

FPSOs with single-point mooring (SPM) system is vulnerable to huge slamming waves on ship bows due to the weather vane effect, causing large amounts of water shipped onto the deck. This phenomenon is defined as green water (see Fig. 11) that will cause violent impact and destruction of important equipment on the bow area of FPSO, such as internal turret, risers, and pipelines. Thus, the consideration of green water events on FPSO is of great significance to ship structures, hydrodynamic performances, and production safety during the process of design.

Green water event is a problem of fluid-structure interaction between highly nonlinear waves and FPSO structures. The difficulty of prediction mainly focuses on the strong nonlinearity of wave fields and couplings between FPSO and waves.



FPSO, Fig. 11 An event of green water on deck

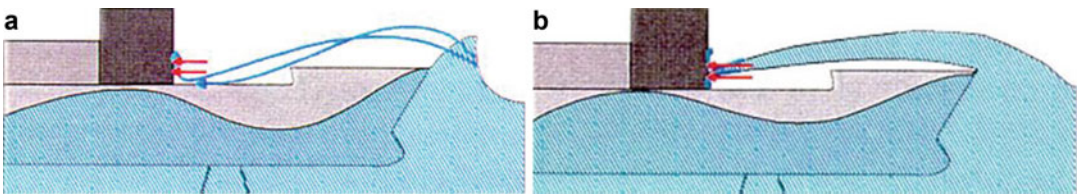
Early studies on green water are mainly based on model tests. Some pioneer studies were widely conducted on various aspects such as freeboard level, bow shape, and wave steepness. Two types of green water on deck were observed and categorized in tests (see Fig. 12). For practical purpose, some simplified methods were proposed afterward. For modeling green water on deck, flooding wave theory and dam-breaking model were applied. Shallow water theory was employed to represent water propagation on deck.

Recently, some high-accuracy methods have been developed to simulate the green water event based on computational fluid dynamic (CFD) methods. Many schemes, like VOF and level-set, have been used to capture the strong motions of nonlinear free surfaces and, thus, highly improve the prediction accuracy of wave loads on structures.

Tandem Offloading

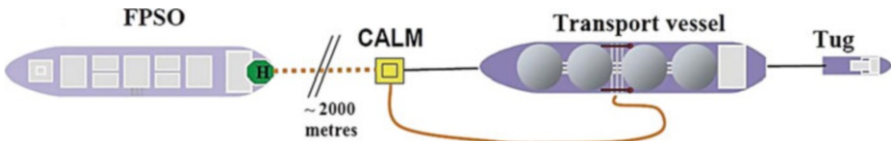
The types of FPSO offloading system are basically as follows: pipeline offloading, tanker transportation, and offloading alongshore. Among all, the tanker transportation is mostly used in FPSO offloading. Oil is directly offloaded from FPSO to shuttle tanker or barge which transports the oil to shore. Tandem offloading is one of the classical and effective ways to transfer the oil from FPSO vessels to transport vessels (Fig. 13).

Tandem offloading operation is carried out in such a way that the transport vessel is located in the sheltered region behind FPSO connected by hawsers (at a distance of about 80 meters), while a side-by-side offloading operation is carried out in parallel positions. Compared with side-by-side

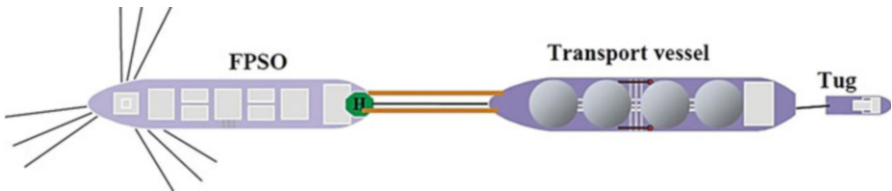


FPSO, Fig. 12 Two types of green water on deck. (a) Dam-breaking-type green water. (b) Wave-plunging-type green water

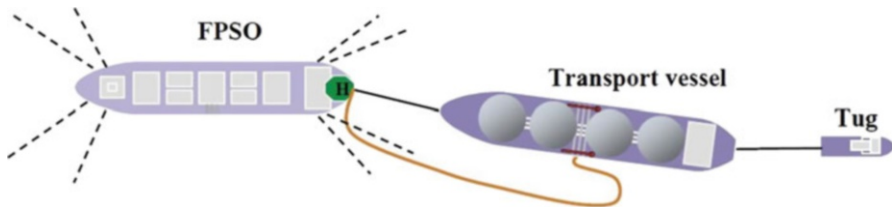
FPSO, Fig. 13 Tandem offloading



FPSO, Fig. 14 Single-point-moored (SPM) FPSO tandem offloading



FPSO, Fig. 15 Turret-moored FPSO tandem offloading



FPSO, Fig. 16 Spread-moored FPSO tandem offloading

mooring, tandem mooring can withstand more rigorous environmental conditions and is more advantageous in terms of fast disconnection. In addition, dynamic positioning (DP) can be applied to transport vessel in tandem offloading,

to eliminate the fatigue risks of mechanical restriction between two floaters.

For FPSOs of different mooring systems, three corresponding offloading scenarios are considered in Figs. 14, 15, and 16, where the black

lines are mooring lines and the brown lines are oil pipes.

The first one is the single-point mooring (SPM) offloading from a catenary anchor leg mooring (CALM) buoy or similar facility. The transport vessel can freely weather vane and assume a heading of “least” resistance relative to the prevailing weather. Beside the existing SPM systems in shallow water, they are nowadays planned as offloading facility in the neighborhood of FPSOs for deepwater fields.

The second one is the tandem offloading from a turret-moored FPSO. The transport vessel can still weather vane, but its behavior is influenced by and coupled to that of the FPSO. In benign and moderate environments, the transport vessel is typically a vessel and assisted by one or more tugs. Depending on their relative loading conditions and possible support by stern thrust or tugs, they will assume headings with respect to the environment. Besides the passive mooring with a bow hawser, DP tandem offloading is used at a number of locations especially in harsh environments.

The third one is tandem offloading from a spread-moored FPSO. The transport vessel is typically a vessel of convenience and assisted by one or several tugs. Such operations take place, e.g., offshore West Africa, where the prevailing environmental conditions are relatively benign compared to those of North Sea, the Gulf of Mexico, and South China Sea, but often characterized by

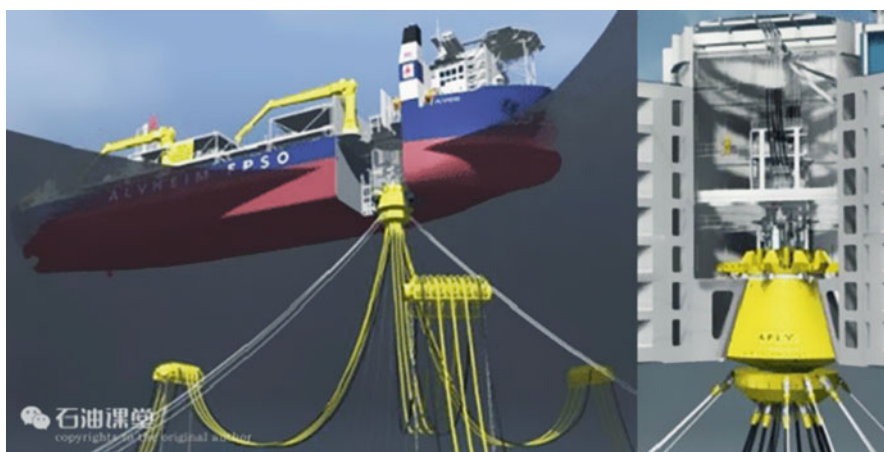
non-collinear swell, wind, local waves, and current conditions. Due to the “fixed” heading of the spread moored FPSO, the vessels do not both have the ability to assume freely a heading with respect to the environment. The behavior of the transport vessel is also heavily influenced by the presence of the FPSO (shielding, etc.).

Internal Turret

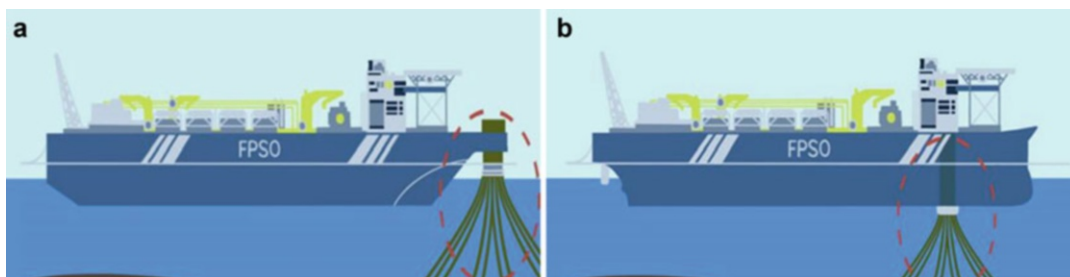
The mooring system of FPSO is an essential part of offshore exploration for providing station keeping of floating systems. With the offshore oil production moving into the deep sea, turret mooring systems (Fig. 17), as a single-point mooring configuration, have been the major facilities for the positioning of floating production unit in open seas.

The turret mooring system consists of a turret assembly that is integrated into a vessel and permanently fixed to the seabed by means of a mooring system. The turret system contains a bearing system that allows the vessel to rotate around the fixed geostatic part of the turret, which is attached to the mooring system.

The turret mooring system can also be combined with a fluid transfer system that enables connection of (subsea) pipelines to a FPSO (Bluewater 2019). The fluid transfer system includes risers between the pipeline end manifold



FPSO, Fig. 17 Turret mooring system (inner turret)



FPSO, Fig. 18 Illustration of two types of turret mooring systems. (a) External turret mooring system. (b) Internal turret mooring system

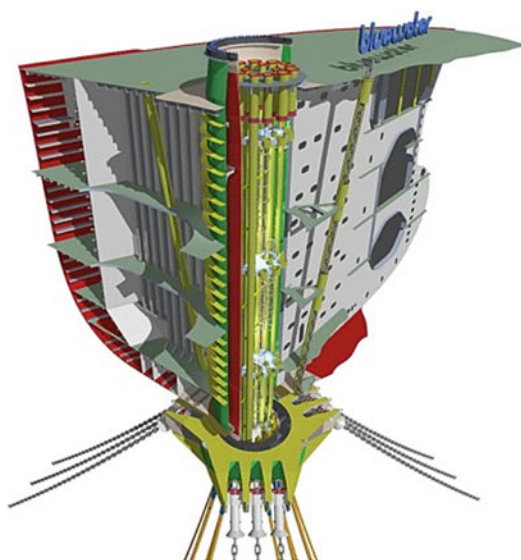
at the seabed and the geostatic part of the turret. In the turret a swivel provides the fluid transfer path between the geostatic part and the free weather vane FPSO that rotates around the turret. The turret system is fully passive and does not require active vessel heading control or active rotation systems in the turret. The turret systems can be divided into two types, which are located externally or internally with respect to the vessel hull structure (see Fig. 18).

Structure of the Internal Turret

The internal turret system is integrated into the hull structure at the bow of the vessel. The internal turret is a slender structure that is connected to the vessel structure via a large-diameter 3-race roller main bearing at the top and via a large-diameter sliding bearing at the bottom.

The slender turret shaft ensures that the horizontal loads from the risers and mooring lines are transferred via the sliding bearing to the vessel, while the remaining forces from the turret and the vertical loads from the risers and mooring lines are transferred via the 3-race roller bearing. The supports of the bearings are integrated into the turret casing, which is part of the modified vessel hull structure. Typically, the turret casing is a cylindrical structure that is integrated between the bottom and the deck of the vessel and transfers the turret loads via the vessel web frames to the vessel hull structures.

The turntable decks are positioned on top of the 3-race roller bearing. The turntables include



FPSO, Fig. 19 Structure of internal turret

the process facilities for commingling fluids from the risers using manifolds, as well as pigging facilities or chemical injection facilities to assure the flow in the risers. In addition, the turntables provide space for subsea control facilities, to enable safe operation of the subsea manifolds and wellheads. The highest deck level of the turntables provides space for the swivel stack, which transfers fluid and electrical power to the turret access structure that is located on the vessel main deck. This access structure also provides support access to the different turntable decks. Figure 19 shows the cross section of internal turret.

Main Characteristics of Internal Turret

The main advantages of the internal turret mooring production storage system are:

1. Compared with other types of single-point mooring systems, the internal turret mooring system is suitable for deeper water and more complex sea states.
2. The facilities of production, storage, oil loading and turret assembly, etc. are arranged concentratively on one ship. This arrangement makes the system get rid of connection module and become more compact. Hence the environmental forces and motion responses are minimized, and mooring performances are improved and the reliability of the system is raised.
3. Because the turret is located in the interior of the hull body, the hull body can protect the turret and pipelines from damage. It is also convenient for the management and maintenance of the turret assembly in complex sea state.
4. Low cost of installation, convenient maintenance, and flexible movement from one production site to another, almost all of the facilities onboard can be used after being moved to another site.

Nevertheless, some problems of internal turret mooring systems should be also mentioned.

1. The large-diameter opening in the hull body is unfavorable for the general arrangement and ship structure strength. Both the hull structure strength and the hull volume will be sacrificed.
2. The position of the internal turret has some negatively influence the weather vane capability of the single-point moored FPSO, compared to that of external turret.

Yoke Mooring System

FPSO has been widely applied to the world marine petroleum exploitation. Without its own dynamical system, it usually relies on the mooring

system as a positioning device to guarantee the safety of operation in a fixed area. As a kind of single-point mooring (SPM) system, soft yoke mooring system realizes FPSO weather vane effects by the multiple-hinge-point connection mode, and it has been widely applied to oil and gas exploitation in the shallow waters in Bohai Bay in China and the Gulf of Mexico (Lyu et al. 2017).

Compared with traditional internal and external turrets and other single-point mooring systems, the soft yoke single-point mooring system owns obvious technological advantages. It can realize vessel-mooring separation, satisfying a set of the mooring system applicable to multiple ship-type structures. It can realize repeated applications of the mooring system in multiple sea areas, which makes it easy to disassemble the whole mooring and its local components by hinge points. Meanwhile, the above water mooring mode of the soft yoke system avoids the direct contact between components and seawater and the impacts of sea ice in ice regions and reduces the influences of environmental corrosion and lengthens the service life (Fig. 20).

Widely applied in the 1980s, the soft yoke mooring system has drawn wide concern from global oil and gas companies. Multi-hinge soft yoke mooring system is considered as the currently best single-point mooring form and has been applied to oil and gas exploitation in shallow water areas (water depth of tens of meters).

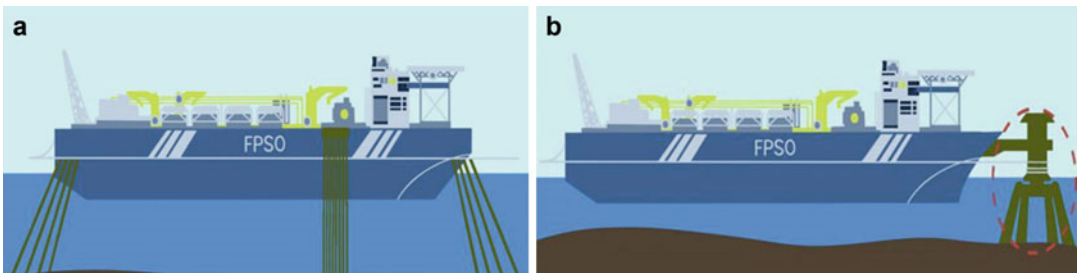
Shallow FPSO Mooring Systems

As an important part of FPSO system, mooring system serves to prevent vessels being drifted away by low-frequency components of irregular waves and keep the vessel in a certain heading against the dominant environment. Generally, mooring systems can be divided into two types—spread mooring system and single-point mooring systems (SPM). SPMs can be further divided as the turret mooring and the soft yoke mooring configuration. Typically, in shallow water, a conventional spread mooring system will be used. In moderate water depth, spread mooring system may prove to be uneconomical and the motions may become unacceptable.

FPSO, Fig. 20 Soft yoke single-point mooring system and FPSO



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FPSO, Fig. 21 Illustration of different kinds of mooring systems. (a) Spread mooring system. (b) Soft yoke SPM tower system

Hence, a system with a fixed structure (jacket) and a yoke arm link to the floating structure (soft yoke mooring system) may be found useful. Figure 21 demonstrates the two kinds of mooring systems.

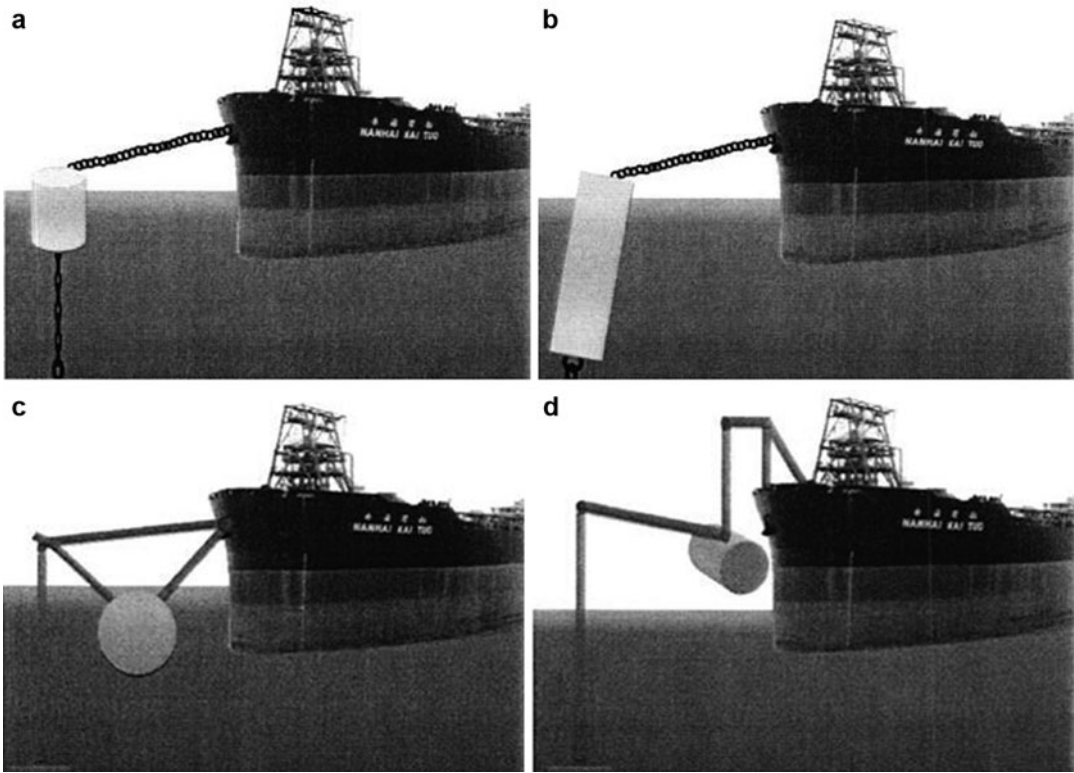
Spread mooring systems are multipoint mooring systems that moor vessels to the seabed using multiple mooring lines. The mooring lines can be directly attached to the hull of the vessel as well as indirectly using buoys on the sea surface. When they are directly attached to the hull, structural modifications are required to provide strong points for the mooring lines.

The soft yoke mooring system includes a rigid steel frame that is connected to the tower by means of hinges and to the vessel by means of a pendulum structure. Typically, the pendulum structure includes two pendulums that provide a restoring force for the oscillating vessel. This restoring force maximizes the range of local environmental conditions in which the SPM system can moor a vessel.

Evolution of Yoke Mooring System

The soft yoke mooring system is developed on the basis of a single anchor leg mooring system (SALM), which is like a large buoy floating on the sea (Fig. 22a), with the lower end on the sea bottom and the upper end connected to the ship bow. This firstly appeared in 1969, and then, the lower part of buoy evolved into a cylinder-type buoyancy tank directly fixed on the bottom of the sea (Fig. 22b). This can reduce the motion radius of the FPSO and prevent the buoy from causing damage to itself and other related equipment when heaving up and down with waves.

After further development, the mooring lines attached to the ship bow gradually evolved into a rigid arm (Fig. 22c), which avoids the huge impact of the mooring line when suddenly straightened. With further improvement, the simple rigid arm evolved into a soft yoke arm (Fig. 22d) that connect the fixed tower and FPSO by means of hinges and a pendulum structure.



FPSO, Fig. 22 Development process of soft yoke

For the final yoke structure, the curve of soft yoke mooring forces to displacement has been completely smoothed to improve hydrodynamic performance of FPSO. In addition to the evolution of structure, the mooring force originally provided by buoyancy tank has changed into that provided by gravity of ballast tank (pendulum structure). Nevertheless, the restoring forces of buoyancy tank fluctuates constantly due to the tank motions under the excitation of waves, while the restoring forces provided by the gravity of the pendulum structure is more stable. Especially when the soft yoke system is completely off the wave surface, it will not be influenced by complex hydrodynamic loads.

Two Kinds of Yoke Mooring Systems

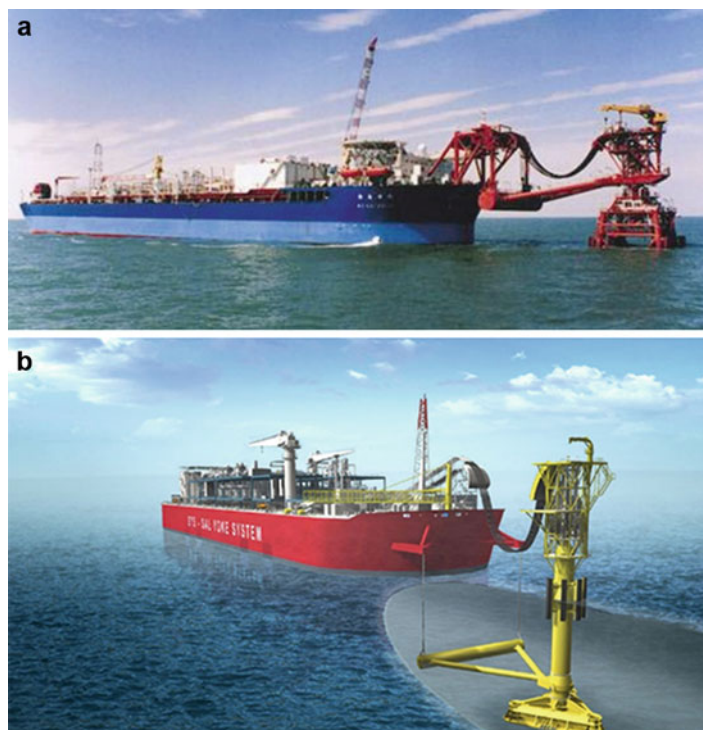
Two types of structures are now applied in yoke mooring systems, namely, the fixed-tower yoke mooring system and the single-column yoke mooring system, as shown in Fig. 23.

As a conventional yoke mooring system, the fixed-tower yoke mooring (Fig. 23a) consists of

several main structures: a truss tower fixed on seabed; a soft yoke that freely revolves around the tower and provides resilience on its own weight, including the A type soft arm structure itself (yoke) and the ballast load (or counterweight); the connection structure of the soft yoke and FPSO, such as the boom or chain; and the supporting structure on FPSO.

The single-column type yoke mooring system (Fig. 23b) also has similar components as mooring tower (column-type), rotational soft yoke, and connection and supporting structures. However, there is different design for the single-column type. Firstly, the yoke is hinged with the single column at the position on seabed, instead of over the free surface, thereby significantly reducing the overturning moment of the fixed structure. Secondly, yoke arms are connected to the main deck of the FPSO, thus avoiding the overhanging support structure above the deck height. This can effectively simplify the steel truss structure for mooring on the bow area. Thirdly, the risers are installed in the hollow single column to protect

FPSO, Fig. 23 Two kinds of yoke mooring systems.
(a) Yoke mooring system connected with fixed tower.
(b) Yoke mooring system connected with single column



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them from wave slamming and ice crushing. The underwater yoke arms also avoid the harsh environmental loads on the surface area.

Shallow Water Effects on FPSOs with Yoke Mooring Systems

Because yoke mooring is more applicable for shallow water areas, thus much attention should be paid to effects of shallow water on FPSO hydrodynamics, which may make the hull collide the seabed and mooring system and then bring about damage.

Shallow water depth has obvious influence on the hydrodynamic characteristics of FPSO. With the decrease of ratio of water depth to draft, the natural periods of FPSO in heave, roll, and pitch modes increase accordingly. This means that when the wave period is larger and wave energy is concentrated in the low-frequency region in shallow waters, the probability of resonance, leading to strong motion responses of FPSO, will increase. Hence, the safety of yoke mooring system is threatened.

For the design of a FPSO soft yoke mooring system, the governing response is undoubtedly

the surge motion and load (Xiao et al. 2007). In shallow waters, especially the ultra-shallow waters, the surge motion and mooring force will be very large, and it will very seriously affect the single-point mooring system of a large FPSO.

With the decrease of the water depth in shallow water, it is found that the low-frequency response of surge motion and load increases enormously, while the high-frequency response keeps almost the same. It indicates that the low-frequency wave drift force on FPSO increases enormously in shallow water. This large amplitude increase of wave drift force may be caused not only by the change of natural period but also by the nonlinearity of the shallow water waves. Much more attention has been paid to the interaction mechanism between nonlinear shallow water waves and FPSO.

Cross-References

► [Drag Anchors](#)

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FPSO (Floating Production Storage and Offloading)

- FPSO
- Tension-Leg platform

FPSO, Floating Production, Storage and Offloading Unit

- Moored Ship in Ice

FPU, Floating Production Unit

- Moored Ship in Ice

Frequency Shift Keying (FSK)

- Underwater Acoustic Communication

Frictional Resistance

- AUV/ROV/HOV Resistance

FSI, Fluid–Structure Interaction

- Numerical Simulation of Ice-Going Ships

Fuel Cell

- Alternative Fuel for Ship Propulsion

Fuel Consumption

- Human Factors in the Role of Energy Efficiency
- Impact of Maritime Transport
- Ship Propulsion System

Fuel Prices

- Technical and Economical Barriers on Green Energy Utilization in Shipping

Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency

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Trend on the Shipping Emission Control Regulations

With the gradual tightening of shipping emission reduction, the future trend on the

shipping emission prevention regulation is as follows.

The Scope of Emission Control Area (ECA) Will Be Further Expanded

The EU has been working for many years to expand the area applicable to the shipping emission control regulations through relevant amendments. Some other countries in the world have reached consensus and form synergy in advancing the establishment or expansion of emission control areas. In 2016, China established three sulfur emission control areas in the Yangtze River Delta, the Pearl River Delta, and the Circle-Bohai-Sea, respectively; see diagram in Fig. 1. From January 1, 2019, the sulfur emission control areas

was extended to the Yangtze River Mainstream, the Pearl River Mainstream, and all coastal waters.

Under the new Domestic Emission Control Area (DECA) regulation, oceangoing vessels that ply the inland river shall use 0.1% sulfur fuel beginning on January 1, 2020; the same applies to ships operating inside China’s Hainan waters from January 2022 onward.

In March 2019, South Korea’s National Assembly passed a Special Act on Air Quality Improvement in Port and Other Areas to combat air pollution from shipping and port activities. The Special Act stipulates a series of measures, the most conspicuous is to set up the ports air quality control zones, which are also called as South Korea ECA. On August 20, 2019, South Korea’s

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Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency, Fig. 1 Map of the domestic ECAs in China. (Picture from <http://www.huataimarine.com/index.php?m=newscon&id=361&twid=543>)

ECA regulation was released in a regulation advance notice, which covers Korea's five main seaports: Busan, Incheon, Yeosu-Gwangyang, Ulsan, and Pyeongtaek-Dangjin, and the regulation will be introduced in stages.

More Pollutant Items Will Be Introduced into the Emission Control Framework

From the initial control of sulfur content in fuel oil, to the control of NO_x in ship emissions, and to the control of the ship's energy efficiency design index (EEDI), more and more pollutant items have been introduced into the shipping emission control framework by IMO. In fact, in the EU *Technical Requirements for Inland Navigation Vessels*, there are more types of pollutants that are to be controlled, including NO_x, carbon monoxide (CO), hydrocarbon gas (HC), and PM, and limit values for pollution index are given to each type of pollutants. In *United States Code of Federal Regulations 40 CFR, Part 94*, the pollutants of NO_x, CO, HC, and PM are also controlled, and limit values for CO and PM pollution index are given, respectively, while for NO_x and HC, a comprehensive limit value is given by the combined pollution index of NO_x and HC. With the increase in the prevention and control of air pollution in ships and applications of new equipment (such as gas fuel engines) and alternative fuels (such as methanol) on ships, it is foreseeable that more and more types of pollutants will be controlled by the ship emission regulations in the future.

The Emission Standards Will Be Continuously Upgraded

Based on the implementation route, it can be seen that IMO's emission standards for SO_x and NO_x are phased and progressive. In the EU's *Technical Requirements for Inland Navigation Vessels* and US federal regulations, the engine emission standards for ships are also continuously upgraded. In China, the coastal waters of Hainan Province will implement 0.1% sulfur fuel oil requirements from 2022 in China. Besides, China will also assess the feasibility of using marine fuel oil with a sulfur content of not more than 0.1% in due course and determine whether it is required for marine vessels

entering the coastal control area to use marine fuel with a sulfur content of not more than 0.1%_{m/m} from January 1, 2025. With the advancement of technology, the improvement of social cognition level, and the increasing environmental protection from the government, the standards for air pollution emissions of ships will be gradually upgraded.

Trend on the Design, Building, and Operation of Ships for Energy Efficiency

Energy-Efficient Design for Ships

Conventional Energy-Saving Technology

For ships in service, the adoption of energy-saving technologies can improve ship energy efficiency and thereby reduce CO₂ emissions. These technologies include but are not limited to:

1. Optimization of propulsion mode, such as electric power propulsion or diesel-electric combined power propulsion
2. Optimization of propellers, such as using new types of propellers or counterrotating propellers
3. Optimization of inflow conditions of propellers, increase propulsion performance, and reduce fuel consumption by using additional devices (such as fins and/or nozzles)
4. Using energy-saving accessories, such as front tubes, flow deflectors, propeller boss cap fins (PBCF), etc.
5. Optimization of hull resistance, such as using anti-drag coatings
6. Optimization of rudder blades, improving rudder blade shape to reduce drag
7. Optimization of diesel engines and boilers, selecting low oil consumption and high-efficiency main and auxiliary engines and boilers
8. Installation of waste heat recovery system

Application of New Energy

By applying sails or kites during voyage, ships can use wind energy to reduce energy consumption and emissions of atmospheric pollutants.

Some fitted ships (such as vehicle carriers, cruise ships, etc.) have also tried to use solar energy. However, due to limitations of cost, utilization, and others, we are still facing a lot of challenges in the application of new energy sources such as wind and solar energy on ships.

Use of Alternative Fuels

Alternative fuels such as natural gas, biofuels, and fuel cell technology can significantly reduce greenhouse gas emissions from ships. Taking natural gas as an example, since its major composition is methane (CH_4), which has low carbon atom content and high hydrogen atom content. Therefore, under the same calorific value, CO_2 produced by the combustion of natural gas is lower than that of marine fuel. In terms of biofuel, the production is currently limited, and if the production is blindly increased, it may have a negative impact on the environment (such as deforestation, wetland reduction, etc.) and may also affect food prices. Fuel cells are environmentally friendly and have a good prospect for marine applications in the future. However, fuel cells are still in the process of trial or pilot and have a certain distance from large-scale practical applications. Overall, natural gas is the fastest-growing alternative fuel on ships, with the most mature technology and product support.

Energy Saving in Ships' Operation

Shore Power

Some types of ships require a large power supply during the port to maintain the operation of critical electrical equipment, such as cargo handling pumps on large tankers and essential equipment such as air conditioning, kitchen, and lighting on cruise ships. For these types of ships, the use of shore power can significantly reduce fuel consumption and CO_2 emissions during ship berthing.

Strengthen Ship Operation Management

Measures such as strengthening ship operation management and optimizing loading and unloading processes can reduce land transportation, alleviate port congestion, shorten ship waiting and

turnaround time, and reduce greenhouse gas emissions accordingly. These measures are both cost-effective and environmentally beneficial. These measures can achieve significant emission reduction effects in the short term and have good development prospects. However, improving the overall operational management level in the transport chain is far from being achieved by the shipowners or ship operators alone but, to some extent, depends on the coordinated actions of many stakeholders. Ship operations involve a wide range of stakeholders, including charterers, ports, and ship traffic management agencies. All parties should jointly consider introducing a coordination mechanism and carry out overall management of ship voyages, routes, cargo capacity, port utilization, etc., to minimize ships CO_2 emissions under the premise of ensuring ship traffic.

Slow Steaming

According to the empirical data of the ship's speed down navigation, the ship's speed is reduced by 4%, and the greenhouse gas emissions can be reduced by 13%. The gradual (mandatory or voluntary) reduction of ship's speed in specific areas (e.g., 200 nautical miles from the shoreline) can achieve the objective of maritime greenhouse gas emission reduction as early as possible. This measure can be effectively combined with measures such as ocean-meteorological navigation and voyage planning to ensure that the fuel consumption and air pollutant emissions for each voyage are minimized. However, the reduction in speed may cause problems in the ship operation. In order to win the market share, ships must have a competitive navigational speed. Since speed reduction will lead to an increase in the transmit time of the cargo and will result in more ships being used to transport the same amount of cargo. No matter applying other transport modes or more ships participating in the operation, more emissions will be generated.

Optimization of Ship Draft and Trim

The ship draft and trim have a certain impact on the ship's energy efficiency and carbon emissions. Ship draft will affect the ship's navigational

resistance. For a given draft, main power, and ballast conditions, the ship has an optimal trim attitude. By considering the hull, propeller pitch, main engine power, and other factors, the software can provide an optimal trim. At present, this software has been developed and put into use, and the CO₂ emission can be reduced by 1–2%.

Optimization of Propeller Energy Efficiency

By adjusting the immersed depth of propellers, the fuel consumption of the ship can be reduced. However, it should be achieved by adjusting ballast, loading, fuel oil, fresh water tanks, and the main engine speed. By checking the surface condition of propellers, maintaining the surface finish of the blade, and repairing the damaged blade (crimping, corrosion) in time, the efficiency of propellers can be improved and the energy consumption of ships can be reduced.

Well Hull Maintenance

The growth of marine organism on the surface of hull will increase the ship's resistance, while regular hull cleaning can reduce ship's resistance and fuel consumption. In order to increase the energy efficiency, some measures can be undertaken on the hull, which include coatings to ensure the smoothness of the hull to reduce the attachment of marine organism and reduce the resistance of the hull; regular hull maintenance, in case ship fouling is accumulated and affects the speed of the ship; regular inspection and cleaning of the hull condition, ensuring that the ship is regularly docked and assessing the performance of the ship; and timely removal and replacement of underwater paint to avoid increasing the hull roughness which is caused by repeated blasting or multiple docking repairs.

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Fuzzy Control

- [Intelligent Control Algorithms in Underwater Vehicles](#)

G

GDW – Generalized Dynamic Wake Theory

► [Analysis of Renewable Energy Devices](#)

GIS: Geographic Information System

► [Global Navigation Satellite System \(GNSS\) Buoy](#)

Glider

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Synonyms

[Autonomous underwater glider \(AUG\); Underwater glider \(UG\)](#)

Definition

Underwater Glider (UG) is a new type of autonomous underwater vehicle (AUV), which can

perform three-dimensional observation from the surface to its working depth. UG follows a typical sawtooth-like profile in the ocean. The observation duration is generally several months, and the cruising range capacity can reach thousands of kilometers. It has the characteristics of low noise, low cost, and round-the-clock service. It can carry different mission sensors (e.g., CTD, Shear Probe) and is equipped with GPS navigation and positioning system. It has become a quasi-real-time, sustainable, large-scale, and high-resolution platform for marine observation and detection.

The concept of UG, a glider with a buoyancy engine powered by a heat exchanger, was first introduced to the oceanographic community by Henry Stommel in 1989, when he proposed a glider concept called Slocum, developed with research engineer Doug Webb from Woods Hole Oceanographic Institution (WHOI), founded by Naval Technology (ONR) (Stommel 1989). Today, the mature UG products on the market mainly include: the battery-powered SLOCUM glider of United States Teledyne Webb Research (TWR) (Webb et al. 2001), the Seaglider of United States Bluefin Robotics (Eriksen et al. 2001), the Spray glider of United States iRobot (Sherman et al. 2001), the Petrel glider of Tianjin University (TJU) (Ma et al. 2018; Liu et al. 2014a), and the Sea-Wing of Shenyang Institute of Automation at Chinese Academy of Sciences (SIA) (Yu et al. 2017).

Scientific Fundamentals

Glider Types

Based on the characteristics of different driving modes, there are few types of UG products: buoyancy-driven UG, hybrid underwater glider (HUG), and environmental-energy-driven UG. Jo Borchsenius proposed to use marine chemical energy to drive UG (Borchsenius and Pinder 2010); M. Arima developed a solar-energy-driven prototype UG (Arima et al. 2011); Andrea Caiti conducted a theoretical study of wave-energy-driven UG (Caiti et al. 2011); Webb Research Cooperation developed a thermal-driven UG (Webb et al. 2001).

Historical Development

Nowadays, the UG technology is mainly developed by the USA, France, Britain, and Australia. As the initiator and leader of the UG technology, the USA has been dedicating to study UGs since the 1990s. Today, the US UG technology has been mature with plenty of types of UGs such as Slocum, Spray, Seaglider, and ANT glider (Кожемякин et al. 2016).

In January 1991, TWR's first Slocum glider was marked as the birth of UG (Schofield et al. 2010).

In 1999, American institutions Scripps Institution of Oceanography (SIO) and WHOI worked together to develop the Spray glider. This glider's working depth was 1500 m (Rudnick et al. 2016). Then, the Spray glider was commercialized by the Bluefin Robotics Corporation.

In 1999, Applied Physics Laboratory at the University of Washington (APL, UW in America) developed the Seaglider. They designed a pressure hull whose compressibility was close to the sea water's, which could decrease the buoyancy change amount and save energy (Rudnick et al. 2004). Today, the Seaglider has been commercialized by Kongsberg Maritime.

In recent years, with the funding from the Office of Naval Research (ONR), an American company, the Exocetus Development LLC, developed the Exocetus Coastal glider (formerly called the ANT Littoral glider), which is applied in the

shallow seas. Till now, it has delivered 18 UGs to the US Navy, which has been operated for 4500 h.

With the extensive application of UG and the increase in their usage demand, the hybrid UG has gradually become the research hotspot in recent 10 years in order to improve underwater glider's antirflow capability in strong ocean currents.

In 2004, R. Bachmayer et al. from Princeton University in America proposed the concept of HUG's design for the first time. The underwater glider is equipped with a propeller on the tail to realize hybrid propulsion (Bachmayer et al. 2004).

In 2008, the French company ACSA developed the HUG SeaExplorer. It is integrated with the propeller system, which can switch modes freely between AUV/glider modes. Combined with an underwater acoustic positioning system, it can realize positioning without coming to the surface. It can be used for long-term and under-ice measurement. Now, SeaExplorer has been commercialized (Claustre et al. 2014).

In 2009, NATO Undersea Research Centre (NURC) began to develop the hybrid UG Folaga (Alvarez et al. 2009). In the same year, the French institution ENSIETA developed the hybrid UG Sterne (Wood and Mierzwa 2013).

In 2011, Monterey Bay Aquarium Research Institute (MBARI) developed a hybrid UG Tethys AUV. The glider's tail is equipped with a propeller which can realize short-term hybrid propulsion.

National Research Council Canada (NRC), the Memorial University of Newfoundland (MUN), and TWR added folding propeller thrusters to the tail of a Slocum glider made by TWR in 2012 to enhance the current glider's mobility in the shallow sea and expand the glider's scope of applications. When propulsion is needed, the propeller of the glider is deployed to increase the speed. When gliding is required, the glider's propeller folds to reduce the gliding resistance.

The UG research in China started in the early twenty-first century. In 2005, TJU developed a prototype of a UG driven by temperature difference energy. SIA developed a prototype of the UG prototype (Wang et al. 2005, 2006).

Since 2007, with the support of China National 863 Program, TJU developed a prototype of the

hybrid UG, while SIA carried out the research and development of the UG prototype in 2008. Zhejiang University (Yang et al. 2014), National Ocean Technology Center of China (Qin et al. 2016), Yichang Research Institute of Testing Technology at China Shipbuilding Industry Corporation (Chen et al. 2014), Shanghai Jiao Tong University, and Northwestern Polytechnical University (Tian et al. 2013) also conducted relevant studies on gliders. In recent years, China Ship Scientific Research Center of China Shipbuilding Industry Corporation has also researched UGs deeper than 500 m (Ma et al. 2007).

In 2012, TJU, SIA, Huazhong University of Science and Technology, and Ocean University of China jointly undertook a National 863 Program's project. This project had developed many types of UG engineering prototypes, which accelerates the deep-sea UG technology engineering.

From March to April 2014, relying on the above project, many types of UGs participated in the sea trial research in the South China Sea, which shortened the technical gap between China and other countries as well as established a foundation for the future productization of the UG.

From April to June 2015, TJU deployed several Petrel-II UGs to participate in the sea trials organized by Ministry of Science and Technology of China, which achieved fault-free work for 30 days and set several Chinese records such as 229 deep-sea observation profiles of 1000 m deep and 1022.5 km cruising range.

In March 2017, the Sea-Wing 7000 UG developed by SIA completed the deep-sea observation mission in the Mariana Trench, reaching a maximum depth of 6329 m.

In April 2018, the Petrel-X UG developed by TJU made its dive to 8213 m in the Mariana Trench, setting a world record for the working depth of the deep-sea gliders.

In May 2018, the long-voyage UG Petrel-L (Fig. 1), developed by TJU, operated for 119 consecutive days, completed 862 profiles, and completed the range of 2272.4 km. Again it has set some new records of Chinese UGs, including the longest continuous working time, the most measured profile, and the longest endurance distance, etc.

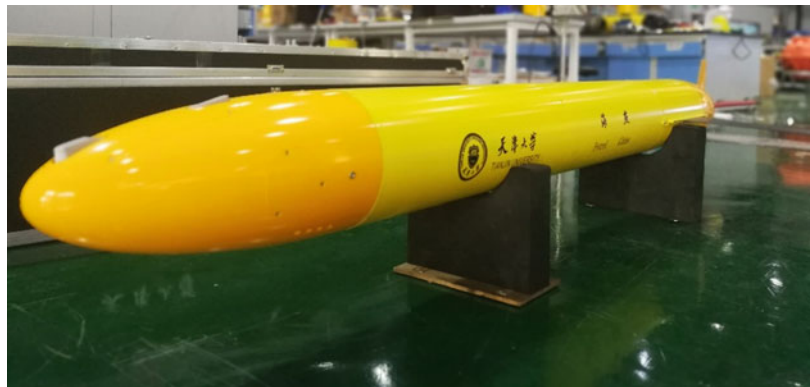
In recent years, China has also made key technological breakthroughs in hybrid UGs. Today, the Petrel HUG of TJU has completed the product design finalization and is being actively promoted and demonstrated.

The specific performance indexes of the mentioned commercial and developing UGs are summarized in Table 1.

The Key Technology in the Development of a Glider

UG's technical direction can be grouped into single-machine technology and networking technology (Fig. 2). Single-machine technology includes buoyancy-driven technology, multimode hybrid propulsion technology, attitude adjustment technology, lightweight pressure-resistant hull technology, and small low-power sensor design

Glider, Fig. 1 Petrel-L glider



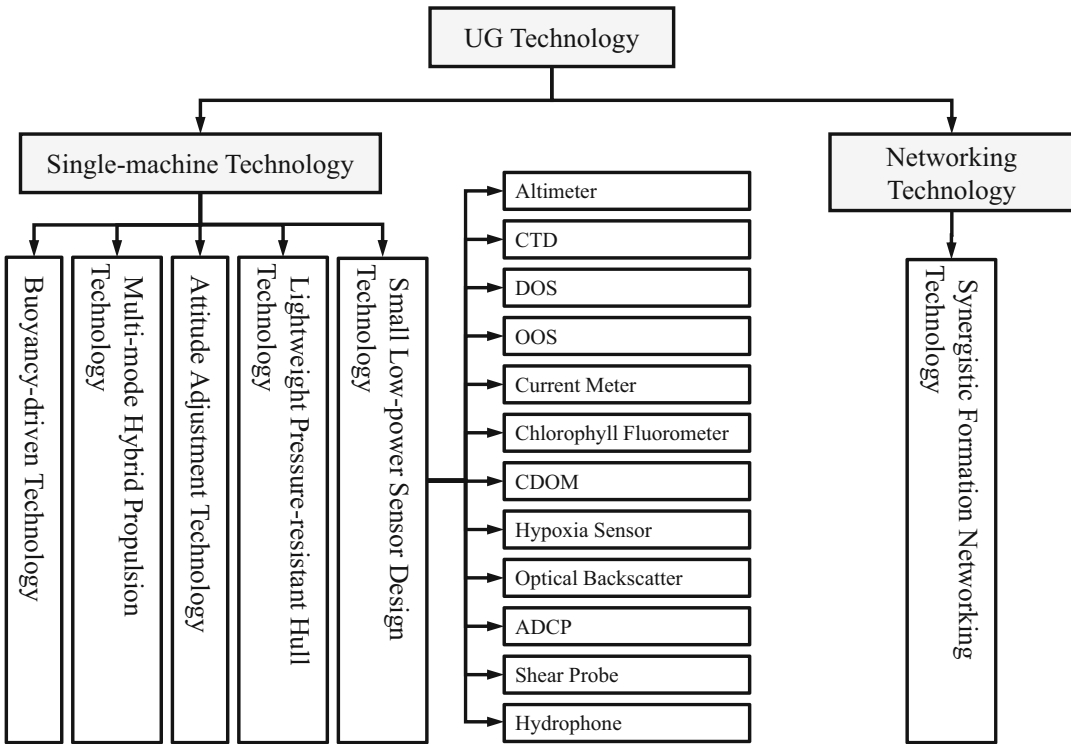
Glider, Table 1 UGs and their specifications

Name	Development organization	Specifications
Slocum electrical glider 	TWR, USA	Slocum G2: 1.5 m long, 0.22 m in diameter, the net weight of 55–70 kg, horizontal speed of 0.35 m/s Depth options of 30 m, 100 m, 200 m, 350 m and 1000 m (modular buoyancy engine dependent) Range: 350–1200 km (Alkaline batteries), 700–3000 km (Rechargeable batteries), and 3000–13,000 km (Lithium batteries) Deployment Length: 15–50 days (Alkaline batteries), 1–4 months (Rechargeable batteries) and 4–18 months (Lithium batteries)
Spray glider 	SIO and WHOI, USA	2 m long, 1.2 m wide, 51 kg net weight, deployment length of 6 months, operation speed of 0.23 m/s (max 0.25 m/s), range of 3600 km, and maximum depth of 1500 m
Seaglider 	APL, UW, USA	1.8 m long, 0.3 m in diameter, 1 m wide, antenna mast length of 1 m, 52 kg, depth of 1000 m, the range of 4600 km, 24 V, and 10 V batteries, 10 MJ, typical speed of 0.25 m/s
ANT Littoral glider/Exocetus coastal glider 	Exocetus Development LLC, USA	2 m long, 0.32 m in diameter, 2 m wide, 120 kg, depth of 200 m, maximum volume changeability of 5 L, the speed of 1 m/s
SeaExplorer hybrid glider 	ACSA, France	2 m long, 0.25 m in diameter, antenna mast length of 0.7 m, 0.565 m wide, 59 kg, maximum volume changeability of 1 L, the maximum horizontal speed of 0.5 m/s, the payload of 8 kg, depth of 700 m, deployment length of 2 months
Folaga hybrid glider 	NURC	2 m long, 0.14 m in diameter, 30 kg, the maximum horizontal speed of 1 m/s, the payload of 8 kg, depth of 100 m

(continued)

Glider, Table 1 (continued)

Name	Development organization	Specifications
Sterne hybrid glider 	ENSIETA Institution, France	4.5 m long, 0.6 m in diameter, 990 kg
Tethys hybrid glider 	MBARI, USA	2.29 long, 0.305 m in diameter
Slocum hybrid glider 	MUN, NRC and TWR, USA	Add folding propeller thrusters to the tail of a Slocum glider
Petrel-II hybrid glider 	TJU, China	1.8 m long, 0.3 m in diameter, 70 kg, depth of 1500 m, the speed of 0.5 m/s, design cruising range of 1500 km, deployment length of more than 40 days
Sea-Wing glider 	SIA, China	2.5 m long, 0.22 m in diameter, 1.2 m wingspan, 65 kg, depth of 1000 m, cruising range of 1000 km, the speed of 0.5 m/s, deployment length of more than 30 days
Other gliders of China 	Huazhong University of Science and Technology, Yichang Research Institute of Testing Technology at China Shipbuilding Industry Corporation, and Ocean University of China, etc., China	All tested in the South China Sea



Glider, Fig. 2 UG's main technical direction

technology; networking technology is oriented to high reliability and practical UG as well as conducting synergistic formation networking technology research.

Buoyancy-Driven Technology

Buoyancy-driven system is a set of a mechanical, electrical, and hydraulic coupling system with the function of volume change. It directly determines UG's performance and reliability which also affects the speed, range, and maneuverability of UG. Buoyancy-driven systems can be divided into two categories according to the method of volume change: One is to transport hydraulic oil between an internal and external bladder by pumps or pistons, such as the Seaglider and Spray gliders using reciprocating piston pump and Slocum electric gliders using single stroke pistons. The other is thermal engine through thermal expansions and contractions of PCM. Thermal engines can directly obtain the required energy from the environment for buoyancy change it

limited by the range of thermocline in different ocean areas. Table 2 shows the comparison of main technical parameters of the buoyancy-driven system of commercial UG, such as Slocum, Seaglider, Spray, and SeaExplorer (France), with Petrel developed by TJU (Davis et al. 2002; ALSEAMAR 2018).

In order to meet the needs of deep-sea observation technology, Deepglider of USA (Osse and Eriksen 2007), UnderDOG UG of New Zealand and high-pressure buoyancy-driven system with a depth of 2100 m of Japan all use the high-pressure buoyancy-driven system with micro axial piston pump as the core (Townsend and Shenoi 2016; Asakawa et al. 2016; Ma et al. 2016).

Multimode Hybrid Propulsion Technology

Multimode HUG has many advantages, such as better maneuverability, lower power consumption, longer cruising range, and stronger stealth. Multimode HUG is mainly driven by buoyancy, supplemented by propeller propulsion or water jet

Glider, Table 2 Technical parameters of typical UG's buoyancy driven system

UG	Institute	Technical parameters
Slocum	TWR, USA	Working depth: 200 m/1000 m Capability: 0.46 L 200 m, 0.52 L 1000 m Drive type: piston driven by lead screw/piston pump System efficiency: 50~60% (lead screw)
Seaglider	APL UW, USA	Working depth: 1000 m Capability: 0.85 L Drive type: piston pump System efficiency: 40%
Spray	Scripps UC, USA	The depth of design: 1500 m Working depth: 1000 m Capability: 0.7 L Drive type: piston pump System efficiency: 40%
ANT Littoral glider/Exocetus coastal glider	ANT, USA	Depth of design: 200 m/1000 m Drive type: pump/variable section piston
SeaExplorer	ACSA, France	Working depth: 850 m/700 m Capability: 1 L Drive type: pump System efficiency: 60%
Petrel	TJU, China	Working depth: 1500 m Capability: 1.4 L Drive type: piston pump System efficiency: 34%@1000 m, 40%@1500 m

propulsion. It realizes the functions of dynamic gliding and horizontal propulsion, improves the speed and manoeuvrability of UG to a certain extent, and enriches the movement mode of UG. However, the realization of these functions will inevitably increase its energy consumption and reduce its endurance. For this kind of technical difficulty, TJU has carried on a system analysis to the HUG movement pattern and the stability (Niu et al. 2017; Yang et al. 2016; Lei et al. 2016). Also, TJU has designed and developed a winged HUG, “Petrel” (Qin et al. 2017; Wang et al. 2011; Liu et al. 2014b). Harbin Engineering University has designed an underwater vehicle with a plastic shell and a water jet propeller (Guo et al. 2010, 2011).

Attitude Adjustment Technology

UG is a multibody system composed of shell, attitude adjustment system, and buoyancy adjustment system. The attitude adjustment system is an important part of UG, which is generally divided into “only the internal attitude adjustment technology” and “tail rudder and internal attitude

adjustment technology.” The internal attitude adjustment system achieves the adjustment of UG's barycenter position by controlling the movement of the mass block in the cabin to achieve effective attitude control.

1. Only the internal attitude adjustment technology

Only the internal attitude adjustment system is composed of a pitch adjustment mechanism and roll adjustment mechanism. According to the set gliding angle, the pitch adjusting mechanism controls the mass block to move along the UG axis to achieve UG's ups and downs. Moreover, the roll adjusting mechanism regulates the turning of UG by controlling the circumferential rotation of the eccentric mass block. The internal attitude control method, which controls the axial translation and circumferential rotation of the mass blocks in the cabin, applies to deep sea UG, such as Spray, Slocum thermal, Seaglider, Deepglider, and so on. These gliders all adopt dc motor to drive eccentric mass block for attitude

adjustments, which are usually the battery pack in the cabin. The attitude control of Petrel-II UG developed by TJU in China also adopts this method (Sang 2018).

2. Tail rudder and internal attitude adjustment technology

The UG structure that adjusts the attitude by adopting the axial translation of the mass cabin block and using the tail rudder attached to the UG's tail is relatively simple and is suitable for performing tasks in shallow water with relatively complex topography and geomorphology. For example, the battery-powered Slocum UG uses a stepper motor to adjust the steering of the tail rudder, which dedicates to control the steering of the UG. Similar to the battery-powered Slocum UG, the Sea-Wing UG developed by SIA, also adjust its attitude of UG by adjusting the tail rudder through stepper motor.

Lightweight Pressure-Resistant Hull Technology

When UG runs in the ocean, the seawater pressure it receives is proportional to the depth of the dive. Under external pressure, if the hull is deformed, buoyancy changes and even failure of the pressure-resistant hull may be caused. Therefore, it is especially important to optimize the structure of the UG pressure-resistant hull (Liu et al. 2013). To meet the demand for pressure requirements as well as having relatively lightweight, Wang et al. (2008) analyzed the structural parameters of the UG pressure-resistant hull. Based on the optimization of the UG's main body shape and improvement of the shell material, Song et al. (2014) used finite element analysis to analyze the structure and size of the UG's pressure shell. He et al. (2018) carried out a multiobjective optimization on the parameterized model of multibubble pressure chamber structure, using Kriging interpolation method and nondominated sorting genetic algorithm.

Small Low-Power Sensor Design Technology

At present, the design of small-scale and low-power sensors applied in gliders abroad has developed rapidly. The integration of multisensor and gliders has been successfully developed and implemented, and the substantial progress has

been made remarkably. In the design and integration of marine sensors, China possesses a good development basis, but the small-scale and low-power sensors for UG application have a large gap existing, compared to the international advanced technology level.

Synergistic Formation Networking Technology

UG can realize continuous observation and detection in long range and large scale in the complex ocean environment, and the formation, networking, and coordinated observation are the most important application modes of the UG. In almost all of the important international ocean observation systems and programs, there are research missions and application experiments for UG formation and network construction, showing the important role of UG network in ocean monitoring and detection. The comprehensive research of multi-UG formation network technology in China is still in its infancy stage. Moreover, key technologies involved in this section include multi-UG formation and collaborative control and network construction technologies.

Key Applications

Today, the UG platform is widely used worldwide, showing important application value in many aspects. It can greatly meet the various needs of marine science, marine economic development, disaster prevention, and national defense technology.

Civilian Demands

Emergency Response

UG can be used for emergency response, such as monitoring marine pollution, marine search and rescue. In the disaster of the oil spill in the US Gulf of Mexico on April 20, 2010, many organizations, enterprises, and research institutes volunteered to deploy multiple UGs to observe the temperature, salinity, and current velocity of the sea area to confirm the further moving direction of the crude oil.

Climate Variability Observation

A wide range of climate variability has emerged with the occurrence of the El Niño phenomenon. Today, the key observation to study climate variability is the boundary flows. Moreover, UG can provide strong support for climate variability study. Compared to other observation systems, UG can sample the dynamic environment continuously, which makes it an ideal platform for boundary flow observation. As early as 2004, UG had observed a boundary flow from Woods Hole and crossed to the Bermuda Islands. In 2009, the famous Scarlet Knight Slocum UG succeeded crossing to Atlantic, which had demonstrated that the UG has reliable sailing performance and long-lasting operational capability.

Marine Disaster Forecast

1. UG has reliable sampling capability in extreme environments, which can observe real time when the hurricane/typhoon passing and feed-back important first-hand dynamic temperature and salinity information; thus, the UGs can greatly improve the accuracy of hurricane/typhoon wind forecast, and make an important contribution to the safety of human life and property.
2. UG has the capability of multidimensional observation for Harmful Algal Blooms (HABs). Currently, a multidisciplinary crossed observation system with multiple platforms, which is constituted of the UG networks, satellites, surface ships, and mooring equipment, has become a new observation method for phenomena such as HABs.
3. Low dissolved oxygen is a big environmental issue, which affects marine water quality and fishery. UG has been successfully applied to complete the sampling task of dissolved oxygen in the offshore waters of the USA, Australia, and China.

Military Demands

The US military has developed advanced underwater winged gliders based on UG's acoustic application, acoustic UG XRay, and ZRay. They can be used for submarine monitoring and various low-frequency sound source monitoring (Stephen

2009; D'Spain et al. 2005). Therefore, UG platform has great utilization potentiality in national marine defense security field.

XRay

The XRay was developed primarily with the aid of the Marine Physical Laboratory, SIO and the APL, UW in 2005. It is designed to achieve the cruise speed of 1–3 knot and can realize the gliding range of 1200–1500 km, and be able to remain on-station up to 6 months in partial buoyant glides (Arima et al. 2014) (Fig. 3).

The vehicle is the largest UG with 6.1 m wingspan, which has an advantage regarding hydrodynamic efficiency and space for energy storage and payload. The lift to drag ratio was 17/1 in the sea trial in 2005. XRay's payload includes a low-power 32-unit hydrophone array placed along the leading edge of the wing, a large physical aperture for medium frequency (1–10 kHz) and a 4-component vector sensor. The cumulative data collected by the array illustrates the performance of the narrowband detection and localization algorithm. The flying behavior development of the flying wing maximizes the detection and positioning capabilities of the array. XRay 2 was improved in 2007 and had conducted 55 field tests in 2008, which verified that its specific flying behavior could improve the detection and positioning of hydrophone arrays.

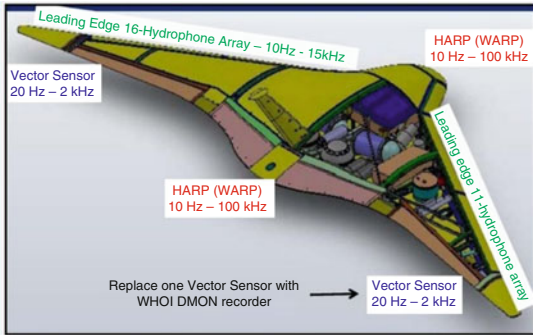
ZRay

ZRay is an autonomous underwater vehicle funded by the ONR and developed by a team funded by the ONR and developed by a team funded by the SIO, the APL, UW, and the US Navy's Space and Naval Warfare Systems Center Pacific. The 6 m wide blended wing-body hull form glider was completed in 2010 and is based on the Liberdade class model developed by ONR since 2004 (Hildebrand et al. 2009).

ZRay collects acoustic data via a 27-element hydrophone array installed in sonar housing in the glider's leading edge. Wide-band acoustic sensors are also located in the nose and tail. The long endurance (1 month, 1000 km range) vehicle has oceanographic and potential antisubmarine warfare applications. In 2011, ZRay conducted



Glider, Fig. 3 XRay glider in SIO (Office of Naval Research 2006)



Glider, Fig. 4 ZRay glider (D'Spain 2012)

testing off San Diego in support of the Navy's Passive Acoustic Autonomous Monitoring (PAAM) of Marine Mammals Program. PAAM is designed to track marine mammals over long periods in Navy acoustic testing areas (Fig. 4).

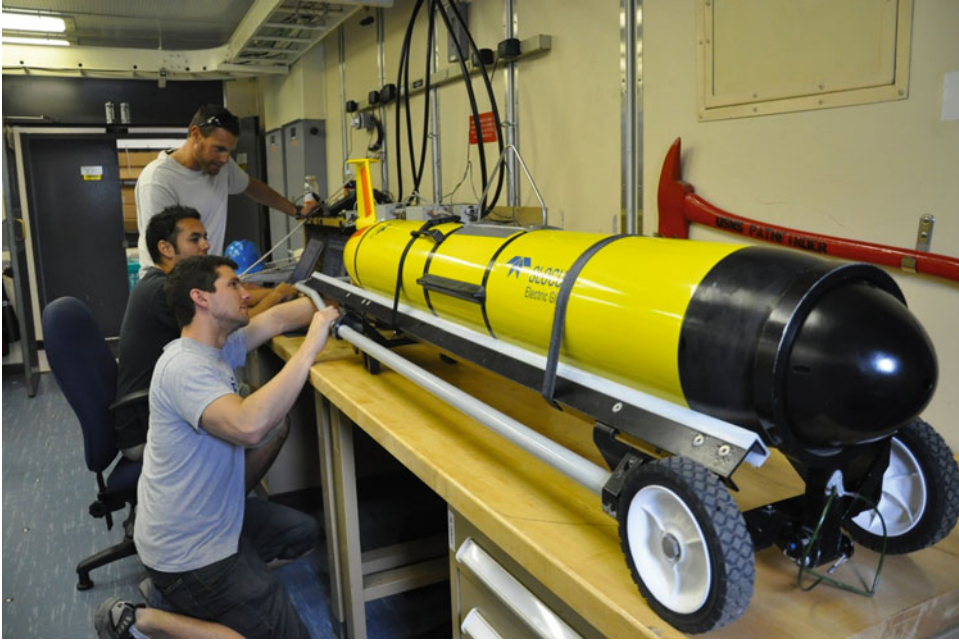
LBS-G

Teledyne Brown Engineering, Inc., built the Littoral Battlespace Sensing-Glider (LBS-G) for the US Navy to obtain important oceanographic data and improve the fleet positioning during naval exercises. Considering the influence of the temperature layer on sonar propagation, the LBS-G

has obvious functions for hidden and submarine hunting.

The LBS-G is propelled by changes in buoyancy along with its wings and tail-fin steering. The LBS-G can operate up to 30 days per time by a Lithium battery. Its dive depth is less than 1000 m, and the single section can glide 5,000 m. The sensors in the “nose” collect oceanographic data. While on the surface, LBS-G reports its location, transmits data via satellite, and receives mission orders.

In 2009, the LBS-G glider completed the transatlantic mission in 223 days. Pilot production



Glider, Fig. 5 Engineers run predeployment inspections on a littoral battlespace sensing glider. (US Navy photo by Rick Naystatt)

began at the end of 2010. In 2011, it was approved by the US Navy to enter the mass production phase of the underwater coastal theater LBS-G project. In 2013, the US Navy increased the investment in this project, hoping to apply it as “the important landmine countermeasures and other missions for expeditionary warfare, which intends to reduce or eliminate the times of sailors and marines enter into dangerous shallow and offshore regions to clear landmines” (U.S. Navy Program Guide 2012 2014) (Fig. 5).

Cross-References

► [Autonomous Underwater Vehicle \(AUV\)](#)

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Global Buckling of Offshore Pipelines

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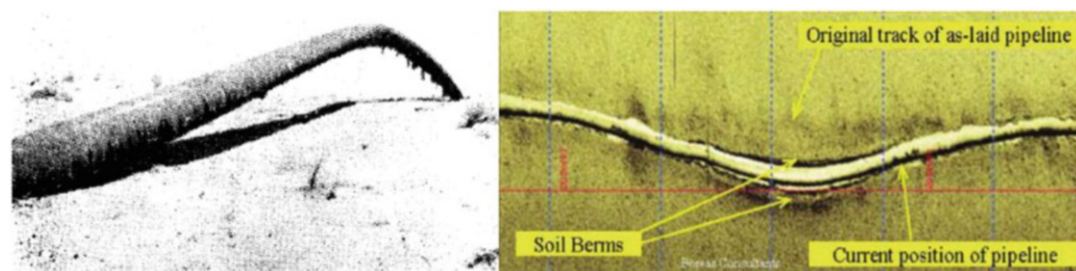
Introduction

Pipelines constitute a vital means of transporting and distributing liquids and gases such as oil, gas, and water over long distance. Since the early 1970s of the last century, pipelines have become one of the main means of transporting offshore oil and gas in many parts of the world. In-service hydrocarbons must be transported at high pressure/high temperature (HP/HT) to ease the flow and prevent solidification of the wax fraction. The difference between the high-pressure/high-temperature (HP/HT) and the as-laid conditions would cause pipelines to expand. The axial

expansion, however, is restrained either by soil friction, rocks, or anchors, thus inducing significant compressive axial loads in the pipeline wall. When the axial force reaches the lowest global buckling capacity, the pipeline will buckle globally, and the effective axial force will drop. If the pressure or temperature is further increased, the post-buckling will happen.

Global buckling of a pipeline means the loss of stability of the pipeline similar as a bar in compression. The global buckling may appear either horizontally (lateral buckling on the seabed) or vertically (upheaval buckling of buried pipelines) (DNVGL-RP-F110 2017). As shown in Fig. 1, both the upheaval and lateral buckling of pipelines can have serious consequence for the integrity of a pipeline. Consequently, considerable studies have focused on the global buckling of pipelines and intend to provide reasonable design guidance to prevent or control global buckling of pipelines.

There is a large volume of material published on both the upheaval and lateral buckling of pipelines, which covered the different related aspects of the buckling pipelines including mechanism of global buckling, parameter analysis, engineering case study, and mitigation measures for global buckling. The buckling mechanism of a pipeline is the basis of other related researches. This entry aims to describe the history of the theoretical studies on the global buckling of offshore pipelines (i.e., vertical buckling of buried subsea pipeline and lateral buckling of pipelines exposed on seabed). This entry also provides summarized information of the current state of the art and achievement in this area by summarizing published literature including also possible review



Global Buckling of Offshore Pipelines, Fig. 1 Upheaval and lateral buckling of pipelines (Bruton et al. 2007)

articles. Engineers are hoped to benefit from the extensive literature review.

Theoretical Background: Global Buckling and Its Mechanism

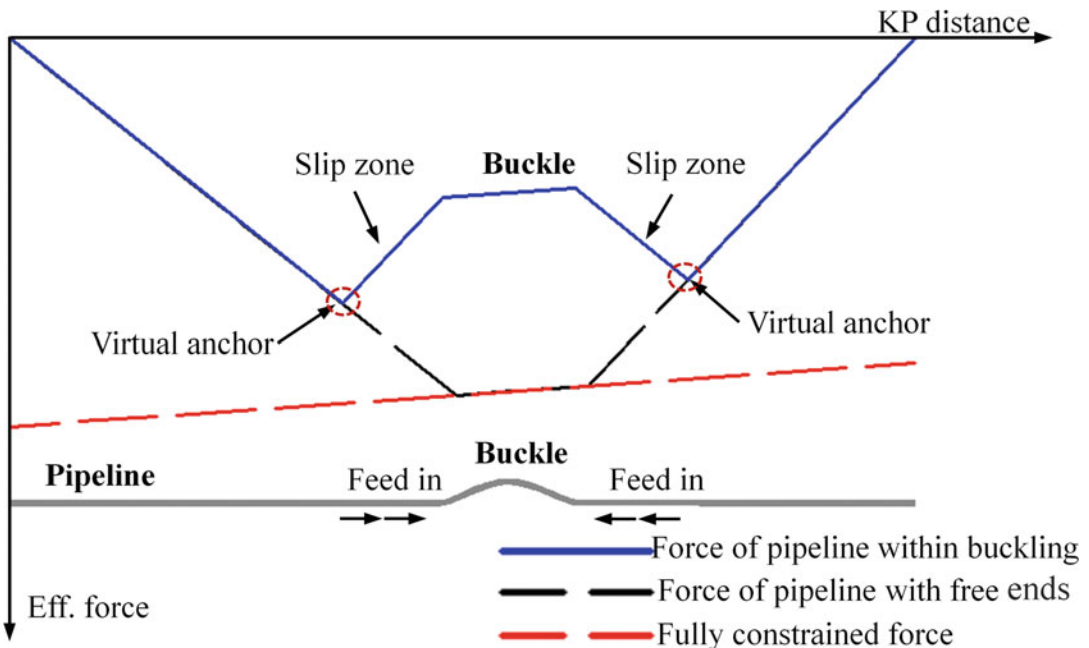
Global buckling of an offshore pipeline operated at HP/HT can be described as Fig. 2. When the temperature or pressure increases, the effective axial compressive force will be induced in the pipeline. If the pipeline is fully restrained, the compressive axial force will increase to the maximum effective axial force that can occur in a pipeline (i.e., the fully constrained force in Fig. 2).

For a pipeline with free ends, the axial force will be released as the pipeline experiences greater end expansion, so the force at the ends is zero. However, as the cumulative axial resistance increases with distance from the pipeline ends, the force can increase to a condition of full constraint over some of its length, as illustrated by the black line in Fig. 2.

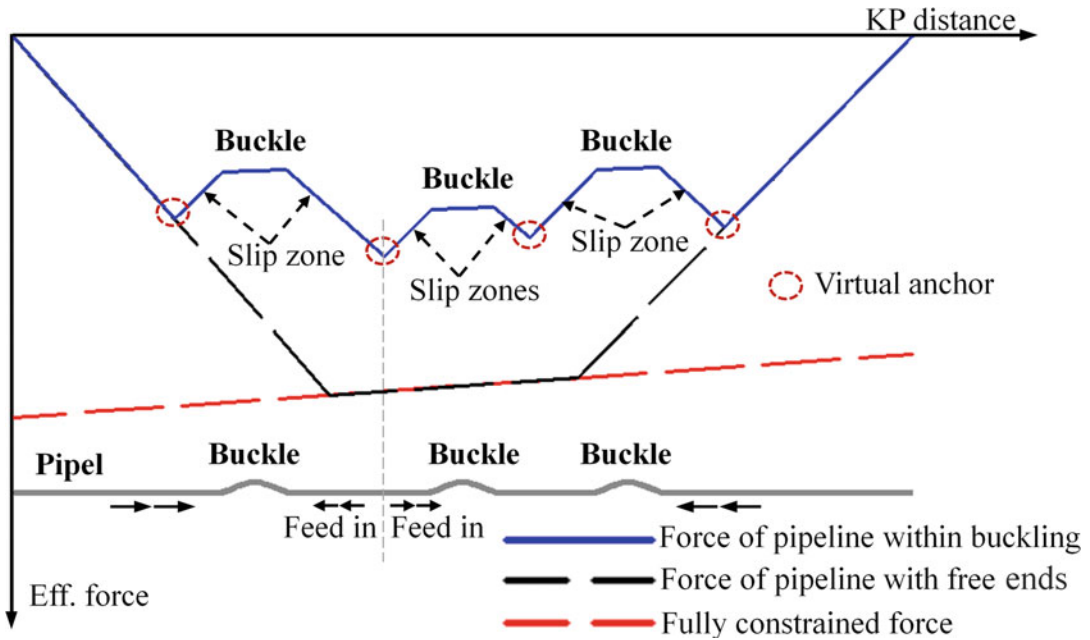
According to Hobbs (1984), this could lead to global buckling of the pipeline if the compressive axial force reaches the critical buckling force. The force in the buckle drops as the buckle develops, which will cause the pipeline section around the buckle to axially feed in to the buckle. The feed-in results in the axial force release in the slip zone. As illustrated by the blue line in Fig. 2, the slope of the force profile in slip zones is governed by the axial friction, and the length of the slip zone depends on the available frictional resistance to oppose the feed-in. And a virtual anchor will be developed at the end of the feed-in zone.

Figure 2 shows the scenario for the post effective force of pipeline within a single isolated buckle. For a pipeline with a longer length, it is possible that more than one buckle forms in the system (Carr et al. 2003), as illustrated in Fig. 3, where the pipeline has formed three buckles (the number of buckles is entirely case dependent).

Between adjacent buckles there must be a virtual anchor point at which the direction of pipeline



Global Buckling of Offshore Pipelines, Fig. 2 Post effective force of pipeline within a single isolated buckle. (Adapted from Carr et al. 2003)



Global Buckling of Offshore Pipelines, Fig. 3 Post effective force of pipeline within more than one buckle. (Adapted from Carr et al. 2003)

expansion changes. This effectively divides the pipeline up into three short pipelines anchored at each end and a discrete series of isolated buckles between anchor points. According to the paper (Bruton et al. 2007), as the buckles are spaced such that the distance between successive buckles is less than the total buckle length of a long pipeline ($L + 2L_s$ in Fig. 4) of an isolated buckle, the feed-in is shared between the two buckles that is known as expansion sharing in DNV (DNVGL-RP-F110 2017).

Global Buckling of Pipeline on Rigid Foundation

The rigid-base model is an approach that uses a beam on a rigid base with Coulomb friction to analyze the problem of in-service buckling of pipelines, both upheaval and lateral. In this model the pipeline-soil interaction is constant and determined by the buoyancy density and friction coefficient, independent of the deflection of the pipeline.

Hobbs' Model for Perfect Straight Pipelines

Vertical buckling model. Based on the rigid-base model of a straight railway track described by Kerr (1974), Hobbs (1981, 1984) has described a rigid-base model for a long pipeline within vertical buckling (ref. to Fig. 4).

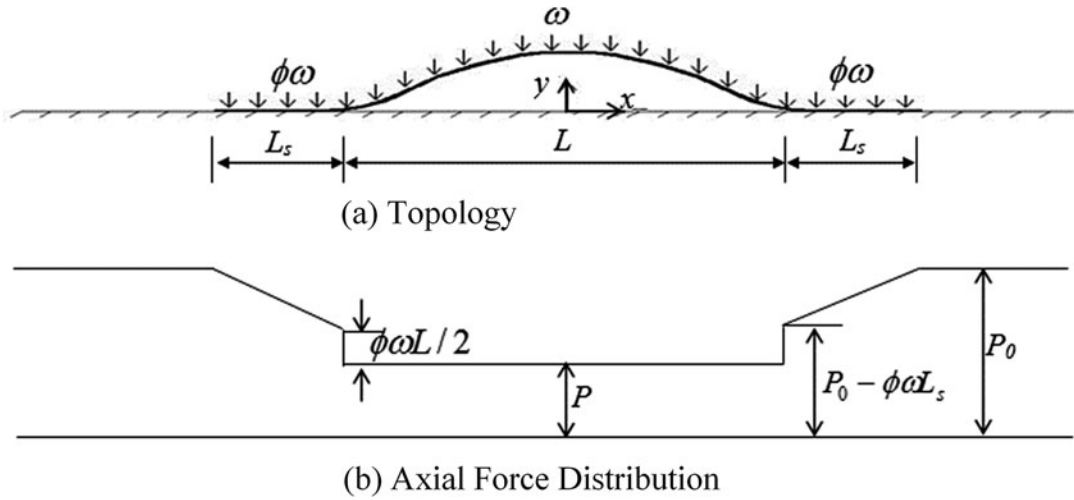
A linear differential equation has been established to describe the imperfection shape of the buckled part of the pipeline. It is assumed that the bending moment at the liftoff point is zero. In the notation of Fig. 4,

$$y'' + n^2 y + \frac{m}{8} (4x^2 - L^2) = 0 \quad (1)$$

where $m = w/EI$, $n^2 = P/EL$, and w is submerged weight of pipeline (including weight coat) per unit length.

With the boundary conditions $y|_{x=0} = v_m$, $y_x|_{x=0} = 0$ and $y|_{L/2} = y_x|_{L/2} = y_{xx}|_{L/2} = 0$, the following results are obtained by

$$P = 80.76 \frac{EI}{L^2} \quad (2)$$



Global Buckling of Offshore Pipelines, Fig. 4 Axial force of a pipeline with vertical buckling. (Adapted from Hobbs 1981)

$$P_0 = P + \frac{wL}{EI} \left[1.597 \times 10^{-5} EA \phi w L^5 - 0.25 (\phi EI)^2 \right]^{\frac{1}{2}} \quad (3)$$

where P is the axial load in the buckle and P_0 is the axial load away from the buckle (Fig. 4). P_0 created by full restraint of thermal expansion is simply $P_0 = AE\alpha T$. ϕ is the coefficient of friction between pipeline and subgrade.

The maximum amplitude of the buckle is

$$v_m = 2.408 \times 10^{-5} \frac{wL^4}{EI} \quad (4)$$

And the maximum bending moment, at $x = 0$, is

$$M_m = 0.06938 w L^2 \quad (5)$$

Lateral buckling model. With the vertical analysis to hand, the lateral buckling had been analyzed for the case of mode 1–mode 4 and mode ∞ (Fig. 5) by Hobbs (1984).

The values obtained for compressive loads and buckle amplitude are of the same form as Eqs. (2–5), but with different constant k_1 – k_5 . The appropriate constants can be found from Table 1:

$$P = k_1 \frac{EI}{L^2} \quad (6)$$

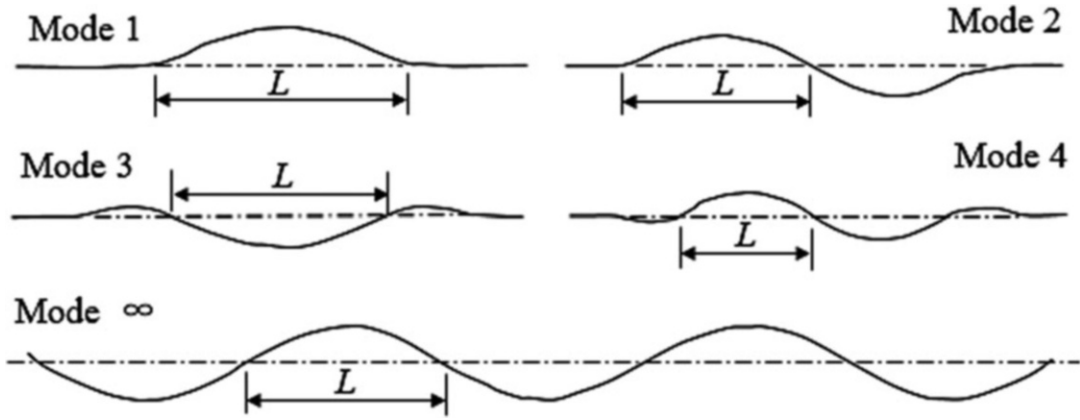
$$P_0 = P + k_3 \phi w L \left[\left(1.0 + k_2 \frac{AE \phi w L^5}{(EI)^2} \right)^{0.5} - 1 \right] \quad (7)$$

$$v_m = k_4 \frac{\phi w}{EI} L^4 \quad (8)$$

$$M_m = k_5 \phi w L^2 \quad (9)$$

Taylor's Model for Imperfect Pipelines

Engineering practice shows that the in situ shape of a pipeline in practical terms is far from being straight. The imperfections concerned may include a crest in the seabed profile, or a prop, as when isolated rock is located immediately below the line or another pipeline is to be crossed. Figure 6 describes three types of imperfection adapted from Taylor and Tran (1993). With the temperature rising, buckling may take place in the pipeline because of the existence of initial imperfection. Many researches have been performed on imperfect pipelines.



Global Buckling of Offshore Pipelines, Fig. 5 Lateral buckling modes of Hobbs' analytical solution (Hobbs 1984)

Global Buckling of Offshore Pipelines, Table 1 Constants for lateral buckling modes (Hobbs 1984)

Mode	k_1	k_2	k_3	k_4	k_5
1	80.76	6.391×10^{-5}	0.5	2.407×10^{-3}	0.06938
2	$4\pi^2$	1.743×10^{-4}	1.0	5.532×10^{-3}	0.1088
3	34.06	1.668×10^{-4}	1.294	1.032×10^{-2}	0.1434
4	28.20	2.144×10^{-4}	1.608	1.047×10^{-2}	0.1483
∞	$4\pi^2$	4.7050×10^{-5}		4.4495×10^{-3}	0.05066

Taylor and Gan (1986) proposed a modified version of the Hobbs' model, in which the pipeline has an initial imperfection of amplitude v_o and length L_o . In their analysis, the morphological expression of the initial geometric imperfection of the pipeline was similar to the buckling deformation of perfect straight pipelines given by Hobbs. And they considered that the buckling of pipeline was the growth of initial imperfections. The essential features of the vertical mode are shown in Fig. 7. The submerged self-weight of the pipeline per unit length is denoted by q , and v_m represents the maximum amplitude of the buckled pipeline.

Based on the buckling amplitude of perfect straight pipelines, the imperfection deflection expression can be assumed to be

$$v_o = \frac{v_{om}}{k_1} \left(1 + \frac{n_o^2 L_o^2}{8} - \frac{n_o^2 x^2}{2} - \frac{\cos n_o x}{\cos n_o L_o / 2} \right) \quad (10)$$

$$-L_o/2 \leq x \leq L_o/2$$

$$v_m = 2.407 \times 10^{-3} \frac{q}{EI} L_o^4 \quad (11)$$

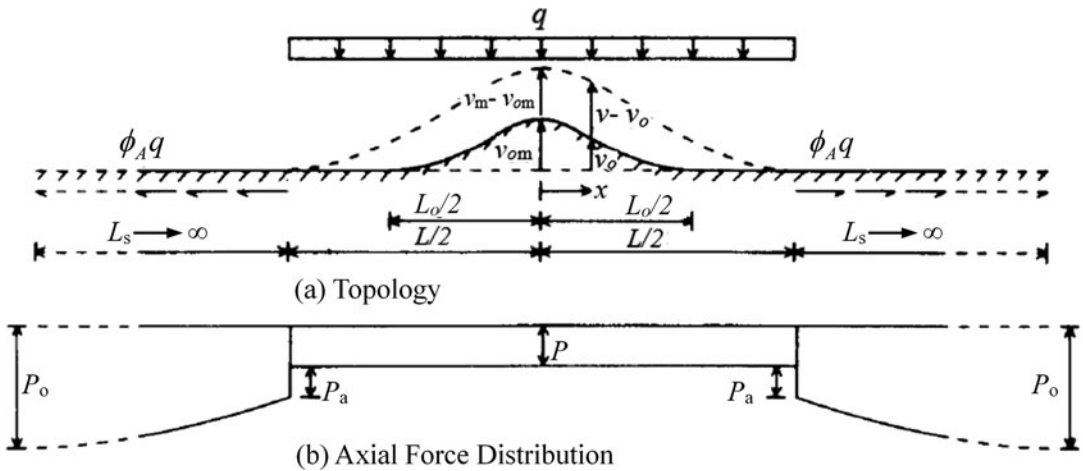
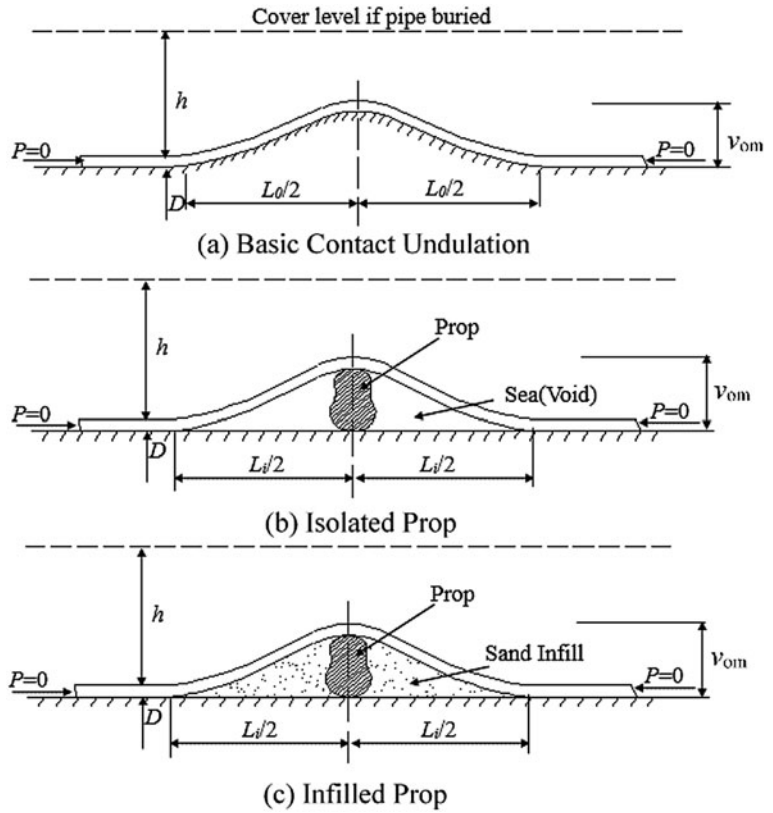
Employing symmetry, the total potential energy relating to the deformed state is given by

$$V = \int_0^{L_o/2} EI/2 (d^2v/dx^2 - d^2v_o/dx^2)^2 dx + \int_{L_o/2}^{L/2} EI/2 (d^2v/dx^2 - d^2v_o/dx^2)^2 dx + \int_0^{L_o/2} q(v - v_o) dx + \int_{L_o/2}^{L/2} q(v - v_o) dx - \int_0^{L_o/2} P/2 [(dv/dx)^2 - (dv_o/dx)^2] dx - \int_{L_o/2}^{L/2} P/2 [(dv/dx)^2 - (dv_o/dx)^2] dx = 0 \quad (12)$$

The corresponding equilibrium state being given by $dV/dv_m = 0$. Substitute Formula (10)

Global Buckling of Offshore Pipelines,

Fig. 6 Typical imperfection configurations (Taylor and Tran 1993)



Global Buckling of Offshore Pipelines, Fig. 7 Vertical buckling of mode 1 triggered by a single-arch symmetric imperfection (Taylor and Gan 1986)

and Formula (11) into Formula (12) to obtain the expression of P :

$$P = 80.76 \frac{EI}{L^2} \left[1 - \frac{R_1}{75.60} \left(\frac{L_o}{L} \right)^2 \right] \quad (13)$$

where

$$R_1 = 4.60314 \left[\sin(4.4934L_o/L) + 2.30157 \left(\frac{\sin 4.4934(1 + L_o/L)}{(L/L_o + 1)} + \frac{\sin 4.4934(1 - L_o/L)}{(L/L_o - 1)} \right) \right]$$

For pipelines with imperfection ratios V_{om}/L_o varied from 0.003 to 0.010 and the remaining pipeline parameters fixed, the relationship between the buckling force and buckling amplitude can be produced by substituting the parameters into the above formula, as shown in Fig. 8. It shows that the smaller the imperfection amplitude is, the greater the buckling force is. For the relatively small imperfection ratio case (0.003), it displays a maximum buckling force P_m , in which the snap buckling phenomenon can be clearly observed. The remaining two cases (0.007 and 0.010) generate stable post-buckling paths.

Based on the analysis of vertical buckling, Taylor also studied the lateral buckling of the first mode and the second mode. The analysis for the buckling region is identical to that of the vertical mode. For the lateral buckling of the first mode,

the values obtained for compressive loads and buckle amplitude are of the same form as Eq. (13). For the lateral buckling of the second mode, the topology and axial force distribution can be depicted as Fig. 9.

Antisymmetric imperfection expression (Taylor and Gan 1986):

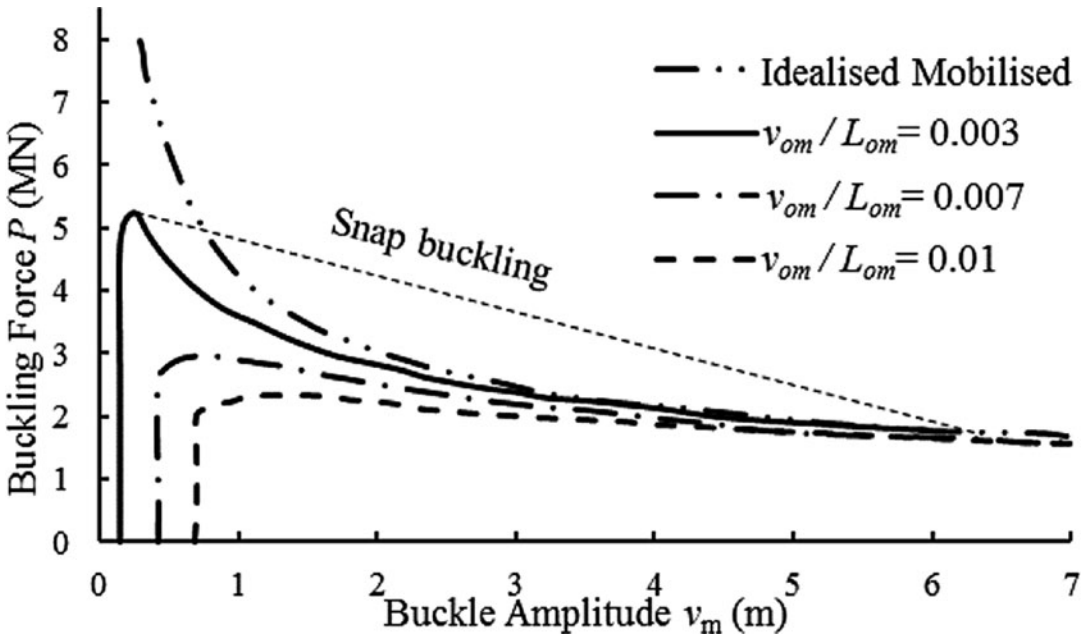
$$v_o = \frac{v_{om}}{k_2} \left[1 - \cos\left(\frac{2\pi x}{L_o}\right) + \pi \sin\left(\frac{2\pi x}{L_o}\right) + \frac{2\pi^2 x}{L_o} \left(1 - \frac{x}{L_o}\right) \right] \quad 0 \leq x \leq L_o \quad (14)$$

where

$$v_m = 5.532 \times 10^{-3} \frac{\phi_L q L_o^4}{EI}$$

Substitute Formula (14) into Formula (12) to obtain the expression of P (Formula 15) by applying the statics criterion $dV/dv_m = 0$:

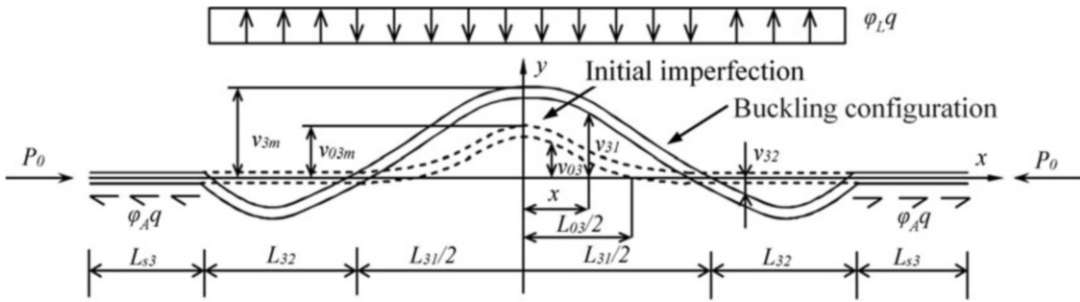
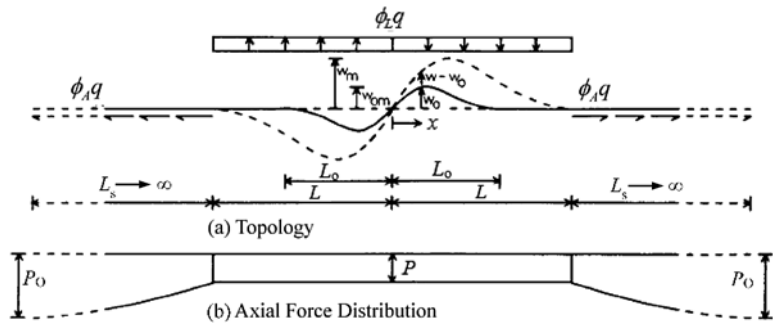
$$P = \frac{4\pi^2 EI}{L^2} \left[1 - \frac{3}{5} \left(\frac{R_2}{3\pi + \pi^3} \right) \left(\frac{L_o}{L} \right)^2 \right] \quad (15)$$



Global Buckling of Offshore Pipelines, Fig. 8 Thermal action characteristics – vertical mode (Taylor and Gan 1986)

Global Buckling of Offshore Pipelines,

Fig. 9 Lateral buckling of mode 2 triggered by a two-arch antisymmetric imperfection (Taylor and Gan 1986)



Global Buckling of Offshore Pipelines, Fig. 10 Global buckling of mode 3 triggered by a single-arch symmetric imperfection (Hong et al. 2015)

where

$$R_2 = \sin(2\pi L_o/L) \left[\frac{(L/L_o)(\pi^2 + L/L_o)}{1 - (L/L_o)^2} \right] + 2\pi L_o/L$$

$$+ \pi [1 - \cos(2\pi L_o/L)] \left(\frac{L}{L + L_o} \right)$$

Further Publications on the Rigid-Base Model

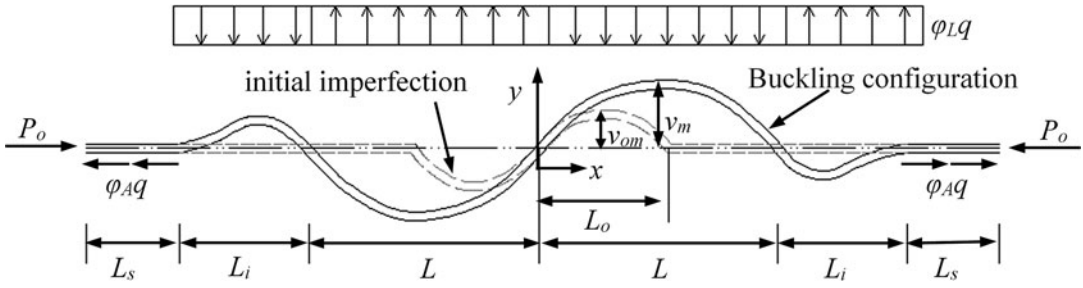
Various permutations of the rigid-base model have been described by other authors. Some of these are described as follows.

The analytical solutions of high-order buckling modes (mode 3 and mode 4 in Fig. 5) of perfect straight pipeline were proposed by Liu et al. with the energy method (Liu et al. 2014). Further, Liu et al. performed a series of analytical studies on the high-order global buckling of pipelines with initial imperfection. For example, Hong and Liu (Hong et al. 2015) suggested that the pipeline with a single-arch initial imperfection would eventually take place the third-order global buckling

(Fig. 10) and proposed an analytical solution containing formulas for calculating the critical axial forces and buckling amplitudes of imperfect pipelines exhibiting buckling of mode 3. The analytical solutions of high-order lateral buckling for a pipeline with a single-arch symmetric (Fig. 10) and a two-arch antisymmetric initial imperfection (Fig. 11) are derived by Liu and Wang using the static equilibrium method (Liu and Wang 2018).

Many other researches have been done on imperfect pipelines, for example, Ballet and Hobbs (1992) investigated the possibility of asymmetric buckling in the prop case; Hunt and Blackmore (1997) studied the effects of asymmetric bed imperfections, typified by a step, and adopted a shooting method to solve the standard fourth-order linear ordinary differential equation; Maltby and Calladine (1995a) proposed a simple formula for the axial load at which the localization of buckling occurs based on sinusoidal imperfection assumption.

These authors report similar conclusions to those of Hobbs and Taylor. The presence of



Global Buckling of Offshore Pipelines, Fig. 11 Buckling of mode 4 triggered by a two-arch antisymmetric imperfection (Liu and Wang 2018)

small initial imperfections does not affect the minimum buckling load; and snap-through buckling will only occur for imperfections which are smaller than a certain critical size. For larger imperfections, the size of the buckling grows steadily as the temperature increases.

Global Buckling of Pipeline on a Deformable Foundation

The rigid-base analyses mentioned in the previous section employed idealized Coulomb friction forces; but real pipeline-seabed interaction can be much more complicated than this (Dingle et al. 2008). Measured pipeline responses on various soils have shown a high degree of deflection dependence. To deal with this problem, a beam on an elastic foundation of stiffness k per unit length is a preliminary to either lateral or upheaval buckling (Kerr 1974). The governing equation for this condition is described as Formula (16):

$$EI \frac{d^4 y}{dx^4} + P \frac{d^2 y}{dx^2} + ky = 0 \quad (16)$$

Subsequently, a modified version of the elastic-base model has been proposed by Tvergaard and Needleman (1980), in which the response of the soil is nonlinear. The governing equation for this condition is described as Formula (17), where $F(x)$ is a deflection-dependent function of the nonlinear pipeline-soil interaction:

$$EI \frac{d^4 y}{dx^4} + P \frac{d^2 y}{dx^2} + F(x) = 0 \quad (17)$$

The rationality of the deflection-dependent friction model depends on two main aspects: the correct expression of pipeline-soil interaction and the accurate solution of the governing equation. Various permutations pipeline-soil interaction models have been applied to this governing equation, and several solutions, such as nonlinear perturbation analysis, Galerkin method, and shooting method, have been proposed to solve the equation. Some of these are presented as follows.

Models with Nonlinear Soil Resistance

$$F = ky - cy^m$$

Hunt et al. (1989) analyzed the global buckling of elastic structures with bending stiffness EI , resting on a deformable foundation with a soil resistance $F = ky - cy^m$. The governing equation was proposed as Formula (18):

$$EI \frac{d^4 y}{dx^4} + P \frac{d^2 y}{dx^2} + ky - cy^m = 0 \quad (18)$$

where P is the axial load, y is the buckling displacement of the elastic structure, and k and c are constant coefficients; here the authors concentrate on the asymmetric form of $m = 2$.

The perturbation analysis was used for the solution of the governing differential equation, both periodic (constant amplitude) and modulated (varying amplitude) solutions.

If the nonlinear term is dropped from Eq. (18), it is a classical buckling problem with critical load (Van Der Meer 1985):

$$P^c = 2\sqrt{kEI} \quad (19)$$

A nonlinear perturbation scheme, reducing the differential Eq. (19) to an ordered sequence of linear equations, can be initiated with the series representations:

$$\begin{cases} y(x, X) = y_1(x, X)s + y_2(x, X)s^2 + y_3(x, X)s^3 + \dots \\ P = P^c + P_1s + P_2s^2 + P_3s^3 + \dots \end{cases} \quad (20)$$

where s is a perturbation parameter growing from zero with movement away from the critical point C. Knowing that C is an unstable symmetric point of bifurcation, it is convenient to set $s = \sqrt{P^c - P}$.

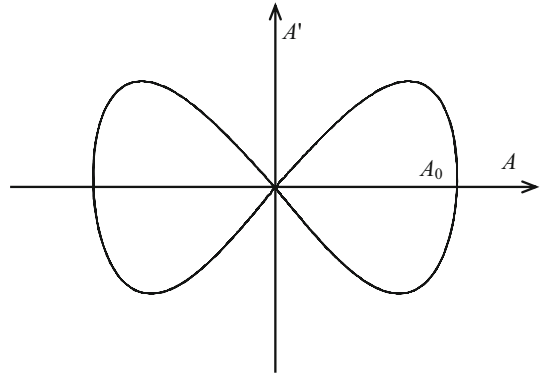
Substituting the expanded Eq. (20) into the nonlinear Eq. (18) and setting the coefficients of s, s^2 , etc. to zero gives an ordered series of sequentially linear differential equations in $y_1(x, X)$, $y_2(x, X)$, etc. (Thompson and Hunt 1973):

$$\begin{aligned} y(x, X) &= A(X) \cos(\omega x) \cdot s \\ &+ \left[\frac{1}{2} \frac{c}{k} A(X)^2 + \frac{1}{18} \frac{c}{k} A(X)^2 \cos(2\omega x) \right] \cdot s^2 \end{aligned} \quad (21)$$

where $A(X)$ is determined by Runge-Kutta solutions, as illustrated in Fig. 12. Both A and A' should lay on homoclinic orbits.

The buckling mode can be obtained by Formula 21 by setting buckling force $P = nP^c$ (n is less than 1). The results of buckling under the condition of $P = 0.98 P^c$ and the condition of $P = 0.95 P^c$ are shown in Fig. 13.

Many other investigators have studied the form in the presence of nonlinear term when they deal with the problem of structural buckling. For the form of nonlinear soil resistance $F = ky - cy^m$, Hunt et al. (1989) studied only the asymmetric case of $m = 2$; Whiting (1997) presented a Galerkin method for the symmetric case of $m = 3$; Zeng and Duan (2014) tackled the case of nonlinear terms covering both $m = 3$ and $m = 5$.



Global Buckling of Offshore Pipelines, Fig. 12 Homoclinic orbits

Further Publications on the Deflection-Dependent Friction Model

Furthermore, many nonlinear lateral soil resistance models, other than nonlinear soil resistance with the form of $F = ky - cy^m$, were applied to the governing equations of buckling of pipelines or elastic beam columns on deformable foundation. The solution to the governing differential equation was proposed based on nonlinear perturbation analysis, Galerkin method, shooting method, and so on.

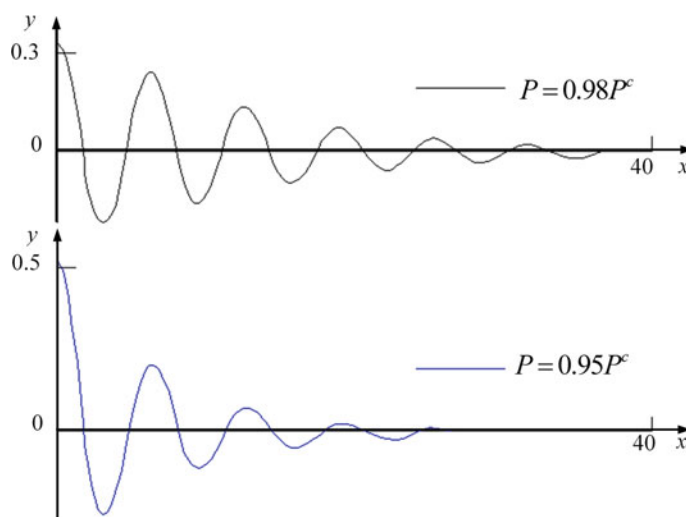
For example, based on the research of Tvergaard and Needleman (1980), an exponential nonlinear soil resistance with the form of $F = Q_0(1 - e^{(v/\Delta)})$ (where Q_0 is the maximum soil resistance per unit length of the pipeline, Δ is the corresponding displacement of Q_0) was employed to obtain the analytical solutions of an imperfect pipeline by Maltby and Calladine (1995b).

Lagrange and Averbuch (2012) have studied the periodic solutions of a strut on a nonlinear elastic Winkler-type foundation with imperfection in the form of a sine shape. The nonlinear restoring force of the foundation was either a bilinear or an exponential profile. Piecewise solution theory was used to solve the governing equations for the bilinear restoring force. Piecewise solution theory was also employed by Karampour et al. (2015) to obtain the analytical solution to lateral buckling of pipelines with a deformable foundation.

In 2013, a new interpretation of localization was given based on an exponential restoring

Global Buckling of Offshore Pipelines,

Fig. 13 Perturbation solution for $m = 2$



force model in the form of $F = Q_0 \left[1 - e^{\left(\frac{v_0 - v}{\Delta} \right)} \right]$. The effect of initial imperfection on buckling behavior is also studied by analytical and numerical methods (Karampour et al. 2013).

Zhu et al. (2015) proposed a new approach for identifying the nonlinear behavior of pipelines under thermal loading. The lateral soil resistance was modeled by a hyperbolic function in their research, which didn't consider the influence of breakout resistance on the lateral buckling of unburied subsea pipelines.

Conclusions

Global buckling of an offshore pipeline operated at HP/HT can be described as a bar in compression. There are two main types of global buckling of submarine pipelines: lateral buckling and vertical buckling (named as upheaval buckling in some literature). A number of papers of analytical studies have been published on both the upheaval and lateral buckling of pipelines. According to Timothy C. Maltby (1992) and Miles (1998), the global buckling described in these published papers can be summarized into the global buckling of pipeline on rigid foundation (rigid-base model) and that on deformable foundation (deflection-dependent friction model).

Literature reviews imply that there are two main types of buckling model presented in the published material. The two types of buckling model stem from distinctly different assumptions. The rigid-base models take a buckle which has already been formed as a starting point and then attempt to find conditions for which, theoretically, the buckle occurs. In this model, the buckling configuration of the pipeline is assumed in advance. The soil resistance to pipeline is modeled by Coulomb friction coefficient regardless of the change of pipeline-soil interaction during pipeline displacement. On the one hand, the rigid-base model provides a useful model of the global buckling of pipelines as it enables the stable and unstable buckling configurations of pipeline to be calculated. On the other hand, the rigid-base model is unsatisfactory for the analysis of pipeline buckling, in practical engineering, due to the following two defects: The rigid-base model is unable to tell us anything about how buckling actually occurs since the buckling mode is assumed in advance. The Coulomb friction force in the rigid-base analyses is unable to reflect the relationship between the forces exerted on the pipeline by the soil and the pipeline displacement.

The deformable foundation models focus on the conditions when buckling occurs from the initial, possibly imperfect, shape. An important phenomenon in the case of a beam with multiple

imperfections is localization, where one imperfection grows at the expense of others; this is associated with a peak in the axial compressive force in the beam. In this model, the buckling configuration of the pipeline is not fixed and varies with the buckling force. The soil resistance to pipeline is modeled by deflection-dependent function of the deformable foundation. To some extent, the elastic foundation model overcomes the two main shortcomings of the rigid-base model. However, it involves increasingly complex numerical solutions which allow for various nonlinearities in the system but which appear to produce a broadly similar type of result to the standard rigid-base analysis.

The industrial studies demonstrate that there are significant problems associated with trying to obtain a single number for a coefficient of friction to be used either in design or in analysis. Behavior of pipelines on different soils is very complicated and is often different in the lateral and axial directions. Combined with the fact that a submarine pipeline is laid imperfectly (the imperfection sizes being unknown), an analysis of submarine pipelines which is simple, yet correlates well with real pipeline data, has proven to be a real challenge.

Cross-References

► Installation of Offshore Pipelines

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Global Navigation Satellite System (GNSS)

► Integrated Navigation

Global Navigation Satellite System (GNSS) Buoy

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Synonyms

GIS: Geographic information system; GNSS: Global Navigation Satellite System; GPS: Global Positioning System

Definition

A GNSS Buoy is defined as a floating buoy equipped with a geodetic GNSS receiver and one or more antennae near-shore capable of measuring its position and velocity, and hence the instantaneous sea surface height can be attained. GNSS stands for Global Navigation Satellite System, and is the standard generic term for satellite navigation systems that provide autonomous geospatial positioning with global coverage

(<https://www.semiconductorstore.com/blog/2015/What-is-the-Difference-Between-GNSS-and-GPS/1550>). GNSS includes constellations of Earth-orbiting satellites that broadcast their locations in space and time, of networks of ground control stations, and of receivers that calculate ground positions by trilateration.

Scientific Fundamentals

The most prominent feature of a GNSS Buoy is the equipment of GNSS receiver and antenna with which the absolute three-dimensional position of the buoy, as a function of time, can be acquired, and thus the GNSS buoy is able to provide the time series of absolute water level height as well as its horizontal coordinates. So it is necessary to introduce GNSS before the working principle of GNSS buoy is introduced.

GNSS Systems

GNSS is a term used worldwide and includes several global and some regional systems. Common GNSS Systems are GPS, GLONASS, Galileo, Beidou, and other regional systems. The advantage to having access to multiple satellites is accuracy, redundancy, and availability at all times. Though satellite systems do not often fail, if one fails GNSS receivers can pick up signals from other systems. Also if line of sight is obstructed, having access to multiple satellites is also a benefit (<https://www.semiconductorstore.com/blog/2015/What-is-the-Difference-Between-GNSS-and-GPS/1550>).

GPS was the first GNSS system. GPS was launched in the late 1970s by the United States Department of Defense. It uses a constellation of 27 satellites, and provides global coverage. GLONASS is operated by the Russian government. The GLONASS constellation consists of 24 satellites and provides global coverage. Galileo is a civil GNSS system operated by the European Global Navigation Satellite Systems Agency (GSA). Galileo will use 27 satellites with the first Full Operational Capability (FOC) satellites being launched in 2014. The full constellation is planned to be deployed by 2020. BeiDou is the

Chinese navigation satellite system. The system will consist of 35 satellites. A regional service became operational in December of 2012. BeiDou will be extended to provide global coverage by end of 2020 (<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-1/>).

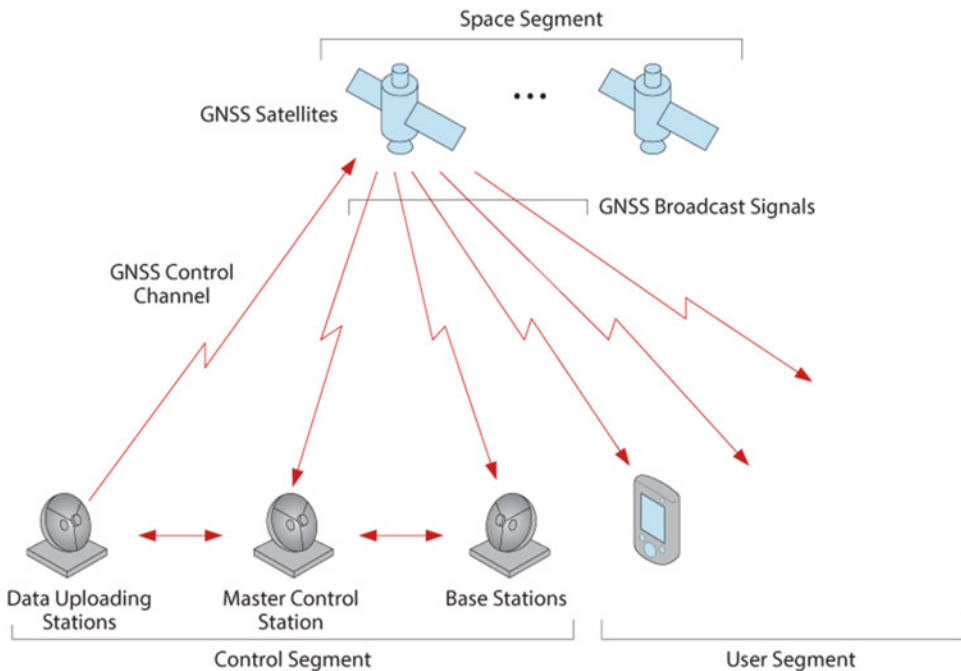
GNSS Architecture

GNSS satellite systems consist of three major components or “segments”: space segment, control segment, and user segment. These are illustrated in Fig. 1 (<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-1/>).

The space segment consists of GNSS satellites, orbiting about 20,000 km above the earth. Each GNSS has its own “constellation” of satellites, arranged in orbits to provide the desired coverage, as illustrated in Fig. 2. Each satellite in a GNSS constellation broadcasts a signal that identifies it and provides its time, orbit, and status.

The control segment comprises a ground-based network of master control stations, data uploading stations, and monitor stations; in the case of GPS, two master control stations (one primary and one backup), four data uploading stations, and 16 monitor stations, located throughout the world. In each GNSS system, the master control station adjusts the satellites’ orbit parameters and onboard high-precision clocks when necessary to maintain accuracy. Monitor stations, usually installed over a broad geographic area, monitor the satellites’ signals and status, and relay this information to the master control station. The master control station analyses the signals and then transmits orbit and time corrections to the satellites through data uploading stations.

The user segment consists of equipment that processes the received signals from the GNSS satellites and uses them to derive and apply location and time information. The equipment ranges from smartphones and handheld receivers used by hikers, to sophisticated, specialized receivers used for high-end survey and mapping applications.



Global Navigation Satellite System (GNSS) Buoy, Fig. 1 GNSS segments(<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-1/>)

Global Navigation Satellite System (GNSS) Buoy,

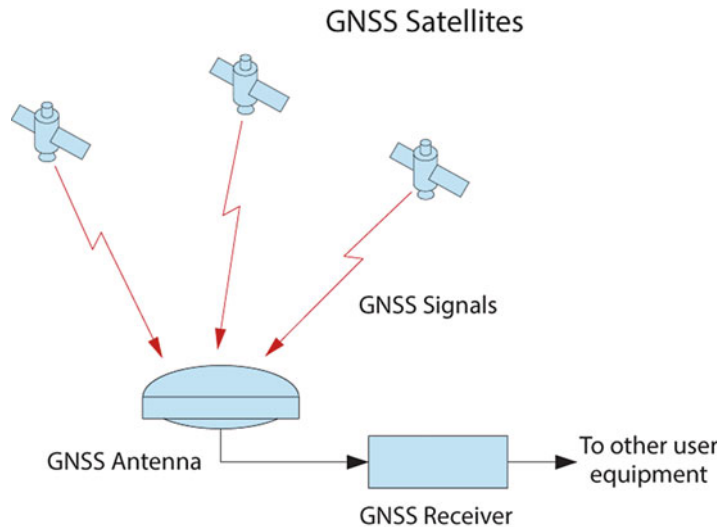
Fig. 2 GNSS satellite orbits (<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-1/>)



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Global Navigation Satellite System (GNSS) Buoy,

Fig. 3 GNSS user equipment (<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-3/>)



User Equipment

The primary components of the GNSS user segment are antennas and receivers, as shown in Fig. 3 (<https://www.novatel.com/an-introduction-to-gnss/chapter-1-gnss-overview/section-3/>). Depending on the application, antennas and receivers may be physically separate or they may be integrated into one assembly. The GNSS equipment of antenna and receiver make GNSS buoy distinguished from other type of buoys.

GNSS antennas receive the radio signals that are transmitted by the GNSS satellites and send these signals to the receivers. GNSS antennas are available in a range of shapes, sizes, and performances. The antenna is selected based on the application. While a large antenna may be appropriate for a base station, a light weight, low-profile aerodynamic antenna may be more suitable for aircraft or unmanned aerial vehicles (UAV) installations.

GNSS receivers process the satellite signals recovered by the antenna to calculate position and time. Receivers may be designed to use signals from one GNSS constellation or from more than one GNSS constellation. Receivers are available in many form factors and configurations to meet the requirements of the varied applications of GNSS.

GNSS Positioning

GNSS positioning is based on a process called “trilateration.” Simply put, if you do not know your position, but do know your distance from three known points, you can trilaterate your location. A GNSS receiver knows where each satellite is in its orbit and compares it with the time required to receive each satellite’s signal. The receiver uses these measurements to calculate its specific position on Earth.

A GNSS receiver can only track satellites orbiting above the horizon. Typically there are between 6 and 12 satellites visible above the horizon at any one time. The receiver will try to track all visible satellites. If some satellites become blocked or “shaded” by tall buildings or other major obstacles, the receiver automatically will try to reacquire the blocked signals. Although a GNSS receiver needs at least four satellites to provide a three-dimensional solution (latitude, longitude, and altitude), it can maintain a (latitude-longitude) position using three satellites (<https://hemispheregnss.com/Resources-Support/About-GNSS-Technology>).

GNSS constellations are designed to provide worldwide positioning services with an accuracy ranging from 5 to 15 meters. More precise accuracies are not possible with standard GNSS due to minor timing errors and satellite orbit errors, plus atmospheric conditions that affect the signals and their arrival time on Earth. However, there are ways to improve GNSS accuracy using additional services. There are four primary services available, each capable of improving position accuracies to better than one meter (<http://www.unoosa.org/oosa/en/ourwork/psa/gnss/gnss.html>):

- Radiobeacon differential GPS (DGPS) corrections.

- Space-Based Augmentation Systems (SBAS), including the United States’ Wide Area Augmentation System (WAAS), Europe’s EGNOS, Japan’s MSAS, and India’s GAGAN.
- Privately owned “L-band” satellites.
- Real-time kinematic (RTK) positioning.

Among the above services, the RTK is a technique that uses carrier-based ranging and provides ranges (and therefore positions) that are orders of magnitude more precise than those available through code-based positioning. RTK techniques are complicated. The basic concept is to reduce and remove errors common to a base station and rover pair, as illustrated in Fig. 4 (<https://www.novatel.com/an-introduction-to-gnss/chapter-5-resolving-errors/real-time-kinematic-rtk/>).

Working Principle of a Typical GNSS Buoy

In this section, we will take a recent example of a GNSS buoy of Taiwan for monitoring the water surface elevation to show the working principle of a general GNSS buoy. The buoy can observe water surface elevations and provide real-time tide and wave data in estuaries and coastal areas without establishing an RTK reference station.

Instrumentation

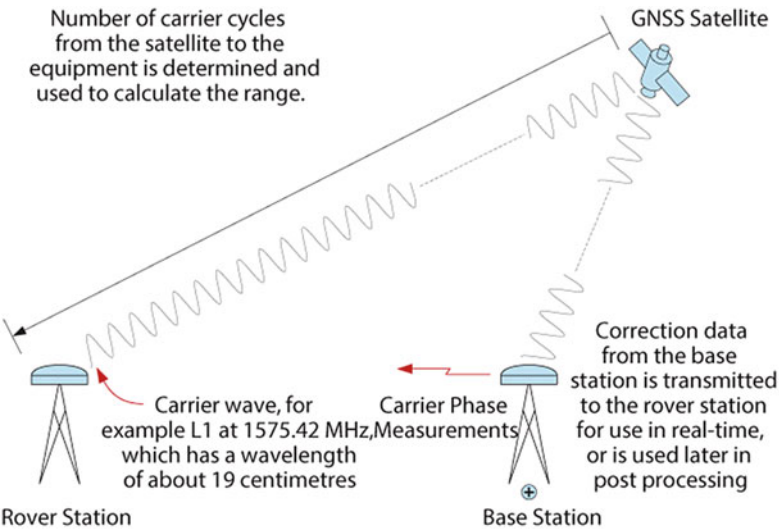
Figure 5 shows the working principle of the GNSS buoy. It can be seen that the buoy utilizes a Virtual Base Station (VBS) combined with the RTK positioning technology and the buoy includes a buoy hull, an RTK GNSS receiver, data-transmission devices, a data logger, and General Purpose Radio Service (GPRS) modems for transmitting data to the desired land locations. The GNSS receiver receives signals from satellites and calculates positions. The position data are then transmitted to the VBS-RTK control center by a GPRS modem in a GGA sentence that includes, among other information, the GPS time, latitude, and longitude. Figure 6 presents the snapshot of the GNSS buoy deployed in the waters of Suao, Taiwan Lin et al. (2017).

Methodology

In the buoy development, the VBS-RTK positioning technology was adopted to determine the

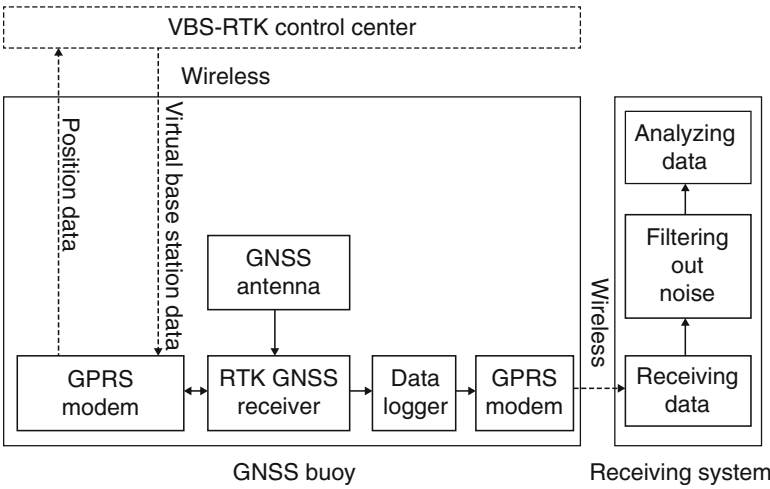
Global Navigation Satellite System (GNSS) Buoy,

Fig. 4 Real-time kinematic positioning (<https://www.novatel.com/an-introduction-to-gnss/chapter-5-resolving-errors/real-time-kinematic-rtk/>)



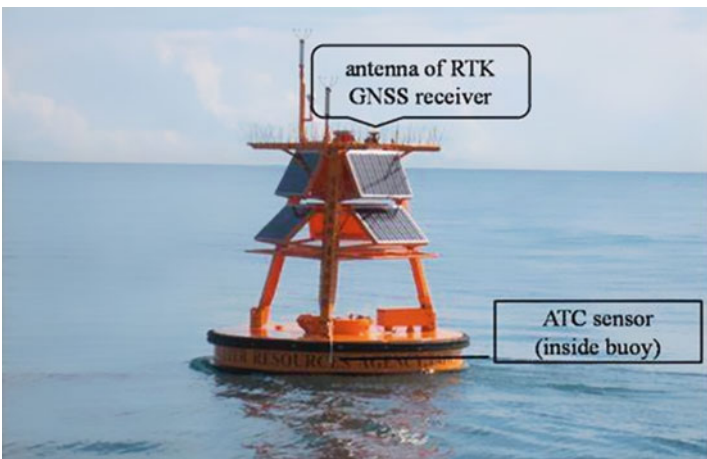
Global Navigation Satellite System (GNSS) Buoy,

Fig. 5 Working principle of a GNSS buoy for monitoring the water surface elevation Lin et al. (2017)



Global Navigation Satellite System (GNSS) Buoy,

Fig. 6 Snapshot of the GNSS buoy deployed in the waters of Suao, Taiwan Lin et al. (2017)



position of the buoy. VBS-RTK stands for Virtual Base Station Real-Time Kinematics. The procedure of VBS positioning includes pre-process network observations, calculating data from regional stations, generating VBS data for the rover, and calculating the coordinate of the rover station. The VBS-RTK system consists of three components: a GNSS base station network, control center, and rover station. Their functions are as follows Lin et al. (2017):

1. GNSS base station network: Each base station receives GNSS observation data and transmits raw data to the control center continuously.
2. Control center: The VBS-RTK control center is for positioning computation and the commercial software developed by Trimble Navigation is run in the center. The functions of the software are as follows:
 - Connecting the control center and each reference station to enable the automatic receipt, storage, and compression of observations from each reference station.
 - The software not terminating the receipt or compression of satellite signals from reference stations during data download.
 - Monitoring the status of the GNSS receiver of each reference station. The GNSS receiver parameters may be configured including the cutoff angle and sampling interval, etc.
 - According to the carrier phase observations, the software calculates continuously the error caused by the multipath, the ionosphere, troposphere, and ephemeris, as well as the integer ambiguity of the carrier phase of L1 and L2.
 - Generating VBS data in Radio Technical Commission for Maritime Services (RTCM) format and transmitting them to the rover station.
3. Rover station: The rover station is a GNSS buoy with a GNSS receiver and a GNSS antenna attached to it.

Thus, the water surface elevation can be obtained from the position data of the GNSS buoy.

Key Applications

GNSS Buoy Application Fields

The applications of GNSS Buoy systems include absolute calibration of satellite altimeters, observing oceanic phenomena such as water surface height, coastal circulation, ocean tide studies, and other coastal applications. For example, in the area where the GNSS buoy system is used to calibrate satellite altimeters, its size could range from a small life-saver type with the receiver and power supply on the tethered boat to big, ruggedized types that accommodate all of the equipment as well as the other sensors providing its orientation and meteorological and auxiliary data for long-term deployment. It could require dedicated personnel for a campaign style design while it could also be an autonomous design that transmits its observations automatically to the base station in the vicinity. GNSS buoys have been implemented for altimeter in the past showed the equivalent ssh measurements between a waverider and a spar design for altimeter absolute calibration.

For the observations of oceanic phenomena, GNSS buoys have demonstrated in bathymetry mapping, in wave height observations and sea state determination, and in wave height spectrum analysis. It can also be used to determine absolute sea surface and mean lake surface height gradient.

In addition, the GNSS buoy can be used in geodetic applications in the coastal region. For example, a height modernization project performed by the National Geodetic Survey (NGS) seeking the use of GPS data on land and oceans to improve the determination of local geoid height and thus to promote GPS applications on measuring elevation, which is traditionally done by spirit leveling. With the continuous improvement of GNSS measurement processing capabilities and computer processing power, GNSS buoys can now be used in oceanography, hydrography, tide gauging, and more Cheng (n.d.).

Examples of French GNSS Buoys

The use of GNSS buoys in the field of oceanography began in the 1990s. Australia, the USA, and France were among the pioneering countries (<http://refmar.shom.fr/documentation/>

[instrumentation/bouee-gnss](#)). In France there are four types of GNSS buoys in use today.

OCA-Géoazur/CNES Buoy

The GNSS Buoy at the Cote d'Azur Tide Station/ Centre National d'Etudes Spatiales (CNES) was the first to be installed in France. By the late 1990s, measurement campaigns were regularly conducted off Ajaccio in order to calibrate the

satellite altimetry missions of Topex-Poseidon and Jason-1 and Jason-2.

IPGP Buoy

The GNSS buoy of the Institut de Physique du Globe de Paris (IPGP) is substantially identical to the OCA-Géoazur/CNES buoy. But it is used for quite a different purpose. The GNSS buoy provides data that are compiled with other data, including pressure measurements on the seabed, in order to study the vertical deformation of the seabed related to seismic activity. Today we cannot globally map (in the marine environment) deformations of the Earth based on satellite signals alone. Ongoing studies aim to circumvent this problem. IPGP buoy is shown in Fig. 7 (<http://refmar.shom.fr/documentation/instrumentation/bouee-gnss>).



Global Navigation Satellite System (GNSS) Buoy, Fig. 7 IPGP buoy (<http://refmar.shom.fr/documentation/instrumentation/bouee-gnss>)

LEGOS/DT-INSU Buoy

The GNSS buoy of the Laboratoire d'Études en Géophysique et Océanographie Spatiales (LEGOS) built by the Institut National des Sciences de l'Univers (INSU) consists of three floats composing the vertices of an equilateral triangle, with the GNSS antenna placed at center of gravity. The LEGOS/DT-INSU GPS buoy is used to monitor instrument drift on tide gauges composing the ROSAME network. Because the ROSAME stations are located in the French Southern and Antarctic Lands, there are strict specifications on the weight and size of the unit. Figure 8 shows the picture of LEGOS/DT-INSU buoy.

Global Navigation Satellite System (GNSS) Buoy,

Fig. 8 LEGOS/DT-INSU buoy (<http://refmar.shom.fr/documentation/instrumentation/bouee-gnss>)





Global Navigation Satellite System (GNSS) Buoy, Fig. 9 SHOM buoy (<http://refmar.shom.fr/documentation/instrumentation/bouee-gnss>)

SHOM Buoy

The SHOM buoy is closer in shape to the LEGOS/DT-INSU buoy, with a base forming an equilateral triangle with a float at each end. However, it is larger and forms a tetrahedron with the geodetic antenna placed at the top of the last vertex, out of the water and up in the air.

The two models do not have the same constraints. The SHOM buoy seeks to achieve floating stability vis-à-vis the chop and swell. The GNSS antenna is raised above the waterline to avoid multipath as much as possible. Multipath produces adverse reflection effects during the acquisition. The SHOM buoy is shown in Fig. 9.

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Website: <https://hemispheregnss.com/Resources-Support/About-GNSS-Technology> (cited on 10/09/2018)

Global Positioning System (GPS)

► Doppler Velocity Log for Navigation System in Underwater Vehicle

GM, Grid Method

► Numerical Simulation of Ice-Going Ships

GNSS: Global Navigation Satellite System

- [Global Navigation Satellite System \(GNSS\) Buoy](#)

Gouging

- [Iceberg Scouring](#)

GPS: Global Positioning System

- [Global Navigation Satellite System \(GNSS\) Buoy](#)

Gravity-Based Structures

- [Shallow Foundations](#)

Green Energy

- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

Green Macroalgal Blooms

- [Green Tides](#)

Green Seaweed Tides

- [Green Tides](#)

Green Tides

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Synonyms

[Green macroalgal blooms](#); [Green seaweed tides](#)

Definition

Green tides are ecological disasters in which massive biomass of one or multiple green seaweed species accumulates in either shallow coastal or open waters, forming drifting canopies and harming the local environment and ecosystem.

Scientific Fundamentals

Green tides are vast accumulations of unattached green macroalgae in either coastal or open waters, causing significant detrimental environmental impacts. As a type of harmful algal blooms (HABs), green tides differ from the other HABs (e.g., red tides, brown tides, and golden tides) obviously with the blooming species, which are predominated by one or multiple green macroalga species. Green microalgae or cyanobacteria blooms (e.g. microcystis blooms in Taihu Lake, China), although, in green color, are blooms of single-cell microalgae, which do not fall into the category of green tides. The green tide is increasing globally, especially along the coasts of America, Europe, and the Asia-Pacific, since the 1970s

(Ye et al. 2011 and references herein). The massive green seaweed thalli detached from the natural seaweed beds are proliferating and accumulating in the eutrophied estuaries, coasts, and bays nearby. They usually comprise multiple ephemeral green macroalgal taxa and even a few brown or red seaweed species and show evident species succession during blooming. These green tides (named as “local green tides” to differentiate with the “trans-regional green tides” below) are restricted in the shallow photic zone of the near-shore water (Fig. 1a) and barely intrude into the open waters.

In 2008, astonishing large-scale drifting canopies of *Ulva prolifera* inundated along the coasts of Qingdao, threatening the sailing regatta of the Beijing Olympics. Satellite remote-sensing indicated that the drifting green mats were widespread throughout the southwestern Yellow Sea (YS) and were probably initiated from the southwestern coast of YS. The large-scale green tide in YS (GTYS) recurred since then and is recognized as the world’s largest green tide in terms of its distribution, coverage, and total biomass produced (Fig. 1b). Compared to the local green tides, GTYS is featured by a single causative species (*U. prolifera*), nonlocal origin of the floating biomass, long-distance transportation, and the cross-regional impacts (Liu and Zhou 2018).

Blooming Species

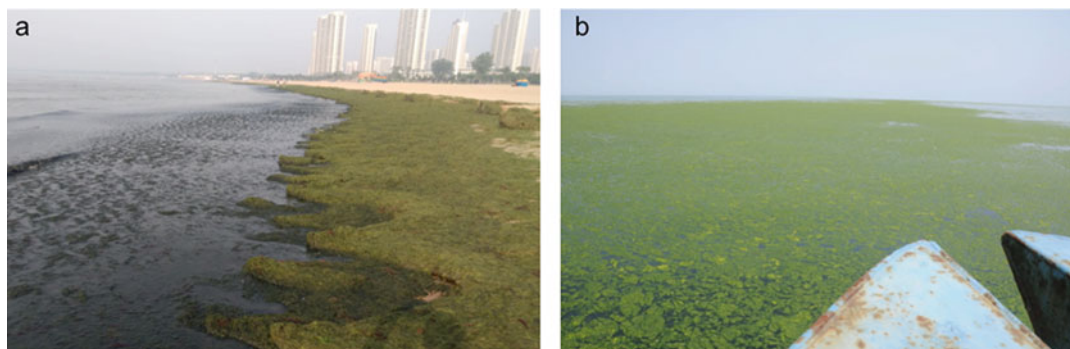
The causative species of the green tides are quite diverse (Table 1). Although some green tides

comprised a few red (Division Rhodophyta) and/or brown (Division Phaeophyta) macroalgae species, the blooming biomass is usually dominated by the green macroalgae (Chlorophyta, Ulvophyceae) and mostly shows green color (which the green tide is named after). A high number of bloom-forming species belong to the genus *Ulva*, including the former *Enteromorpha*. The genus *Ulva* is species-rich, comprising over 100 species. They are cosmopolitan and could be found in almost every littoral waters. The *Ulva* spp., including *U. prolifera*, *U. intestinalis*, and *U. australis* (reported as *U. pertusa*), formed nuisance green tides along the coasts of Europe, N. and S. America, Australia, and Asia. *U. prolifera*, for example, is responsible for the large-scale green tide in YS, which has been persisting for over a decade. *Ulva* spp. also dominated the green tides in Brittany, France, and the Venice Lagoon, Italy. A number of other taxa, such as *Gracilaria* (Rhodophyta; Gracilariaceae) and *Pylaiella* (Ochrophyta, Phaeophyceae) etc., are also involved in massive green tide events worldwide (Table 1).

Blooming Dynamics

Causes

Abiotic Factors Among the various environmental factors, eutrophication is the most important cause for the green tides. The increasing green tides in the temperate zone of N. America, Europe,



Green Tides, Fig. 1 Local green tide in Qinhuangdao, Bohai Sea, on June 17, 2017 (a, courtesy of Dr. Wei Song), and drifting mat of *Ulva prolifera* in open water of western YS on June 14, 2012 (b, courtesy of Dr. Shiliang Fan)

Green Tides, Table 1 List of the common green-tide-forming macroalgae species^a

Phylum, class	Family	Species	Blooming locations	Reference
Chlorophyta, Ulvophyceae	Ulvaceae ^b	<i>Ulva prolifera</i>	Yellow Sea and Bohai Sea, China	Leliaert et al. 2009; Song et al. 2019
		<i>Ulva Australis</i> (reported as <i>U. pertusa</i>)	Tokyo Bay, Japan; Bohai Sea, China	Yabe et al. 2009; Song et al. 2019
		<i>Ulva ohnoi</i>	Tokyo Bay, Japan	Yabe et al. 2009
		<i>Ulva intestinalis</i>	Baltic Sea, Finland; Maasholm Bay, Germany; Mondego estuary, Portugal; S. California, USA	Bäck et al. 2000; Lotze et al. 2000; Kamer et al. 2001; Cardoso et al. 2004; Wrom and Lotze 2006
		<i>Ulva expansa</i>	S. California, USA	Kamer et al. 2001
		<i>Ulva lactuca</i>	Washington State and Massachusetts, USA	Nelson et al. 2008; Teichberg et al. 2008
		<i>Ulva compressa</i>	Langstone Harbour, S. England	Taylor et al. 2001
		<i>Ulva linza</i>	Langstone Harbour, S. England	Taylor et al. 2001
		<i>Ulva curvata</i>	Langstone Harbour, S. England	Taylor et al. 2001
		<i>Ulva rigida</i>	Langstone Harbour, S. England	Taylor et al. 2001
		<i>Ulva</i> spp.	Venice Lagoon, Italy; NW Black Sea; Baltic Sea; Laholm Bay, Sweden; Bay of Cádiz, Spain; Brittany, France; Peel Inlet and Great Barrier Reef, Australia; Tuggerah Lakes Estuary, New South Wales; Coos Bay, Branford River, San Francisco Bay and Waquoit Bay, U.S.; Gulf of California, Mexico; Nuevo Gulf, Patagonia; Seto Inland Sea, Japan	Fletcher 1996; Valiela et al. 1997; Vahteri et al. 2000; Nelson et al. 2003; Charlier et al. 2007; Sugimoto et al. 2007; Nelson et al. 2008; Corzo et al. 2009; Teichberg et al. 2010; Ye et al. 2011
		<i>Ulvaria obscura</i>	Washington State, USA	Nelson et al. 2003, 2008
		<i>Percursaria percursa</i>	Langstone Harbour, S. England	Taylor et al. 2001
	Siphonocladaceae	<i>Dictyosphaeria</i> spp.	Kanehoe Bay, Hawaii of USA	Valiela et al. 1997; Teichberg et al. 2010
	Caulerpaceae	<i>Caulerpa</i> spp.	N. Mediterranean coast	Valiela et al. 1997;
	Cladophoraceae	<i>Cladophora glomerata</i>	Archipelago of SW Finland, N. Baltic Sea, Finland	Vahteri et al. 2000; Berglund et al. 2003
		<i>Cladophora vagabunda</i>	Waquoit Bay of MA, USA	Hauxwell et al. 2001
		<i>Cladophora dalmatica</i>	Langstone Harbour, S. England	Taylor et al. 2001
		<i>Cladophora</i> spp.	N. Mediterranean coasts; southern Fyn archipelago,	Valiela et al. 1997; Eriksson et al. 2009;

(continued)

Green Tides, Table 1 (continued)

Phylum, class	Family	Species	Blooming locations	Reference
			Denmark; NW Black Sea; Baltic Sea; Laholm Bay, Sweden; Peel Inlet and Great Barrier Reef, Australia; Bermuda; Great Lakes, Hog Island and Waquoit Bay, USA; Gulf of California, Mexico	Teichberg et al. 2010 ; Ye et al. 2011
	Monostromataceae	<i>Monostroma</i> spp.	Arcachon Bay, France; W. Baltic Sea, Netherland	Eriksson et al. 2009 ; Ye et al. 2011
	Cladophoraceae	<i>Chaetomorpha</i> spp.	Peel-Harvey Estuary, W. Australia; Langstone Harbour, S. England	Taylor et al. 2001 ; Ye et al. 2011
		<i>Rhizoclonium tortuosum</i>	Langstone Harbour, S. England	Taylor et al. 2001
	Codiaceae	<i>Codium isthmocladum</i>	SE Florida, Hog Island, USA; Bermuda	Teichberg et al. 2010 ; Thomsen et al. 2006 ; Charlier et al. 2007
		<i>Codium fragile</i>	Bohai Sea, China	Song et al. 2019
Rhodophyta, Florideophyceae	Gracilariaceae	<i>Gracilaria tikvahiae</i>	SE Florida, Waquoit Bay of MA, Hog Island, USA;	Hauxwell et al. 2001 ; Charlier et al. 2007 ; Teichberg et al. 2008
		<i>Gracilariopsis lemaneiformis</i> (reported as <i>Gracilaria lemaneiformis</i>)	Bohai Sea, China	Song et al. 2019 ;
		<i>Gracilaria</i> spp.	Venice Lagoon, Italy; Mondego estuary, Portugal; Brittany, France; Gulf of California, Mexico; Waquoit Bay, USA;	Valiela et al. 1997 ; Charlier et al. 2007 ; Teichberg et al. 2010 ; Cardoso et al. 2004
	Rhodomelaceae	<i>Yuzurua poiteaui</i> (reported as <i>Laurencia poitei</i>)	SE Florida, USA; Bermuda	
Ochrophyta, Phaeophyceae	Acinetosporaceae	<i>Pylaiella littoralis</i> (reported as <i>Pilayella littoralis</i>)	Maasholm Bay, Germany; Massachusetts Bay, USA; N. Baltic Sea, Finland	Valiela et al. 1997 ; Lotze et al. 2000 ; Berglund et al. 2003
		<i>Pylaiella</i> spp. (reported as <i>Pilayella</i> spp.)	Nahant Bay, USA; Maasholm Bay, Germany	Valiela et al. 1997 ; Teichberg et al. 2010

^aThe nomenclature of taxa is based on WoRMS (<http://www.marinespecies.org/index.php>, accessed on June 5th, 2021)

^bThe genus *Ulva* included the former *Enteromorpha* (Hayden et al. 2003)

and Asia are coincident with the rapid population growing and urbanization along these coasts. The significant nutrient input (especially N) from river discharge, wastewater sewage system, and costal aquaculture has driven the coastal eutrophication, which causes a series of ecological changes (including green tides). The green tide in Finland, for example, continued every summer between 1992 and 2000, and the Gulf of Finland is one of the most heavily eutrophied areas in the Baltic Sea, receiving 138 million kg nitrogen and 7.6 million kg phosphorus annually (Lappalainen and Pönni 2000).

Physiological advantages of these green tide species could boost the green tides dominating by one or a few ephemeral species under the general eutrophication environment. Both field and laboratory experiments revealed multiple physiological advantages of these bloom-forming species (Table 1), such as high nutrient assimilation, fast photosynthesis and growth, and strong buoyancy ability, compared to the non-bloom-forming taxa (Teichberg et al. 2008; Fu et al. 2019). These physiological traits assist the rapid tissue expansion and biomass proliferation and hence dominate the green tide in response to the increased nutrient supplies. The selective pressure in green tide areas leads to the amplification of *Ulva* genotypes best adapted for this environment (Fort et al. 2020). This also explains why the green tides comprise only a few species, while higher species diversity is usually accounted for the benthic macroalgae community. Such selection of fast-growing strains indicates that green tides are likely to persist in the near future if the eutrophic condition wouldn't change.

The other abiotic factors, such as light availability, salinity, and seawater temperature, may also be involved in determining species composition of the green tide at specific locations. For example, the green tides in Washington State, USA, are dominated by *Ulva* spp. in intertidal zone, while *Ulvaria* is abundant in subtidal. Physiological comparison shows that *Ulvaria* is relatively euhaline, so it is abundant in the subtidal water, where the salinity is more stable, while absent or less common in the lower-salinity portions of estuaries (Nelson et al. 2008).

Biotic Factors Besides the various environmental factors, a number of biotic factors, such as propagule bank, grazers, etc., could also regulate the population development and dominance patterns of the green tides. Propagules are microscopic forms of macroalgae, which function as “seed banks” (like terrestrial plants) for the recruitment in the following vegetation period (Lotze et al. 2000). The propagule bank allows macroalgal populations to persist through unfavorable conditions and sudden disturbances in a variable environment. Although the propagule bank is the important overwinter mechanism for macroalgae, the abundance of recruitments may not directly correlate with the quantity of propagules since the environmental conditions could interact with propagule source and regulate the abundance and species composition of recruitments. For example, the field experiments in Baltic Sea showed that the rocky substrata favor recruitments of *Ulva* instead of *Pylaiella* (reported as *Pilayella*) when propagules of both species are present (Lotze et al. 2000).

Grazers or disturbance on trophic cascade is another important biotic component in controlling the abundance and community structure of littoral macroalgal. As indicated by a series of field observations, abundance herbivores at lower trophic level is directly linked to the density and community structure of the benthic macroalgae through selective consumption (Lotze et al. 2000). This top-down effect could be more complicated when higher trophic levels are taken into account and various interactions are involved. In Baltic Sea, for example, abundance of piscivorous fish had a strong negative correlation with the bloom-forming macroalgae, while depleted top-predators displayed massive increases of ephemeral algae (Eriksson et al. 2009). The following field experiments indicated that excluding larger piscivorous fish resulted in increased abundance of small-bodied predatory fish, decreased the smaller gastropod scrapers, and ultimately increased the abundance of ephemeral macroalgae (Eriksson et al. 2009).

Anthropogenic Disturbance The influencing factors listed above are somehow related to the

anthropogenic disturbances (e.g., terrestrial nutrient input, overfishing, etc.), representing the profound but indirect impacts from the anthropogenic activities on the coastal ecosystems. More recent studies revealed that anthropogenic activities in the coastal regions may directly contribute for or stimulate the biomass expansion and bloom development of green tides. The best case is the large-scale green tide in YS (GTYS). A series of research have confirmed that the substantial green macroalgal wastes disposed from the *Pyropia* aquaculture rafts are the primary source of the initial floating macroalgae, which is transported out of the original region and latterly develops into large-scale green tide in western YS (Wang et al. 2015). Although various biochemical and geophysical factors, such as high photosynthetic and growth rates of *U. prolifera*, high nutrient level of the seawater, northward seasonal monsoon and surface circulation, etc. are involved in the bloom development, the anthropogenic activities (detaching and disposing the green macroalgae) are considered to be the direct culprit stimulating detachment of green macroalgae, which subsequently drift and develop into the million tons of floating biomass with favorable environmental conditions (Zhou et al. 2015).

The cause for each individual green tide is actually quite complicated and distinct among the green tides in different regions. Not a single factor alone could directly stimulate the green tides, especially those persisting over the years. Multiple biotic and abiotic factors often interact with each other to regulate the abundance and structure of the coastal macroalgal communities and hence drive the continuous green tides. Beside the eutrophication prevalent along the coasts globally, specific biotic or anthropogenic disturbances (e.g., aquaculture, loss of predatory, etc.) may trigger the cascade effects to boost or enhance the green macroalgal blooms.

Seasonality

Due to the significant temperature variation at different latitudes, bloom season of the green tides could vary from early spring to early summer, but they usually occur when the sea surface temperature increases rapidly and persist for 1 to

4 months. For example, drifting bands of *U. prolifera* in YS could be observed as early as April in Subei Shoal based on the shipboard field survey. The floating biomass then drifts out of the shoal and is widespread in western YS. The green tides are quite persisting along the coasts of Shandong Province and even last until early September in 2019. The duration of the green tides in YS could reach to over 100 days based on the remote-sensing (MNR, 2008–2020), which may be underestimated since satellite remote-sensing can barely detect the floating bands in the turbid Subei Shoal.

The environmental parameters of the blooms, such as seawater temperature, salinity etc., vary significantly depending on the blooming species and geographic locations. It often lies within the range favored by the blooming species. Most *Ulva* spp. bloom between 15 °C and 25 °C along the temperate coasts, while some species (e.g., *U. intestinalis*) could bloom once the ice melts in the N. Baltic Sea (around 60°N).

Blooming Process

The development processes of the blooms are diversified at different locations. As described above, local green tides often comprised multiple macroalgae species and show evident species succession in response to the environmental gradients (e.g., nutrient, seawater temperature, etc.) at different locations of the blooming region or different stages of the blooms. The green tide in Bohai Sea, for example, started to appear in late April, reached the peak in July and August, and declined rapidly in September. The entire blooming process could be divided into three phases based on the dominant species: I, Low biomass was dominated by *Ulva australis* (reported as *U. pertusa*) from April to mid-May; II, the growing biomass was dominated by the red algae *Bryopsis plumosa* from late May to mid-June; III, green seaweed *U. prolifera* proliferated rapidly since late June and became dominated, and the biomass of *U. prolifera* along with the other two common species *B. plumosa* and *Gracilariopsis lemaneiformis* (reported as *Gracilaria lemaneiformis*) reached the peak until mid-September (Song et al. 2019). Nelson et al. (2008) examined the green tides in

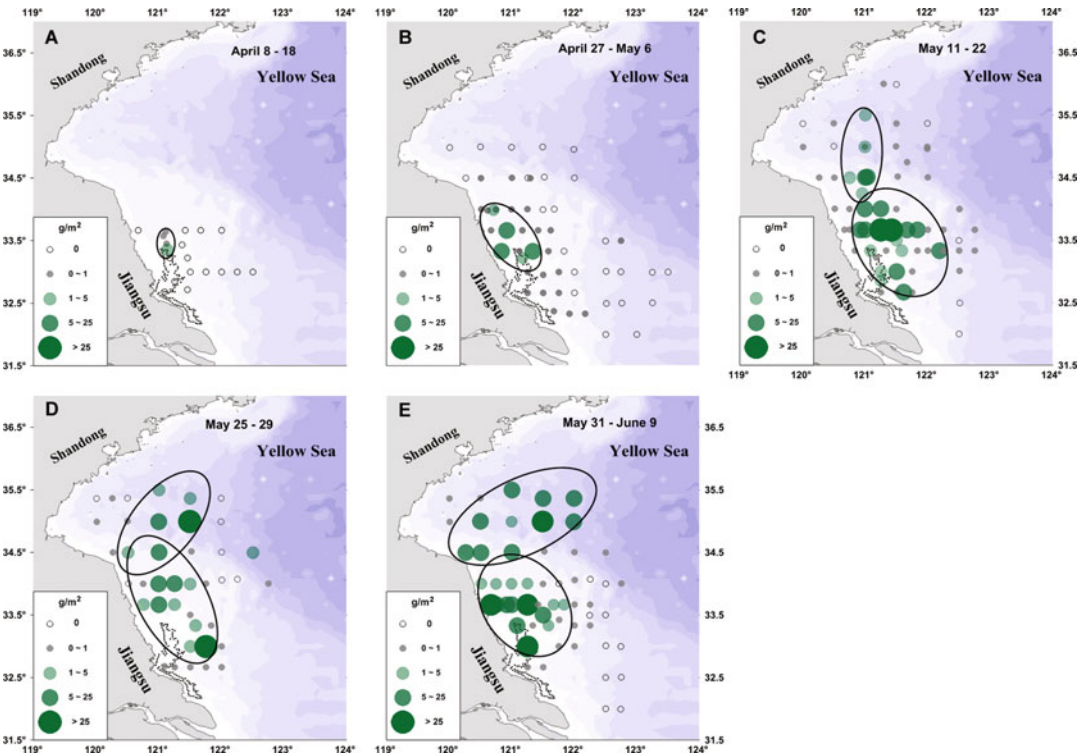
Washington State, USA, and found the blooms are often dominated by *Ulva* spp. intertidally and *Ulvaria obscura* subtidally. The environmental factors, such as nutrient availability, grazers, and light attenuation, influence the species assemblage of the local green tides. Bottom-up factors allow *Ulva* to dominate the intertidal zone, while resistance to grazers appears to allow *Ulvaria* to dominate the subtidal zone.

The genesis and development of the green tide in YS (GTYS) are quite different with the local green tides. The single species, *U. prolifera*, proliferates rapidly once the detached thalli drift into the coastal water and dominates the floating biomass in the Subei Shoal. A series of biological, geochemical, and physical processes are involved in the subsequent transportation and spreading of the floating *U. prolifera* in western YS. Based on the decadal field surveys, the floating green macroalgal bands are initiated in late April to early May in the shallow water of Subei Shoal,

close to the raft region. With the strong buoyancy and growth abilities, *U. prolifera* dominates the floating biomass rapidly with a few sympatric *Ulva* spp. detected sporadically (<10%, Wang et al. 2015). The biomass of floating *U. prolifera* amplifies with increasing seawater temperature and drifts out of the shoal with strong tidal force and sea surface currents in May and June. They develop into large-scale floating mats in open water of western YS (Fig. 2) and further drift northwardly assisted by the seasonal monsoon and the resulted northward surface circulation. As a result, million tons of floating *U. prolifera* biomass reach the south coastline of Shandong Peninsula in late June to early July and could persist for over a month.

Impacts

Tons of green macroalgae are piled up in the shallow coastal waters, smothering the shoreline, harming the coastal aquaculture, and affecting the



Green Tides, Fig. 2 Spatial and temporal development of floating green algal biomass in 2012 based on the shipboard survey. The black ovals denoted the major

distribution and the drifting patterns of the floating biomass within each survey period. (Wang et al. 2015)

local tourism and recreational activities. Significant economic losses directly resulted not only from the large cost for removing the beached biomass and waste treatment but also from the losses of coastal aquaculture. The green tide in YS, for example, caused large mortality of cultured *Apostichopus japonicus*, *Placopecten magellanicus*, etc. along the coasts of Shandong Province in 2008, which resulted in losses of 800 million RMB (Ye et al. 2011). In addition to these evident economic losses and social impacts, persisting green tides have profound and long-lasting effects on the coastal ecosystems.

Although mostly nontoxic to humans, degrading morass of green tides can produce toxic hydrogen sulfide (H_2S) and a few bloom-forming *Ulva* spp. are likely to produce toxins. H_2S gas from rotting *Ulva* biomass on a Brittany Beach caused the death of a horse in 2009 and death of around 30 wild boars in 2011 (Smetacek and Zingone 2013). Some green tide-forming *Ulva* spp. have been confirmed to be allelopathic to *Fucus* zygote, oyster larval, sea urchins, and herbivore invertebrates, which suggested possible toxins from these species (Nelson et al. 2003; Van Alstyne et al. 2007).

A series of deleterious ecological effects included increasing light attenuation, altering nutrient biogeochemical cycling, inducing hypoxia to anoxia at the sediment interface, and hence changing the benthic vegetation and macrofauna. In Waquoit Bay, MA, USA, higher N estuary and the subsequent increase of green macroalgal biomass caused decline of eelgrass *Zostera marina* (Hauxwell et al. 2001). In Åland Island, Finland, drifting macroalgae dramatically altered the abundance and species composition of the macrofauna, with mainly opportunistic and mobile taxa such as *Hydrobia* spp., Chironomidae, and Ostracoda benefiting from the algae, whereas infaunal species such as *Macoma balthica* and *Bathyporeia pilosa* are negatively affected (Norkko et al. 2000).

Key Applications

Monitoring Technologies

Monitoring or surveys are important for understanding the blooming pattern and underlying

mechanism. According to the different objectives of each research project, monitoring could be quite diverse on method, technology, and even subject. Overall, two monitoring approaches are often adopted: on-site survey and remote-sensing monitoring. The former is used to monitor the benthic seaweed beds (source biomass), floating biomass at specific locations, and the various detaching and blooming processes, which could provide the detailed information on abundance, species composition, and the other related biological and environmental conditions contributing for the blooms. The latter is used to detect the drifting macroalgal mats in open waters and currently is the most important approach to estimate the distribution and coverage of the trans-regional green tide in YS.

To evaluate the potential source biomass of the green tides, scuba diving and random sampling are often carried out to quantify the abundance and species composition of the attached green macroalgae. Depending on the various attaching substrata, abundance of the benthic green macroalgae could be quantified as g m^{-1} (for the green macroalgae attached on the ropes and bamboos of aquaculture rafts) and g m^{-2} (for the macroalgae on benthic seaweed beds). Species of green macroalga is identified based on the gross morphology and molecular sequencing. A few rapid identification methods (e.g. RFLP and FISH) are recently developed to assist fast batch identification (Xiao et al. 2013; Zhang et al. 2015). On-site survey on the floating biomass is also important for monitoring the blooming dynamics and evaluating the severity and size of the green tides. The floating biomass could be surveyed through bio-trawling, in which a phytoplankton net is towed through the water column to collect the green macroalgal biomass in the certain volume or area of the seawater. The abundance is then calculated as fresh wet weight of floating biomass per volume or area (g m^{-3} , g m^{-2}).

Remote-sensing is to detect and track the floating green macroalgae through multisource images of imagery instruments, such as optical satellites (e.g., MODIS, GOCI, Landsat TM, HJ-1 A/B-CCD, Gaofen-1, and Sentinel-2), microwave ENVISAT-ASAR, and airborne synthetic-

aperture radar (SAR). Through satellite optical images, the floating macroalgae pixels can be differentiated from the surrounding water and thus delineated using the various vegetation indices, such as NDVI (normalized difference vegetation index), FAI (floating algal index), and SAI (Scaled Algae Index). These technologies provide the time series of distribution and coverage of the floating green macroalgal mats, which are currently popularly used to indicate the size of the large-scale green tide in YS. Distribution is estimated as the smallest geographic area that encloses all floating-algae pixels. The coverage is the cumulative sea surface area pure covered by the floating-algae pixels, which is computed by multiplying the pixel size (e.g., $250 \times 250 \text{ m}^2$ for MODIS and $30 \times 30 \text{ m}^2$ for Landsat TM and HJ-1 CCD) with the number of identified macroalgae pixels. Due the resolution limitations and restrict requirements on weather conditions, these aeronomy remote-sensing technologies are probably not suitable for monitoring the local green tides. But some low-altitude (high-resolution) remote-sensing, such as airborne SAR, unmanned aerial vehicle (UAV), etc., can assist the field surveys on the geographic scale and abundance of the local green tides (Cui et al. 2018; Xing et al. 2019).

Containment Measures and Bioconversion

Control and prevention are complicated since the genesis of each individual green tide involves a series of cascade processes which are influenced by various biogeochemical factors as described above. In response to the eutrophication as the primary cause for the green tides worldwide, measures such as decreasing the nutrient input, increasing water flushing, etc. are requested to alleviate the eutrophication problem and hence abate green tides. These measures, however, require long-term efforts on waste treatments, coastal ecosystem restoration, and balancing the economy development and environment protection. Removal of blooming biomass is absolutely necessary to avoid nutrient pollution and possible toxic effects on coastal waters and benthic organisms. And the proper treatments and

bioconversion are also critical after the algae are harvested; otherwise, the aquatic pollution would turn into a land pollution. The harvested algae can be turned into fertilizer, mixed with waste to be composted, dehydrated and mixed into poultry and aquaculture feed, and processed as food and have sometimes been used as a fuel (Charlier et al. 2007). A series of technologies, such as methanization, hydrolysis, and pressing, are developed to stabilize and assist the efficient bioconversion of the harvested algae. Along with the significant increase of the massive green tides throughout the world, various firms and laboratories are devoted into or even established to transform this gross biomass into commercially valuable bio-resources. They included a number of private companies, such as GOEMAR, SECMA, and PHYTOMER, in Europe and Qingdao Seawin Biotech Group (Seawinner®) in China (Charlier et al. 2007; Ye et al. 2011). The fertilizer line of Seawinner, for example, can consume 10,000 metric tons fresh *U. prolifera* per day.

Given large geographic distribution and consistent point source of the trans-regional green tide in YS of China, containment measures are widely implemented throughout the coastline of the affected regions. Beside those measures (e.g., clearing algae mass from beaches and coastal waters manually, blocking drifting mats by arresting nets, etc.) to clean and protect the coastal facilities, recreation, and aquaculture along the downstream cities (Fig. 3), strategies to reduce the source biomass are also proposed and have been recently performed in Subei Shoal. These strategies range from pretreatments of the aquaculture facilities to reduce the epiphytic green macroalgae (e.g., antifouling coating, freezing, modified clay, etc.) to series of managements on aquaculture activities to reduce the initial floating biomass (e.g., removing the aquaculture facilities earlier without in situ cleaning, harvesting the initial floating biomass from the major transporting waterways etc., Wang et al. 2020). Apparently, more hygienic and sustainable marine aquaculture should be developed to abate the green tides and further protect the ecosystem in YS.



Green Tides, Fig. 3 Tons of fresh *U. prolifera* cleaned from the coastal water of Jiangsu in May 2016 (with permission of Qingdao Seawin Biotech Group)

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GT MBA: Global Tropical Moored Buoy Array

► Tropical Atmosphere Ocean (TAO) Buoy

Harmful Algal Blooms

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Synonyms

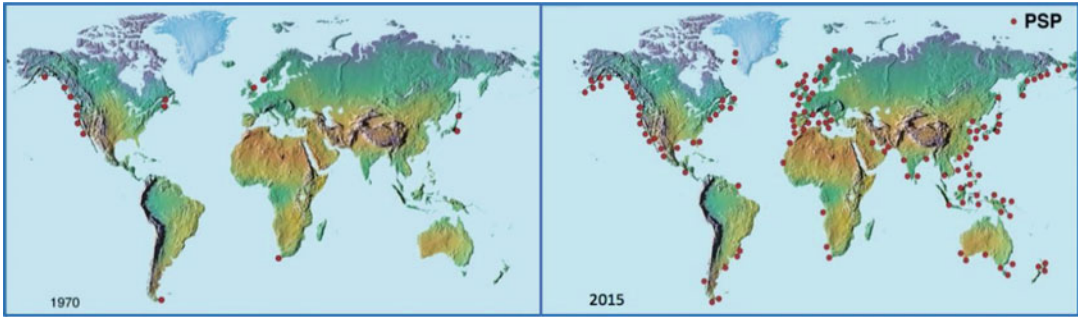
Phytoplankton bloom; Red tide

Definition

Harmful algal blooms (HABs) are a major ecological disaster in the ocean or freshwaters. From an ecological point of view, discoloration is mainly caused by the rapid growth or concentration of harmful or toxic microalgae, macroalgae, and cyanobacteria in water bodies under suitable environmental conditions. HAB species comprise two groups: those that are toxin producers, which can contaminate seafood or kill fish, and those that are high biomass producers, which, at high cellular concentrations, can have negative impacts on aquatic ecosystems, such as hypoxia and indiscriminate killing of marine organisms.

Introduction

HABs are natural ecological phenomenon, caused by rapid algal blooms or accumulation, which can have catastrophic economic or ecological consequences, such as mass mortality of marine animals, contamination of seafood, alteration of marine food webs, or deterioration of tourism (Yu et al. 2018). As shown in Fig. 1, the increased frequency and intensity of HAB worldwide in recent decades has been associated with anthropogenic eutrophication and climate change (Fu et al. 2012; Wells et al. 2015). Algal blooms can last from a few days to several months. Some microalgae are simple photosynthetic organisms that live in oceans and freshwater and can produce toxins that affect humans, fish, shellfish, marine mammals, and birds (Wells et al. 2015). There are different species of HABs, caused by different groups of algae, with different toxins (Granéli and Turner 2006). At the same time as algal blooms, especially during the termination phase, microorganisms may consume oxygen in the water column as they break down the extracellular organic matter of dead algae, resulting in the death of the fish. When some hypoxic zones cover large areas for long periods of time, they are called dead zones, and neither fish nor plants can survive in them. HABs are the end result of physical, chemical, and biological interactions, with excess nutri-



Harmful Algal Blooms, Fig. 1 Global distribution of paralytic shellfish poison (PSP) recorded in 1970 and 2015. (Source: <https://www.whoi.edu/redtide/regions/world-distribution>)

ents in the water considered one of the most important factors. Common nutrients are mainly nitrogen (nitrate, nitrite, ammonia, urea) and phosphate (Anderson et al. 2002).

Causative Organisms of HABs

Microalgae

Diatoms and Dinoflagellates

Diatoms and dinoflagellates are two distinct groups of phytoplankton that are found primarily in the marine environment and are the primary source of HAB species in marine and brackish water. They can form algal blooms and cause discoloration of the water column.

Some diatoms, such as *Pseudo-nitzschia* spp., can produce neurotoxins and domoic acid, which can affect the human nervous system and cause serious injury or death (Kim et al. 2015). Domoic acid (DA) can readily accumulate in the bodies of shellfish, sardines, and anchovies, which is then eaten by people, sea lions, otters, cetaceans, or birds. Due to high levels of DA detected in shellfish during the summer of 2015, the US government closed important shellfish fisheries on the US West Coast, including Washington, California, and Oregon (Ekstrom et al. 2020).

Many dinoflagellates produce a chemical that acts as a neurotoxin in other animals. For example, when dinoflagellates are ingested by shellfish, the chemical can accumulate to high enough levels in shellfish tissues to cause severe

neurological effects in humans, birds, or animals that ingest the shellfish. Some chemicals may affect the action of nerves and interfere with the movement of ions across cell membranes, thereby affecting muscle activity. The toxin saxitoxin, produced by *Alexandrium* and *Gonyaulax*, accumulates in shellfish. Eating such contaminated shellfish can lead to paralytic shellfish poisoning (PSP). Symptoms are typical of PSP, ranging from numbness of the lips and fingers to complete respiratory failure in mild cases to death in severe cases (Mee et al. 1986). Another toxin that accumulates in shellfish is brevetoxin, produced by the dinoflagellate *Karenia brevis* (Wang et al. 2019). Brevetoxin is unique in that when dinoflagellates end up in the surf zone, it turns into an aerosol that then blows onto the beach and causes irritation to the human respiratory tract. Diarrheal shellfish toxins (DTXs) are a class of polyether compounds produced by *Dinophysis* and *Prorocentrum*, among others, with at least eight congeners, the parent compound being okadaic acid (Hackett et al. 2009). Ciguatera is another form of dinoflagellate poisoning in the tropics and is caused by eating fish contaminated with the dinoflagellate *Gambierdiscus* toxin (Skinner et al. 2011; Smith et al. 2016).

Prymnesiophytes

Prymnesium parvum, commonly referred to as “golden algae,” is a single-celled microscopic (about 10 µm), flagellated algae with ichthyotoxic capabilities, and large blooms of *Prymnesium* spp. are often associated with fish kills

(Skovgaard and Hansen 2003). *P. parvum* is toxic not only to fish but also to organisms with gills, such as mussels and clams, and to the gill-breathing juvenile stage of frogs and other amphibians. This algae is found worldwide, with Europe reporting the highest number of large blooms of *P. parvum* (Edvardsen and Paasche 1998; Moestrup 1994).

Cyanobacteria

Cyanobacteria, also known as blue-green algae, are naturally occurring microscopic organisms found in all types of water. These single-celled organisms live in fresh, brackish (combined salt and freshwaters), and salt water. Toxicity by cyanobacteria has been reported since the late nineteenth century, primarily in freshwater environments (Carmichael et al. 1990). Some cyanobacteria can produce hazardous toxins, such as microcystins and hepatotoxins that target mammalian liver. HABs in freshwater lakes and rivers are a serious problem for drinking water and human health. Freshwater and cyanoHABs have major economic, social, and environmental impacts (Berdalet et al. 2017).

Macroalgae

Seaweed (*Ulva prolifera*) is a type of algae found all over the world that creates green tides (Li et al. 2020). The proliferation of *U. prolifera* has negative impact on local communities, aquaculture, and tourism, causing significant damage to the local ecosystem service and economic loss (Li et al. 2020; Zhang et al. 2014). The rapid growth of *U. prolifera*, on the other hand, makes it the strongest competitor for nutrients and light in the bloom area, thereby driving significant impacts on marine biodiversity and community structure (Li et al. 2020).

Sargassum is a genus of brown macroalgae (seaweed). The floating *Sargassum* bloom is called as the golden tides (Xing et al. 2017). The large recurring *Sargassum* beaching has caused serious environmental and economic problems (Wang and Hu 2017). As of June 2018, the Greater Atlantic Sargassum Belt, which extends 8,850 km, consists of more than 20 million tons of biomass. In another case, economic losses were

estimated at 50 million yuan (\$73 million) due to damage to seaweed farming in the Yellow Sea in China's Jiangsu Province in December 2016.

Causes

Among the causes of algal blooms, the following factors are generally considered to be responsible: pollution, eutrophication, anthropogenic impacts, and climate change.

With the progress and development of industry, especially with increasing urbanization and tourism activities, as well as growing coastal population, marine pollution is a growing problem on a global scale (APEC-OFWG 2019). There are three main types of pollution inputs to the ocean: direct discharge of waste into the oceans, runoff from rainwater into waters, and atmospheric releases of pollutants. Eighty percent of marine pollution comes from land-based sources. Air pollution is also a contributing factor, as it carries iron, carbonic acid, nitrogen, silicon, sulfur, pesticides, or dust particles into the oceans.

Eutrophication occurs when a body of water becomes overly enriched with minerals and nutrients, thus inducing excessive algae growth (Chislock et al. 2013). Nutrients enter the freshwater or marine environment in the form of surface runoff from agricultural pollution and urban runoff and from sewage treatment plants that lack nutrient control systems (Anderson et al. 2002; Heisler et al. 2008). Human activities have accelerated the rate and extent of eutrophication through point-source discharges and nonpoint loadings of limiting nutrients, such as nitrogen, phosphorus, and iron, into aquatic ecosystems.

Climate change stresses will affect marine plankton systems globally, and it is conceivable that the frequency and severity of HABs may increase. These climate change pressures will manifest as alterations in temperature, stratification, light, ocean acidification, precipitation-induced nutrient inputs, and grazing (Wells et al. 2015). Global warming, with climate change driving warmer waters, is also considered to be a key factor for algal blooms and making conditions

more favorable for algae growth in more regions and farther north (Lee et al. 2019).

Harmful Effects

Human Health

The main focus is marine, brackish, and freshwater red tide species that produce toxins that affect public health (Berdalet et al. 2017; Geohab 2001). The most virulent species are recorded in dinoflagellates, such as *Pyrodinium bahamense* var. *compressum* and several species of *Alexandrium*, which have significant effects at almost undetectable concentrations (10^2 – 10^3 cells l^{-1}). Fish and shellfish containing toxic phytoplankton are harmful to humans, and other fish, gastropods, crabs, etc. that accumulate algal toxins through the food chain are also toxic (Lewis et al. 1995; Shumway et al. 1995). Toxic syndromes caused by HABs include ciguatera fish poisoning, as well as paralytic, diarrhetic, neurotoxic, azaspiracid, and amnesic shellfish poisoning, primarily associated with the consumption of shellfish. Another concern is the respiratory problems associated with the migration of toxins from aerosols ashore under certain wind and wave conditions (Baden 1983). Although blue-green or other algae do not usually pose an immediate health threat, the toxins they produce are considered dangerous to humans, animals, birds, and fish. Symptoms of drinking toxic water can appear within hours of exposure. They may include nausea, vomiting, and diarrhea or cause headaches and gastrointestinal problems (Grattan et al. 2016). High concentrations of algal toxins can cause skin rashes and irritation to the eyes, nose, mouth, or throat (Grattan et al. 2016). If symptoms persist after 24 h, patients with suspected symptoms are advised to call a doctor.

Natural and Cultured Life Resources

Toxic algae indirectly affect the use of marine resources, while other HAB events directly affect wild and farmed fish and invertebrates, which are valuable seafood products. This is caused by algal toxins but also by mechanical blockages, gill damage, and hypoxia. HABs led

to massive fish kills (Grattan et al. 2016), in addition to the 23 million salmon farmed in Chile that died in 2016 due to toxic algae outbreaks (Gerhart 2017). To remove the dead fish, fish fit for consumption was made into fishmeal, and the rest was dumped 60 miles offshore to avoid any risk to human health (Gerhart 2017).

Recreational Activities and Tourism

Algal blooms can have significant negative impacts on tourism and recreational use of coastal areas. Tourism and recreation rely on relatively high water quality with no visible scum and noxious odors. On the Mediterranean coast, water discoloration and mucus caused by various nontoxic species occur frequently and often coincide with the tourist season, while in the Baltic Sea, during the warmest months, cyanobacterial blooms often hinder the recreational use of the sea (Geohab 2001).

Water Ecosystem Impacts

In addition to their effects on human health and seafood, toxins can move through ecosystems in a manner similar to carbon or nutrient flows, so their effects can be far-reaching. Negative impacts on viability, fecundity, recruitment, and growth are only just beginning to be recognized. Marine mammals have suffered morbidity and mortality from exposure to HAB toxins transmitted through the food web (Scholin et al. 2000). Mass mortality of seabirds, including brown pelicans and cormorants, has also been linked to the spread of diatoms, whose toxins are transferred through the consumption of contaminated pelagic fish (Work et al. 1993).

Clearly, the loss of noncommercial marine resources could also have serious consequences. High biomass, nontoxic HABs can have significant adverse effects on the entire ecosystems, such as reduced light intensity and damage to seagrass or coral reef communities following hypoxia. Severe attenuation of light by HABs results in reduced photosynthetic capacity and leaf biomass in eelgrass beds, leading to substantial loss of fish and shellfish nursery habitat. Oxygen depletion from the degradation of nontoxic blooms kills not only commercially important species but also

other plants and animals (Geohab 2001). When the algae eventually die, the microbes that break down the dead algae consume more oxygen, resulting in more fish dying or leaving the area. When oxygen is continually and heavily consumed, it can lead to hypoxic dead zones where neither fish nor plants can survive. Marine mammals have also been seriously harmed, with more than 50% of unnatural marine mammal deaths caused by HABs (Wilson et al. 2016). Many bottlenose dolphins died from Florida's red tide (Brown et al. 2018). Birds die after eating dead fish contaminated with toxic algae. Decomposing fish are poisoned when eaten by birds such as pelicans, gulls, cormorants, and possibly marine or land mammals (Grattan et al. 2016). The nervous systems of the dead birds have been examined, and due to the toxins, their nervous systems have failed (Starr et al. 2017). It has even been suggested that HABs are responsible for the death of the animals found in the fossil reservoir (Gramling 2017).

Other Economic Consequences

According to a 2016 report, the economic impact of HABs comes primarily from damage to human health, fisheries, tourism and recreation, and the cost of monitoring and managing areas where HABs occur (Ritzman et al. 2018). In addition to the damage to businesses, losses from human diseases include lost wages and impaired health. The imposition of fishing bans in areas where algal blooms are occurring has had a significant negative impact on fisheries, coupled with the consequent high fish mortality rates, increased prices due to shortages of available fish, and reduced demand for seafood due to concerns about toxin contamination (Avdelas et al. 2021). This has resulted in significant economic losses to the industry.

Potential Remedies

Chemical Treatment

Although some algacides are effective at killing algae, they are mostly used in small bodies of water. For large algal blooms, however, the

addition of algacides such as silver nitrate or copper sulfate may have worse effects, such as killing fish directly (Tsai 2016). So the negative effects may be worse than allowing the algae to die out naturally (Tsai 2016).

Microcystis is prevalent, using a new algacide formulation (by BlueGreen Water Technologies) that floats the granular product and slowly releases its active ingredient, sodium percarbonate, which releases hydrogen peroxide (H_2O_2) at the water's surface (Sinha et al. 2020). Therefore, effective concentrations are limited vertically to the water surface and spatially limited to cyanobacteria-rich areas. This provides a "safe haven" for aquatic organisms in untreated areas and avoids the adverse effects associated with the use of standard algacides.

Red Clay and Modified Clay

The flocculation of natural minerals is used to kill red tide organisms. The natural minerals are mainly clay minerals, supplemented by other minerals. They are characterized by abundant sources, low cost, low pollution, and high adsorption capacity. In Korea, natural red clay is often used to control serious red tide of *Cochlodinium polykrikoides* in order to reduce the risk to farmed economic fish. This has been proven to be the most effective and environmentally friendly mitigation strategy compared to other methods. In China, modified clays have been used several times to successfully treat HABs in oceans and lakes. For example, cyanobacteria bloom in Xuanwu Lake of Nanjing during the National Games and the site for sailing competition of Qingdao in 2008 Olympic Games. Compared to natural clays, modified clays are much more efficient at resisting red tide, but the cost will increase accordingly.

Monitoring and Predicting of HABs

A growing number of scientists agree on the urgent need to protect the public by predicting HABs. One way they hope to do that is through the use of sophisticated sensors to help warn of potential algal blooms (Binding et al. 2020). Water treatment facilities can also use the same type of sensors to help them prepare for higher toxicity (Brooks et al. 2017). In 2016, the

automatic early warning monitoring system was successfully tested and was shown for the first time to identify rapid algal growth and consequent oxygen depletion in the water column (Brooks et al. 2017).

It is widely believed that HABs have been increasing in frequency and geographic distribution worldwide. Over the past decade, HABs have appeared in all coastal areas around the world, and in some places new species have emerged that were not previously known to cause problems. The frequency of HABs is also thought to be increasing in freshwater systems (Schmale et al. 2019). The report recommends, among other remedial measures, the use of improved monitoring methods in an effort to increase predictability and the testing of new potential methods for controlling HABs (Schmale et al. 2019).

Reducing Chemical Runoff

Nitrates and phosphorus in fertilizers can lead to algal blooms when they flow into lakes and rivers after heavy rains. Changes in farming practices, such as targeted use of fertilizers only at the right time, have been suggested to reduce potential runoff (Hayder et al. 2019). Drip irrigation is a successfully used method that does not spread the fertilizer widely over the field but rather drips the plant roots through a network of pipes and emitters, leaving no trace of the fertilizer being washed away (Çetin and Akalp 2019).

Cross-References

► Green Tides

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Harmful Ecological Effects

► Eutrophication

HAWT – Horizontal-Axis Wind Turbine

► Analysis of Renewable Energy Devices

Heading Sensor (HS)

► Ultra-short Baseline Underwater Acoustic Location Technology

Health, Safety, and Environmental Management System (HSE)

► Polar Offshore Engineering

Heat Preservation

► Thermal Insulation

Heave Plate

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Definition

The *heave plate* is a damping device, usually attached to the submerged end of Spar-like floating platforms. Its primary function is to suppress the heave motion responses of the floating

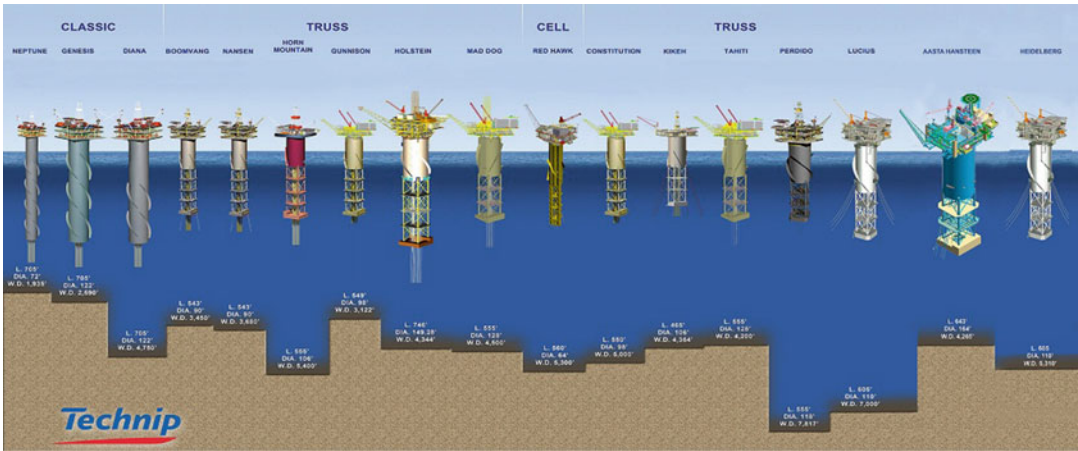
structures and also shift the natural heave period further away from the typical incident wave by increasing the *added mass* and *viscous damping* of the floating platform.

Scientific Fundamentals

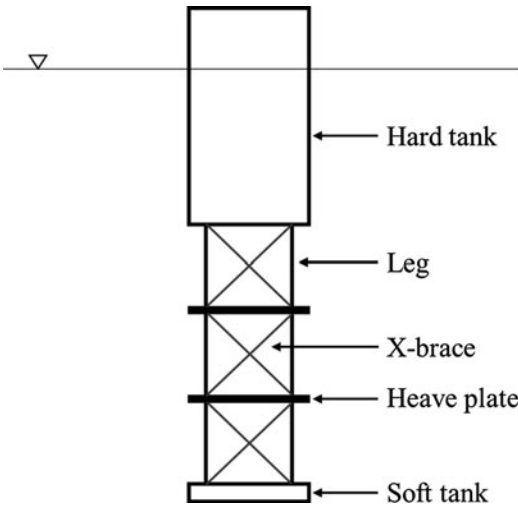
Introduction

After over 30 years of development, the Spar platform has become the most cost-effective floating platform for offshore oil and gas production in deep and/or ultra-deep water. As a deep draft floating oil platform, the Spar platform consists of a large diameter single vertical cylinder. Due to its deep draft, the Spar platform has excellent heave performance. After the first Spar platform (Neptune Spar) was designed by Kerr McGee in 1996, three primary types of Spar platform have been formed, including the Classic Spar, Truss Spar, and Cell Spar platforms as shown in Fig. 1.

The Classic Spar platform almost relies on the deep draft to obtain an excellent heave performance. However, the cost sharply grows with the increase of the draft. Thus the Truss Spar platform had been widely used to overcome the shortcomings. It has a shorter hard cylindrical tank than the Classic Spar and truss structure connected to the bottom of the hard tank (see Figs. 1 and 2). The truss structure consists of four sizable orthogonal legs with X-braces and a small keel (or soft tank), which houses the heavy ballasting material. Generally, the truss structure can't supply adequately the added mass and viscous damping. Although the natural period of heave response of a Spar platform is about 20s to 30s which is far away from the wave period (Subbulakshmi et al. 2015), heave resonant oscillations may occur due to long period waves. Thus, additional damping devices and active damping systems are introduced to reduce the heave motion. The most typical device is the heave plate. As shown in Figs. 1 and 2, a heave plate is a thin plate structure to reduce the heave response by increasing the added mass and damping of Spar platform. The circular, square, and hexagonal are the three primary shapes of the heave plate.



Heave Plate, Fig. 1 Typical types of the Spar platform. (www.ourworldofenergy.com)



Heave Plate, Fig. 2 Typical structure of the Truss Spar platform

Similar to the Truss Spar platform, the Cell Spar platform also designs the heave plate.

Hydrodynamic Performance

In general, the motion response of a floating platform should be small enough to enable the operation of rigid risers. Further minimizing the heave response and maximizing the natural heave period of Spar platform, respectively, are two key processes to reduce the heave motion. The Keulegan-Carpenter number (KC) and frequency parameters (β) are the two

nondimensional characteristic parameters representing hydrodynamic response and heave period, respectively (Tao and Thiagarajan 2003a).

$$KC = 2\pi x/D \quad (1)$$

$$\beta = D^2 f/\nu \quad (2)$$

Note that D is the diameter of the cylinder (side length when the Spar platform hull is square), x and f are the amplitude and frequency of the oscillation, and ν is the kinematic viscosity of the fluid, respectively. The natural heave period is formulated by $T = 2\pi/f = 2hC_m/g$, where h is the Spar draft and C_m is the added mass coefficient. The coefficient C_m and viscous damping coefficient C_d in the Morison equation are often used to describe the hydrodynamic characteristics.

$$(M + \rho \nabla C_m) \ddot{x} + 1/2 \rho A C_d |\dot{x}| \dot{x} = F \quad (3)$$

Note that M is the mass of the Spar platform, ρ is the density of water, ∇ is the volume displacement, A is the projection area of the structure, $\dot{x}$ and $\ddot{x}$ are the speed and acceleration of the structure, and F is the force on the system, respectively (Sarpkaya et al. 1981). On one hand, the heave plate increases the added mass and mass coefficient C_m . It increases the heave period T and decreases the value of frequency parameter. On the other hand, increasing the viscous

damping C_d also can reduce the heave response x of the Spar platform based on Eq. (3). In short, the heave plate increases the added mass and viscous damping to gain an excellent heave performance.

The Influencing Factors of Heave Plate to Spar Hydrodynamic Performance

The principal factors of heave plate affecting the hydrodynamic performance of the Spar platform are listed in Table 1.

The Size of Heave Plates

Downie et al. (2000) obtained the damping ratio ξ and natural period T of solid and perforated heave plates which have the side length of 0.395 and 0.295 m as showed in Table 2. It is indicated that the larger heave plate makes the bigger damping

ratio and the longer natural period. Thus the heave plate should be as large as possible to achieve a superior movement stability.

The Number of Heave Plates

Due to the shielding effects between the heave plates, the increase rate of the added mass is usually lower than the growth rate of the number of the heave plate. Thus, the average added mass of each heave plate will decrease based on these effects. Using the numerical and experimental data, many researchers studied the heave motion of the Spar platform with single and two heave plates (Downie et al. 2000; Tao and Thiagarajan 2003b; Tao and Cai 2004; Tao et al. 2007; Sudhakar and Nallayarasu 2013; Subbulakshmi et al. 2015). As shown in Table 3, two or three heave plates are usually installed for the most Truss and Cell Spar platforms in practice.

The Spacing of Heave Plates

From structural design, the angle of the X-braces is usually about 45° . The leg spacing of legs L1 is equal to the spacing of heave plates H ($L1 = H$) and the standard value of L1 is $0.707D$ ($L1 = 0.707D$). The width of heave plate L is usually extended to the diameter of hard tank D ($L = D$). The typical spacing of heave plates H is $0.70L-0.75L$, and the added mass of each heave plate is 85–95% of the single heave plate. The efficiency of the heave plate is about 90%. It is concluded that reducing the spacing of heave plates will reduce its effectiveness. But for resistance, the effect of spacing is not significant (Prisilin et al. 1998). Ji et al. (2003) conducted the experiment with the triangular heave plate using the forced oscillation method. The vicious damping coefficient C_d obtains the maximum when $H = 1.5L$, and the added mass coefficient C_m obtains the maximum when $H = 1.0L$. Shen et al. (2011) simulated the square heave plate, and the result suggested that the added mass coefficient C_m obtains the maximum (about 90% of the added mass coefficient of the single heave plate) when $H > 1.5L$. On the other hand, the damping coefficient C_d would not increase anymore with the spacing increasing when $H > 1.5L$ (Fig. 3).

Heave Plate, Table 1 The influencing factors of heave plate to Spar hydrodynamic performance

The influencing factors	Researchers (year)
The size of heave plate	Downie et al. (2000)
The number of heave plates	Downie et al. (2000), Tao and Thiagarajan (2003b), Tao and Cai (2004), Tao et al. (2007), Sudhakar and Nallayarasu (2013), Subbulakshmi et al. (2015)
The spacing of heave plates	Prisilin et al. (1998), Ji et al. (2003), Shen et al. (2011)
The shape of heave plate	Wu et al. (2012)
The thickness of heave plate	He (2003), Liu et al. (2012)
The porosity of heave plate	Wu et al. (2012), Zhang et al. (2006)
The KC number	Liu et al. (2012)
The load of heave plate	Gao and Liu (2007)

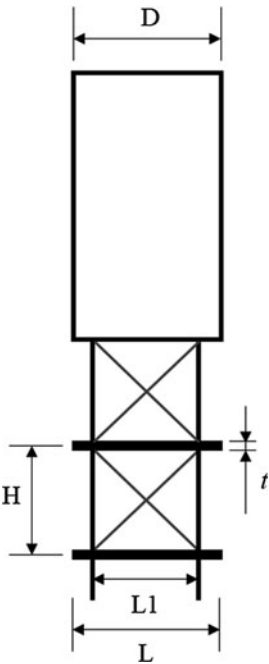
Heave Plate, Table 2 Damping ratio and natural period of heave plates

Configuration	$\xi/\%$	T/s
Small perforated	4.43	1.83
Large perforated	7.61	2.13
Small solid	6.71	2.56
Large solid	8.57	2.84

Heave Plate, Table 3 Truss and Cell Spar platform with the heave plates

Spar platform	Length/ft	Outside diameter/ft	Water depth/ft	Number	Type
Boomvang	543	90	3680	3	Truss
Nansen	543	90	3450	3	Truss
Horn Mountain	666	106	5400	3	Truss
Gunnison	549	98	3122	3	Truss
Holstein	746	149.2	4344	3	Truss
Mad Dog	555	128	4500	2	Truss
Red Hawk	560	64	6300	2	Cell
Constitution	550	98	5000	3	Truss
Kikeh	465	106	4364	2	Truss
Tahiti	555	128	4200	2	Truss
Perdido	555	118	7817	2	Truss
Lucius	606	110	7000	2	Truss
Aasta Hansteen	643	164	4256	2	Truss
Heidelberg	605	110	5310	2	Truss

H



Heave Plate, Fig. 3 Heave plate on the Truss Spar platform (Ji et al. 2003)

The Shape of Heave Plates

The circular, square, and hexagonal are the three primary shapes of the heave plate. With the same area and thickness of abovementioned three plates, the circular heave plate has the maximum damping coefficient C_d , and the hexagonal

has the maximum added mass coefficient C_m (Wu et al. 2012).

The Thickness of Heave Plates

The main damping of heave plate is caused by the vortex shedding at the thin edge. When the thickness t is small, the strong interaction between the vortices appearing at the upper and lower edges appeared. He (2003) found that the damping effects are significantly reduced when $t > 1/50L$. Experimental study also demonstrates that the added mass decreases as the aspect ratio t/L increases, and with aspect ratio t/L (exceeds $1/50$) increasing, the viscous damping coefficient greatly decreased (Liu et al. 2012).

The Porosity of Heave Plates

Opening in the heave plate increases the perimeter in contact with the water and creates more vortex shedding to enhance the damping effects. Numerical and experimental studies show that the damping of heave plate increases first and then decreases with the increase of the rate of opening (Wu et al. 2012; Zhang et al. 2006). However, the added mass decreased, and the influence of the rate of opening is remarkable at a small KC (Wu et al. 2012). The maximal damping coefficient C_d is acquired at 10% porosity. Further the opening should be as small as possible to obtain a larger opening perimeter (Zhang et al. 2006).

Besides, the added mass and viscous damping coefficient do not significantly increased with the diameter of opening decreasing and the number of the openings increasing at the same porosity (Wu et al. 2012).

The KC Number

KC reflects the impact of the amplitude to the hydrodynamic characteristics of heave plates. Liu et al. (2012) studied the Spar model with three heave plates in the experiment. The added mass coefficient C_m increased as KC changed from 0.2 to 1.3. The viscous damping coefficient C_d decreased significantly when KC is changed from 0.2 to 0.4. But it decreased slowly with the varying of KC from 0.4 to 1.3.

The Load of Heave Plate

The shear deformation of the truss caused deformation and stress of the heave plate. In key regions, the stress caused by shear deformation could reach about 50% of the total stress (Gao and Liu 2007). The solution is to add displacement to the leg on the local finite element model of the heave plate and then superimpose the analysis results on the basic of the previous analysis. Besides, the Spar platform floats on its buoyancy and is towed by a tug in the process of wet tow. About one-third of heave plates are submerged in water, and the load distribution is different from that when it is in place.

Conclusions

As a vital component of the Spar, the heave plate increases the added mass and extra viscous damping of the marine structure to reduce the heave motion. It's an economical and practical strategy to improve the stability of the structures by installing a heave plate. Further research on its hydrodynamic performance and structural optimum design is still significant to ensure the safe, reliable, and high-efficiency operation of the Spar platform.

Cross-References

► Spar Platform

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High Chromium Duplex Corrosion-Resisting Stainless Steel (HDR)

► [Piping Technology](#)

High Latency

► [Underwater Acoustic Sensor Network](#)

High Nutrients

► [Eutrophication](#)

High-Performance Boats

► [High-Performance Ship](#)

High-Performance Ship

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Synonyms

[High-performance boats](#); [High-performance vessels](#); [High-speed craft](#)

Definition

High-performance ship is a general name of a ship with a significant advantage over a conventional draining-type ship to achieve high speed, good navigability, and economical efficiency.

Scientific Fundamentals

Development History

High-performance ships are completely or partially out of the water during high-speed navigation, to reduce water resistance, especially wave resistance, and greatly reduce the swaying of the hull caused by waves, which can effectively improve the ship's speed and wave resistance. When the ship is at high speed, the buoyancy of the ship will not be used as the main factor to support the weight of the ship (Zhao and Xie 2009).

A series of new variations of high-performance ships have been developed in the last half century, including improved planing craft from the 1940s, hydrofoils from the 1950s, air-cushion vehicles and surface effect ships from the 1960s, small water plane area twin hull craft from the 1970s, high-speed catamarans from the 1980s, wave piercing craft from the 1990s, high-speed trimarans in the first decade of the twenty-first century, and wing in ground effect craft from 1970s to the present. These various concepts and the hybrids that we will describe form an interacting group of vehicle concepts (Liang and Bliault 2012). The following are the main types of high-performance ships, which are briefly introduced.

Key Technology

High-performance ships are a new type of ship developed from the 1930s of last century. The starting point of the exploration, research, and development of high-performance ships is how to make use of the supporting action of the water body by using the different supporting methods and the fluid mechanics principle with the conventional ship and at the same time get rid of the friction resistance of the surface water of the hull and the free surface wave resistance.

The high-speed navigation of the ship is realized. After decades of research and development, the research, design, construction, and operation technology of high-performance ships have become increasingly mature. The industry has developed various types of high-performance ships, which meet the special requirements of shipping and military to all kinds of shipping and operation of ships. As a young branch of ancient water transportation, high-performance ships are playing a more and more significant role in the field of civil and national defense (Liang and Bliault 2012).

Key Applications

Planing Craft

The concept of planing ship was recognized in the late nineteenth century; the first practical application of the concept can be traced to the development of seaplane hulls during the beginning of the twentieth century. Planing ships have important commercial applications, such as in offshore supply operations for the oil industry; high-speed support is becoming increasingly important as oil production operations move into deeper water farther offshore. Militarily, the role of special forces is increasing, and speed is necessary for effective littoral zone operations. The military high-speed sealift effort has recently renewed its interest in the development of smaller, high-speed

ships, of which planing or semi-planing hull forms are a principal alternative.

There are two basic types of ship hulls, displacement and planing, respectively. Ships with displacement hulls move through the water by pushing the water aside and are designed to cut through the water with little propulsion. Ships with planing hulls are designed to rise up and glide on top of the water when enough power is supplied. These ships may operate like displacement hulls when at rest or at slow speeds but climb toward the surface of the water as they move faster.

A planing craft is a high-powered watercraft that develops the lift force that supports its weight primarily through hydrodynamic water pressure. In order to develop sufficient dynamic supporting lift, the speed of the planing craft must be high relative to its size, and the surfaces of the hull must be properly configured. The proper hull surface configuration is generally a shallow V-section with some variation both longitudinally and transversely. The planing ship is typically run with a bow-up trim angle of a few degrees (Vorus 2017). This trim results in development of high pressure on the hull bottom, which pushes the ship upward toward the water surface, thereby reducing its immersed surface area and, consequently, the ship resistance relative to the case of full displacement (Fig. 1) (Sakaki et al. 2017).

The planning ship has high speed, small tonnage, and flexible manipulation, but the skiing

High-Performance Ship,

Fig. 1 Planing craft (http://www.boated.com/washington/studyGuide/Planing-Hulls/10105001_700060590)



waves have great impact on the hull, and the stall in the wind and waves is great, so the skiing boat is not suitable for sailing in the large wind and waves. The sewerage of the planing ship is generally within 200 t, and the speed can reach 40–50 knots.

Hovercraft

A hovercraft, also known as an air-cushion vehicle or ACV, is a craft capable of traveling over land, water, mud, or ice and other surfaces both at speed and when stationary. Hovercrafts are hybrid vessels operated by a pilot as an aircraft rather than a captain as a marine vessel. A hovercraft is a high-speed ship that uses the principle of surface effect to form air cushion between the hull and the support surface (surface or ground) to make the whole or part of the ship sail out of the support surface, depending on the principle of surface effect. The speed of a hovercraft is usually 30–80 knots, and that of a military hovercraft can reach 90–100 knots.

Hovercraft use blowers to produce a large volume of air below the hull that is slightly above atmospheric pressure. The pressure difference between the higher-pressure air below the hull and lower-pressure ambient air above it produces lift, which causes the hull to float above the running surface. For stability reasons, the air is typically blown through slots or holes around the

outside of a disk- or oval-shaped platform, giving most hovercraft a characteristic rounded rectangle shape. Typically, this cushion is contained within a flexible “skirt,” which allows the vehicle to travel over small obstructions without damage. Vehicles designed travel close to but above ground or water. These vehicles are supported in various ways. Some of them have a specially designed wing that will lift them just off the surface over which they travel when they have reached a sufficient horizontal speed (the ground effect). Hovercrafts are usually supported by fans that force air down under the vehicle to create lift. Air propellers, water propellers, or water jets usually provide forward propulsion. Air-cushion vehicles can attain higher speeds than can either ships or most land vehicles and use much less power than helicopters of the same weight (Amit 2015) (Fig. 2).

Hovercraft has the following characteristics:

1. Low resistance and high speed. The effect of air cushion is free from the obstacle of the increase of water resistance exponentially with the speed of the conventional sewerage, which provides a basic condition for increasing the ship's speed.
2. Amphibious. The full hovercraft has a unique amphibious nature, which can pass through shallow water, swamps, ice and snow, beach

High-Performance Ship,
Fig. 2 Air-cushion vehicle
(<http://military.wikia.com/wiki/hovercraft>)



and land, and so on. It has good navigability, and it can be landed and cross barrier without terminal facilities.

3. Large deck area and storage capacity can be set up. Because most of the hull of the hovercraft is off the water, the volume is increased, and there is no excessive resistance. Therefore, a large deck area and storage capacity can be set up as required, without causing too much resistance.
4. Good overloading ability. Unlike hydrofoil and gliding ships, the hovercraft is a cushion lift by the lift system separated from the propulsion system. Overloading only causes a slow decline in the speed of the aircraft, unlike the skidding ship or the hydrofoil ship that is unable to slide or lead to the speed of the flight.

Hovercraft has the advantage of being applied all-weather. It can be used in shallow water rapids, swampy areas, shallow deepwater beaches, and icicles. In particular, it is driven through harsh waterways and roads, which shows the unparalleled advantages of ordinary ships. The resistance of the hovercraft is very small during the driving process, and the speed is fast. The disadvantages include poor stability, difficulty in handling, high stall speed and poor wind resistance, and high noise.

Hydrofoil Craft

As early as 1905, the Italian inventor invented the first hydrofoil and carried out trial tests with a speed of 36 kn. Hydrofoil boats have received much attention since then. Then in 1918, American Bell improved the hydrofoil craft and built a hydrofoil craft that could theoretically reach a speed of 60 kn and successfully sailed. In 1935, Germany developed and cut a type V hydrofoil with a speed of 30 kn. The Soviet Union has developed rapidly in the construction of hydrofoil craft. It has built 12 types of hydrofoil boats and put them into use since 1957. Most of them are self-stabilizing and self-settling controlled hydrofoils, operating at speeds up to 37 kn. The United States is also one of the major hydrofoil developing countries. Its research and

development of fully flooded controlled hydrofoil passenger boats use water jet propulsion devices. In 2005, the China Ship Science Research Center designed and built China's first mixed hydrofoil catamaran, which greatly reduced the resistance and roll response to navigation in waves.

Hydrofoil craft is a ship with submerged hydrofoils under the hull. At high speed, the weight of the ship is entirely supported by the lift force generated by the water to the hydrofoil, so that the hull is all away from the water surface, reducing the resistance of the water, so it can reach a higher speed. The principle of the action of a hydrofoil is similar to the wing of an aircraft. Some speed ships only have hydrofoil in the capital, leaving the front part off the water and the rear still sliding on the water.

Hydrofoil is the wing that works in water or cuts the surface of the water. It is the basic component of hydrofoil. It moves in water, like airplane wings moving in the air, which gives rise to certain lift. Because the density of water is about 800 times that of air density, the diameter of hydrofoil is much smaller than that of the wing under the same lift. The hydrofoil based on the lift force of the hydrofoil has smaller resistance than the planing ship, little wave propagation, and little influence on wave interference, so it has good rapidity and seaworthiness.

When used as a lifting element on a hydrofoil ship, this upward force lifts the body of the vessel, decreasing drag and increasing speed. The lifting force eventually balances with the weight of the craft, reaching a point where the hydrofoil no longer lifts out of the water but remains in equilibrium (Ling et al. 2011). Since wave resistance and other impeding forces such as various types of drag on the hull are eliminated as the hull lifts clear, turbulence and drag act increasingly on the much smaller surface area of the hydrofoil, and decreasingly on the hull, creating a marked increase in speed (Fig. 3).

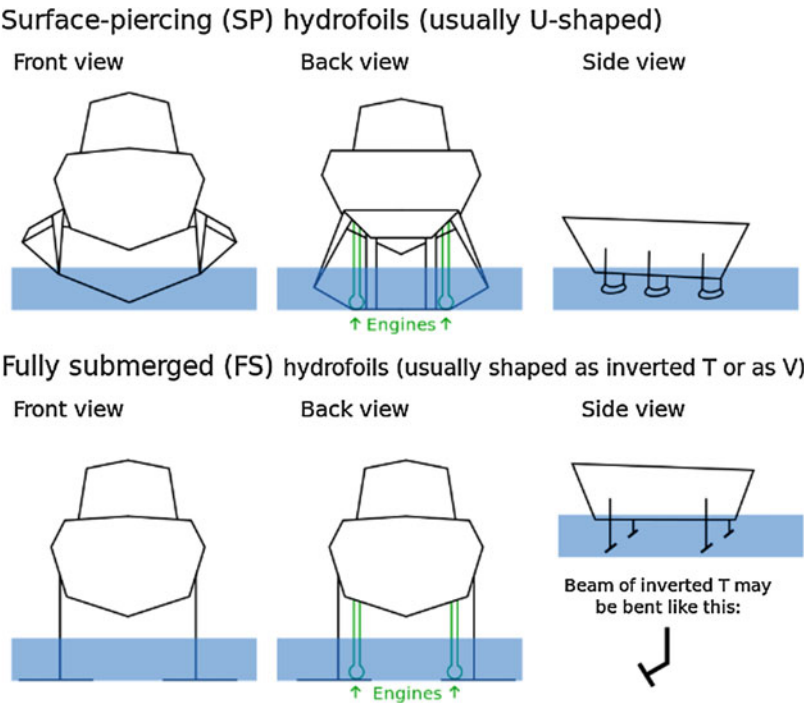
Early hydrofoils used V-shaped foils. Hydrofoils of this type are known as "surface-piercing" since portions of the V-shape hydrofoils rise above the water surface when foilborne. Some modern hydrofoils use fully submerged inverted T-shape foils. Fully submerged hydrofoils are less

High-Performance Ship,
Fig. 3 Hydrofoil craft
(<http://military.wikia.com/wiki/hovercraft>)



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High-Performance Ship,
Fig. 4 Two types of
hydrofoils: surface-piercing
and fully submerged
(<https://ipfs.io/ipfs/>)



[QmXoypizjW3WknFiJnKLwHCnL72vedxjQkDDP1mXWo6uico/wiki/Hydrofoil.html](https://ipfs.io/ipfs/QmXoypizjW3WknFiJnKLwHCnL72vedxjQkDDP1mXWo6uico/wiki/Hydrofoil.html)

subject to the effects of wave action and, therefore, more stable at sea and more comfortable for crew and passengers. This type of configuration, however, is not self-stabilizing. The angle of attack on the hydrofoils must be adjusted continuously to changing conditions, a control process

performed by sensors, a computer, and active surfaces (Fig. 4). The main advantages of hydrofoil craft are low resistance at high speeds. When the hydrofoil is in the wing state, the hull is lifted by the hydrofoil, and only the hydrofoil, the rudder, and the shaft

system are used for the underwater components. Therefore, the water resistance has been greatly reduced, and the speed has been greatly improved. The existing hydrofoil can reach a maximum speed of 70 knots; the wave resistance is good. As the hull is lifted off the water surface in the wing state, the disturbance of the wave to the hydrofoil is reduced, so the swaying motion of the hydrofoil in the wind and waves is reduced, and its wave resistance is greatly improved. The wave of high-speed sailing is small. Since only the hydrofoil, rudder, and other components are in the water during the wing state, the rising waves are very small. Therefore, hydrofoil craft is very suitable for inland shipping, which can greatly reduce the damage of the wave on the embankment; physical field radiation is low. When the hydrofoil is in the wing state, the physical field radiation of the hull will be weakened due to the hull being separated from the water surface. In particular, the sound radiation will be greatly reduced. The low-physics radiation characteristics of the hydrofoil craft help to increase the stealth of the ship and reduce the risk of underwater attack; the maneuverability is good. When the hydrofoil is under wing flight, it uses the combined action of the flap and the rudder to make the hull produce a safe inward rotation and thus has a good gyration.

The main disadvantages of hydrofoil craft are hydrofoil under the hull, not suitable for shallow waterways; hydrofoil width is often greater than the width of the ship; berthing docks is inconvenient; high-speed internal combustion engines or gas turbines are often used to reduce the weight of the main engine; and the service life is short. The demand for fuel is high and the economy is poor. Due to the limitations of hydrofoil lift and structural strength, there are still technical difficulties in manufacturing large-scale hydrofoil vessels or further increasing the speed. The buoyancy provided by the hydrofoil is squared with the length, but the weight of the vessel is related to the length of the vessel (square/cubic law). Therefore, it is difficult to manufacture larger hydrofoil vessels. To further increase the speed, the problem that the hydrofoils will generate bubbles at high speeds also needs to be solved. In addition, the structure

and control of the full-immersion hydrofoil are relatively complicated, which also increases the cost. The use of fuel gas engines for hydrofoil vessels is also one of the considerations for commercial operation.

Wing in Ground Effect Vehicles

Wing in ground effect vehicles (WIG) are an emerging technology that provide a surface transport utility that is between aircraft and existing marine craft, in terms of speed, cost (capital and running), and load carrying capacity (Klaus and Hanno 2003). With a cruising speed of circa more than 90 knots, they are faster than almost all other seagoing craft, and over short/medium distances, they provide journey times comparable with helicopter and aircraft. Yet they can be built, certified, and operated as ships. Skimming above the water but without water contact provides efficiency and long endurance, with no sea motion fatigue and no wake and having natural stealth. Such vehicles are of potential use in both civil and paramilitary fields (Fig. 5).

A WIG vehicle is a dynamic hovercraft. Unlike a conventional hovercraft that generates a static cushion which is dragged around within a skirt, the WIG generates a dynamic cushion between itself and the surface below it by virtue of its own forward passage. The WIG utilizes wing-in-ground effect, a phenomenon that relates to the airflow around a wing when it flies in close proximity to a surface, wherein the presence of the surface distorts the downwash from the wing and inhibits the formation of vortices. This results in two beneficial effects (Moore et al. 2002). There is a pressure increase on the lower surface of the wing due to the conversion of the dynamic pressure of the forward motion to static pressure, which results in a substantial increase in lift. At the same time, there is a substantial reduction in (lift-induced) drag because the vortices and downwash are less, so there is less energy stored in them. The combined effects lead to a higher lift/drag ratio for a wing in ground effect than could be achieved in free flight and hence higher efficiency. Indeed, heavy seabirds use this ground effect to save their energy on long-distance flights. However, birds have a natural automatic

High-Performance Ship,
Fig. 5 Wing in ground
 effect vehicles (http://www.savorlife.com/main_travel.htm)



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height and longitudinal stabilization system. Dr. Lippisch regarded aircraft as “frozen birds” that needed special methods for self-stabilization. He solved this by using a reversed Delta wing, flying in ground effect, in combination with a high T-Tail, which flies out of ground effect. This configuration guarantees a safe stability at all heights from the surface. The ground effect gets stronger as the surface is approached and efficient cruise height is a balance between wave clearance and the optimum power efficiency and generally equates to a cruise height of equivalent to 10% of wing span.

High cruise speed and transportation efficiency are the main advantages. However, poor maneuverability and high accident risk are the main disadvantages.

Multihull Vessels

A multihull is a ship with more than one hull. Multihull ships (vessels, craft, ships) include multiple types, sizes, and applications. Hulls range from two to five, of uniform or diverse types and arrangements. Catamarans are the most common type, with two symmetric hulls. They are used as racing, sailing, tourist, and fishing ships. A trimaran is a multihulled vessel, consisting of one long main hull and two shorter outriggers, also called wing hulls, located on both sides of the main hull. The outriggers provide the trimaran with excellent stability and seakeeping characteristics (Kang et al. 2001). A trimaran can be either

wind driven as shown or mechanically driven. Another unique feature of the trimaran design is the small ratio of width to length. Trimarans are typically designed with very long and slender hull shapes in order to decrease the wave-making resistance of the ship. The main hull and outriggers can also be arranged so that the waves produced by the ship destructively interfere, producing smaller waves and thereby losing less energy to wave-making resistance (Xu and Zou 2001).

Wave-making resistance is classified as the drag force on the trimaran which is created by the loss of energy to the formation of waves. The excellent seakeeping performance and the lower resistance at high speeds make the trimaran design appropriate for high-speed travel applications. Currently, trimarans are being used for business, recreation, and military applications because of their particular advantages over monohull ship designs. Trimarans have a much smaller draft than most ships, which allows them to access areas inaccessible to ships of similar size; draft is defined as the distance which a ship extends below the water surface. Ferries and sail ships have been designed using the trimaran configuration in order to provide the high speeds necessary for commercial transportation and racing. The trimaran design is also favorable for military and research applications due to the large open deck space available for equipment (Figs. 6 and 7).

High-Performance Ship,**Fig. 6** A catamarans

(<https://bvisail.com/crewed-yachts/bvi-power-cat-charters/power-catamaran-charters/>)

**High-Performance Ship,****Fig. 7** A trimaran (<https://wordlesstech.com/adastra-trimaran-superyacht/>)**SWATH**

A small waterplane area twin hull, better known by the acronym SWATH, is a twin-hull ship design that minimizes hull cross section area at the sea's surface. Minimizing the ship's volume near the surface area of the sea, where wave energy is located, maximizes a vessel's stability, even in high seas and at high speeds. The bulk of the displacement necessary to keep the ship afloat is located beneath the waves, where it is less affected by wave action. Wave excitation drops exponentially as depth increases (deeply submerged submarines are normally not affected by wave action at all). Placing the majority of a ship's

displacement under the waves is similar in concept to creating a ship that rides atop twin submarines (Misra 2015) (Fig. 8).

A narrow waterline distinguishes a SWATH ship from a conventional catamaran. The twin-hull design provides a stable platform and large, broad decks. Compared with conventional catamarans, SWATH vessels have more surface drag, but less wave drag; they are less susceptible to wave motion but more sensitive to payload, which affects draft. Additionally, SWATH vessels cannot operate in planing or semi-planing modes and thus gain no drag reduction when operating at speeds normally associated with such modes,

High-Performance Ship,
Fig. 8 A SWATH (<http://worldmaritimeneews.com/archives/45871/danish-yachts-presents-seastrider-swath/>)



require a complex control system, have a deeper draft, and have higher maintenance requirements. The design of SWATH vessels is also considerably more complex due to the structural complexities inherent to the design (Busch 1990).

Cross-References

- [Special Marine Vehicle](#)
- [Surface Warships](#)

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High-Performance Vessels

- [High-Performance Ship](#)

High-Pressure Zone

- [Ship-Iceberg Interactions](#)

High-Speed Craft

- [High-Performance Ship](#)

Hinge Connectors

- [Connectors of VLFS](#)

Hollow Core PCFs (HC-PCFs)

► Fiber Optic Hydrophone

Homotopy Analysis Method

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Introduction

Nonlinear equations are difficult to solve, especially analytically. Perturbation techniques (Dyke 1975; Hinch 1991; Murdock 1991; Nayfeh 1973) are widely applied in science and engineering, and do great contribution to help us understand many nonlinear phenomena. However, it is well known that perturbation techniques are essentially based on the existence of small or large parameters or variables called perturbation quantity. Briefly speaking, perturbation techniques use perturbation quantities to transfer a nonlinear problem into an infinite number of linear subproblems and then approximate it by the sum of solutions of the first several subproblems. The existence of perturbation quantities is obviously a cornerstone of perturbation techniques; however, it is the perturbation quantity that brings perturbation techniques some serious restrictions. Firstly, it is impossible that every nonlinear problem contains such a perturbation quantity. This is an obvious restriction of perturbation techniques. Secondly, analytic approximations of nonlinear problems often break down as nonlinearity becomes strong, and thus perturbation approximations are valid only for nonlinear problems with weak nonlinearity.

To overcome the restrictions of perturbation techniques, some traditional nonperturbation methods are developed, such as Lyapunov's artificial small parameter method (Lyapunov 1992), the δ -expansion method (Awrejcewicz et al. 1998; Karmishin et al. 1990), Adomian's decomposition method (1976, 1991, 1994), and so on. In principle, all of these methods are based

on a so-called artificial parameter, and approximation solutions are expanded into series of such kind of artificial parameter. This artificial parameter is often used in such a way that one can get approximation solutions efficiently for a given nonlinear equation. Compared with perturbation techniques, this is indeed a great progress. However, like perturbation techniques, both the Lyapunov's artificial small parameter method and the δ -expansion method themselves do not provide us with a convenient way to adjust convergence region and rate of approximation series. And Adomian's decomposition method has also some restrictions. Approximate solutions given by Adomian's decomposition method often contain polynomials. In general, convergence regions of power series are small, thus acceleration techniques are often needed to enlarge convergence regions. This is mainly due to the fact that power series is often not an efficient set of base functions to approximate a nonlinear problem, but unfortunately Adomian's decomposition method does not provide us with freedom to use different base functions. Generally speaking, all traditional non-perturbation methods, such as Lyapunov's artificial small parameter method, the δ -expansion method, and Adomian's decomposition method, cannot guarantee the convergence of approximation series.

In summary, both of the perturbation techniques and the non-perturbative methods talked above themselves cannot provide us with a simple way to adjust or control the convergence region and rate of given approximate series. The efficiency to approximate a nonlinear problem has not been taken into enough account, and therefore it is necessary to develop some new analytic methods have the following advantages.

- It is valid even if a given nonlinear problem does not contain any small/large parameters at all.
- It can be employed to efficiently approximate a nonlinear problem by choosing different sets of base functions.
- It itself can provide us with a convenient way to adjust and control the convergence region and rate of approximation series when necessary.

In theory, the analytical approximation method with the above advantages should not be unique. In this entry, we introduce an analytical approximation method based on the basic concept of homotopy topology theory, namely homotopy analysis method (HAM) (Liao 1992, 1997a, b, 1999, 2003, 2004, 2009, 2010a, b, 2011a, b; Liao and Tan 2007; Liao and Zhao 2016). Since its introduction, the method has been gradually developed and continuously improved in application. It has initially formed a relatively complete theoretical system, and has been successfully applied in many fields of science and engineering, and even found new solutions which are not found by other analytical methods and numerical methods of some nonlinear problems.

The Basic Idea of HAM

In 1992, Liao (1992) proposed a preliminary framework for homotopy analysis method (HAM) based on the homotopy theory in topological theory. In order to ensure the convergence of the series solution, Liao (1997a, b) further generalized the zero-order deformation equation by introducing a nonzero auxiliary parameter $\hbar$ in it. This nonzero auxiliary parameter $\hbar$ which is called the “convergence-control parameter” is replaced by c_0 to avoid confusion of $\hbar$ meaning in the later researches. The introduction of the “convergence-control parameter” $\hbar$ (c_0) is a milestone for the HAM. And then, in 2003, Liao (2003) introduced a nonzero auxiliary function $H(\mathbf{r}, t)$ into the so-called zero-order deformation equation which makes the zero-order deformation equation more general. The basic ideas of HAM are explained below.

Deformation Equations

For a nonlinear equation,

$$\mathcal{N}[u, (\mathbf{r}, t)] = f(\mathbf{r}, t). \quad (1)$$

In the framework of homotopy analysis method (HAM) which was proposed by using the homotopy, a basic concept in topology, we first construct a family of solution $\varphi(\mathbf{r}, t; q)$ in

$q \in [0, 1]$ by means of the so-called zeroth-order deformation equation:

$$(1 - q)\mathcal{L}[\varphi(\mathbf{r}, t; q) - u_0(\mathbf{r}, t)] = c_0 q H(\mathbf{r}, t) \mathcal{N}[\varphi(\mathbf{r}, t; q) - f(\mathbf{r}, t)], \quad (2)$$

where $\mathcal{L}$ is an auxiliary linear operator with the property $\mathcal{L}[0] = 0$, $\mathcal{N}$ is the nonlinear operator related to the original Equation (1), $q \in [0, 1]$ is the embedding parameter in topology (called the homotopy parameter), $\varphi(\mathbf{r}, t; q)$ is the solution of (2) for $q \in [0, 1]$, $u_0(\mathbf{r}, t)$ is an initial guess, $c_0 \neq 0$ is the so-called “convergence-control parameter,” and $H(\mathbf{r}, t)$ is an auxiliary function that is nonzero almost everywhere, respectively. Note that, in the framework of HAM, we have great freedom to choose the auxiliary linear operator $\mathcal{L}$, the initial guess $u_0(\mathbf{r}, t)$, the auxiliary function $H(\mathbf{r}, t)$, and the value of the “convergence-control parameter” c_0 .

When $q = 0$, due to the property $\mathcal{L}[0] = 0$, from (2), we have the solution

$$\varphi(\mathbf{r}, t; 0) = u_0(\mathbf{r}, t). \quad (3)$$

When $q = 1$, since $c_0 \neq 0$ and $H(\mathbf{r}, t) \neq 0$ almost everywhere, Eq. (2) is equivalent to the original nonlinear Equation (1) so that we have

$$\varphi(\mathbf{r}, t; 1) = u(\mathbf{r}, t), \quad (4)$$

where $u(\mathbf{r}, t)$ is the solution of the original Equation (1). Thus, as the homotopy parameter q increases from 0 to 1, the solution $\varphi(\mathbf{r}, t; q)$ of Eq. (2) varies (or deforms) continuously from the initial guess $u_0(\mathbf{r}, t)$ to the solution $u(\mathbf{r}, t)$ of the original Equation (13). For this sake, Eq. (2) is called the zeroth-order deformation equation.

Here, it must be emphasized once again that we have great freedom and flexibility to choose the auxiliary linear operator $\mathcal{L}$, the auxiliary function $H(\mathbf{r}, t)$, and especially the value of the convergence control parameter c_0 in the zeroth-order deformation Equation (2). In other words, the solution $\varphi(\mathbf{r}, t; q)$ of the zeroth-order deformation Equation (2) is also dependent upon all of the auxiliary linear operator $\mathcal{L}$, the auxiliary function $H(\mathbf{r}, t)$, and the convergence-control parameter c_0 as a whole, even though they have no physical meanings. Assume that $\mathcal{L}$, $H(\mathbf{r}, t)$, and c_0 are

properly chosen so that the solution $\varphi(\mathbf{r}, t; q)$ of the zeroth-order deformation Equation (2) always exists for $q \in (0, 1)$ and besides it is analytic at $q = 0$, and that the Maclaurin series of $\varphi(\mathbf{r}, t; q)$ with respect to q , i.e.,

$$\varphi(\mathbf{r}, t; q) = u_0(\mathbf{r}, t) + \sum_{m=1}^{+\infty} u_m(\mathbf{r}, t) q^m, \quad (5)$$

converges at $q = 1$. Then, due to (4), we have the approximation series

$$u(\mathbf{r}, t) = u_0(\mathbf{r}, t) + \sum_{m=1}^{+\infty} u_m(\mathbf{r}, t). \quad (6)$$

Substituting the series (5) into the zeroth-order deformation Equation (2) and equating the like-power of q , we have the high-order approximation equations for $u_m(\mathbf{r}, t)$, called the m th-order deformation equation

$$\begin{aligned} \mathcal{L}[u_m(\mathbf{r}, t) - \chi_m u_{m-1}(\mathbf{r}, t)] \\ = c_0 H(\mathbf{r}, t) R_{m-1}(\mathbf{r}, t) \end{aligned} \quad (7)$$

Where

$$R_k(\mathbf{r}, t) = \frac{1}{k!} \left\{ \frac{\partial^k}{\partial q^k} \left(\mathcal{N} \left[\sum_{n=0}^{+\infty} u_n(\mathbf{r}, t) q^n \right] - f(\mathbf{r}, t) \right) \right\} \Big|_{q=0}, \quad (8)$$

with the definition

$$\chi_k = \begin{cases} 0, & \text{when } k \leq 1 \\ 1, & \text{else.} \end{cases} \quad (9)$$

For various types of nonlinear equations, it is easy and straightforward to use the theorems proved in Chap. 4 of Liao's book (Liao 2011a) to calculate the term $R_k(\mathbf{r}, t)$ of the high-order deformation Equation (7).

When we get the m th-order deformation equations and the initial approximation $u_0(\mathbf{r}, t)$, the auxiliary linear operator $\mathcal{L}$ and the auxiliary function $H(\mathbf{r}, t)$ are properly chosen, the $u_m(\mathbf{r}, t)$ can be got order by order. The solution of the high-order

deformation Equation (7) can be expressed in the form

$$\begin{aligned} u_m(\mathbf{r}, t) = -\chi_m u_{m-1}(\mathbf{r}, t) \\ + \mathcal{L}^{-1}[c_0 H(\mathbf{r}, t) R_{m-1}(\mathbf{r}, t)]. \end{aligned} \quad (10)$$

where $\mathcal{L}^{-1}$ is the inverse operator of $\mathcal{L}$. At this time, we get the solution $u(\mathbf{r}, t)$ containing the "convergence-control parameters" c_0 . By choosing the appropriate c_0 , we can get a fast convergence solution.

Base Function and Some Fundamental Rules

It is well known that a real function $u(\mathbf{r}, t)$ can be approximated by many different base functions and thus can be more efficiently approximated by a relatively better set of base functions. For example, if $u(t) = 1/(1+t)$ is expressed in power series, we have

$$u(t) \sim 1 - t + t^2 - t^3 + \dots \quad (11)$$

which only converges in the region of $|t| < 1$. But, if we choose

$$1, \frac{1}{1+t}, \frac{1}{(1+t)^2}, \frac{1}{(1+t)^3}, \dots \quad (12)$$

as the base function, the exact solution can be got by only one item. The type of base function is therefore rather important for the efficiency of approximating a nonlinear problem. The key of efficiently approximating a given nonlinear problem is to choose a relatively better set of base function. Different from the perturbation and non-perturbation methods, the first step to apply HAM is choosing a proper base function. In many cases, by mean of analyzing its physical background and/or its initial/boundary conditions and/or its type of nonlinearity, we might know what kinds of base functions are proper to represent the solution, even without solving a given nonlinear problem. In this entry, let

$$\{e_k(\mathbf{r}, t) | k = 0, 1, 2, \dots\} \quad (13)$$

denote such a set of base function for the illustrative nonlinear problem (1). We can represent the solution in a series

$$u(\mathbf{r}, t) = \sum_{n=0}^{+\infty} c_n e_k(\mathbf{r}, t), \quad (14)$$

where c_n is a coefficient. As mentioned above, a real function $u(\mathbf{r}, t)$ might be expressed by many different base functions. Thus, there might exist some different kinds of solution expressions and all of them might give accurate approximations for a given nonlinear problem. In this case, we might gain the best one by choosing the best set of base function. The solution Expression (14) is the starting point of the HAM and is of great significance in the framework of the HAM.

When we choose the set of base function of the nonlinear problem, the solution expression is determined. Then, based on the solution expression, we can choose the initial approximation $u_0(\mathbf{r}, t)$, the auxiliary linear operator $\mathcal{L}$, and the auxiliary function $H(\mathbf{r}, t)$ by some principles. In Chap. 2 of Liao's book (Liao 2003), Liao proposed the following three principles to guide the selection of $u_0(\mathbf{r}, t)$, $\mathcal{L}$, and $H(\mathbf{r}, t)$:

1. Rule of solution expression: The auxiliary function $H(\mathbf{r}, t)$, the initial approximation $u_0(\mathbf{r}, t)$, and the auxiliary linear operator $\mathcal{L}$ must be chosen in such a way that all solutions of the corresponding high-order deformation equations exist and can be expressed by this set of base functions.
2. Rule of coefficient ergodicity: All coefficients in the solution expression, such as c_n in (14), can be modified to ensure the completeness of the set of base functions.
3. Rule of solution existence: The high-order deformation equations should be closed and have solutions.

The so-called *rule of solution expression*, *rule of coefficient ergodicity*, and *rule of solution existence* play important roles and greatly simplify the application of the HAM.

The "Convergence-Control Parameter" c_0

Independent of small/large physical parameters, the HAM provides us great freedom. Using this kind of freedom, we introduce a nonzero auxiliary parameter c_0 , namely the "convergence-control parameter," into the so-called zeroth-order deformation Equation (2). When a set of base function is determined and the initial approximation $u_0(\mathbf{r}, t)$, the auxiliary linear operator $\mathcal{L}$ and the auxiliary function $H(\mathbf{r}, t)$ are all be chosen properly. Note that, the high-order deformation Equation (7) contains the convergence-control parameter c_0 . Therefore, we can get the HAM series which also contain the "convergence-control parameter" c_0 . More importantly, we have great freedom to choose the value of c_0 so that we can find some proper values of c_0 to guarantee the convergence of the HAM series. Thus, the "convergence-control parameter" c_0 provides us a convenient way to guarantee the convergence of HAM series. Note that the introduction of the so-called "convergence-control parameter" c_0 in the zeroth-order deformation Equation (2) is a milestone for the HAM.

From physical viewpoint, the "convergence-control parameter" c_0 has no physical meanings so that convergent series of solution given by the HAM must be independent of c_0 . This is indeed true: there exists such a region $\mathbf{R}_c$ that, for arbitrary $c_0 \in \mathbf{R}_c$, the HAM series converges to the true solution of the original Equation (1), as illustrated by Liao (2003, 2011a). However, if $c_0 \notin \mathbf{R}_c$, the solution series diverges! So, from a mathematical viewpoint, the "convergence-control parameter" is a key point of the HAM, which provides us a convenient way to guarantee the convergence of the solution series (6). In fact, it is the "convergence-control parameter" c_0 that differs the HAM from all other analytic methods.

At the m th-order of approximation, the optimal value of the convergence-control parameter c_0 can be determined by the minimum of residual square of the original governing equation, i.e.,

$$\frac{d\epsilon_m}{dc_0} = 0, \quad (15)$$

where

$$\varepsilon_m = \int_{\Omega} \left\{ \mathcal{N} \left[\sum_{n=0}^{+\infty} u_n(\mathbf{r}, t) q^n \right] - f(\mathbf{r}, t) \right\}^2 d\Omega. \quad (16)$$

Besides, it has been proved by Liao (2003) that a homotopy series solution (6) must be one of solutions of considered equation, as long as it is convergent. In other words, for an arbitrary convergence-control parameter $c_0 \in \mathbf{R}_c$, where

$$\mathbf{R}_c = \left\{ c_0 : \lim_{m \rightarrow +\infty} \varepsilon_m(c_0) \rightarrow 0 \right\}, \quad (17)$$

is an interval, the solution series (6) is convergent to the true solution of the original Equation (13). For details, please refer to Liao and Chap. 3 of his book (Liao 2011a). Up to now, as the convergence-control parameter c_0 is determined, the convergence solution $u(\mathbf{r}, t)$ can be got successfully based on the HAM.

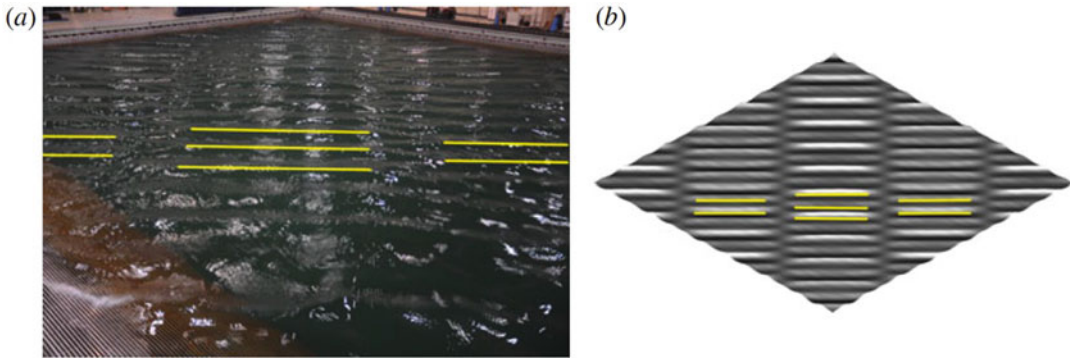
Applications of HAM

HAM is an analytic technique for highly nonlinear problems and has been successfully applied for nonlinear problems in various areas which governed by various types of ordinary differential equations (ODEs) and partial differential equations (PDEs), such as boundary-layer flows (Liao 2005; Liao and Magyari 2006), finance (Cheng et al. 2010; Zhong and Liao 2017a; Zhu 2006), water waves (Liao 2011b; Liao et al. 2016; Liu and Liao 2014; Liu et al. 2018; Xu et al. 2012; Yang et al. 2018; Zhong and Liao 2018b), and so on (Zhong and Liao 2017b, 2018a). Here we mainly introduce the applications of HAM on water waves which are governed by complicated nonlinear PDEs.

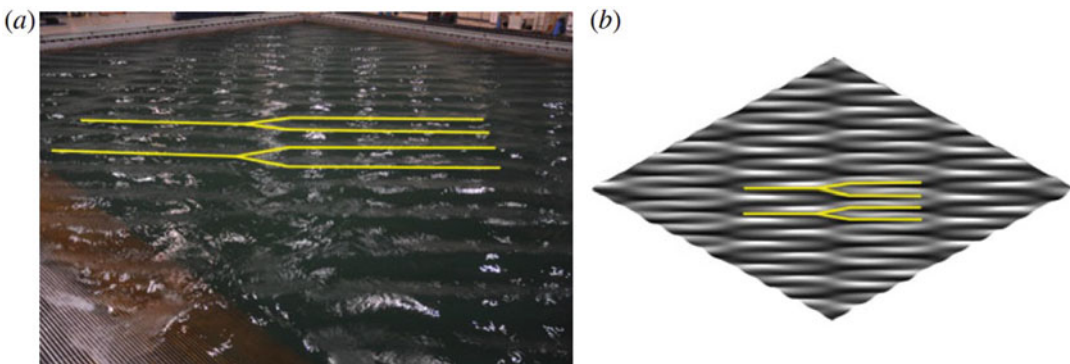
Liao (2011b) considered the nonlinear interactions between pairs of intersecting wave train in deep water which exactly satisfied the resonance criterion and found steady-state resonant gravity waves by means of the HAM for the first time, to the best of our knowledge. Then it was found that the steady-state resonant gravity waves exist not only in infinite water depth but also in finite water

depth by Xu et al. (2012). Note that Liao (2011b) and Xu et al. (2012) only consider a special quartet. Liu and Liao (2014) further extended the existing results for steady-state resonance from a special quartet to more general and coupled quartets, together with consideration of higher-order resonant interactions. The significant role of the near resonance was revealed. It was found that all near-resonant components as a whole contain more and more wave energy, as the wave patterns tend from two-dimensional to one-dimensional, or as the amplitudes of the steady-state resonant wave system increase, as pointed out by Liu and Liao (2014). What is exciting is that the existence of steady-state resonance waves were confirmed experimentally by Liu et al. (2015). These steady-state resonant waves are first calculated theoretically under the exact resonance criterion with sufficiently high nonlinearity, and then are generated in the basin at the State Key Laboratory of Ocean Engineering (SKLOE) in Shanghai by means of the main wave components that contain at least 95% of the wave energy. The comparison of experimental results and theoretical results are shown in Figs. 1 and 2. In addition, about the steady-state gravity waves, Liao et al. (2016) successfully applied the HAM to obtain the steady-state nearly resonant gravity waves in deep water, and Liu et al. (2018) found finite amplitude steady-state waves with multiple near-resonant interactions. Recently, the steady-state acoustic-gravity waves were also obtained by Yang et al. (2018). It was found that the dynamic pressure on the sea bottom caused by the resonant hydroacoustic wave component is much larger than that in the case of non-resonance, which might even trigger microseisms of the ocean floor. This will greatly deepen and enrich our understandings about resonant waves.

Moreover, on the limiting Stokes wave of extreme height in arbitrary water depth, Zhong and Liao (2018b) gained convergent results of the limiting Stokes waves with this kind of sharp crest in arbitrary water depth, even including solitary waves of extreme form in extremely shallow water, based on HAM and found that, in the frame of the HAM, the Stokes wave is a unified theory for all kinds of progressive waves, even including



Homotopy Analysis Method, Fig. 1 Perspective plot of the free-surface in case S1 in (Liu et al. 2015). Crests in the middle region are marked as yellow lines. (a) Experimental result; (b) theoretical result



Homotopy Analysis Method, Fig. 2 Perspective plot of the free surface for case S3 (Liu et al. 2015). Crests in the middle region are marked as yellow lines. (a) Experimental result; (b) theoretical result

the limiting (extreme) solitary waves with a sharp crest of 120° angle in extremely shallow water, as shown in Fig. 3.

All of these successful applications show the originality, validity, and generality of the HAM for nonlinear problems in science and engineering.

Conclusion

As mentioned above, the HAM has the following advantages:

- It is independent of any small/large physical parameters.
- It provides us great freedom and large flexibility to choose equation type and solution

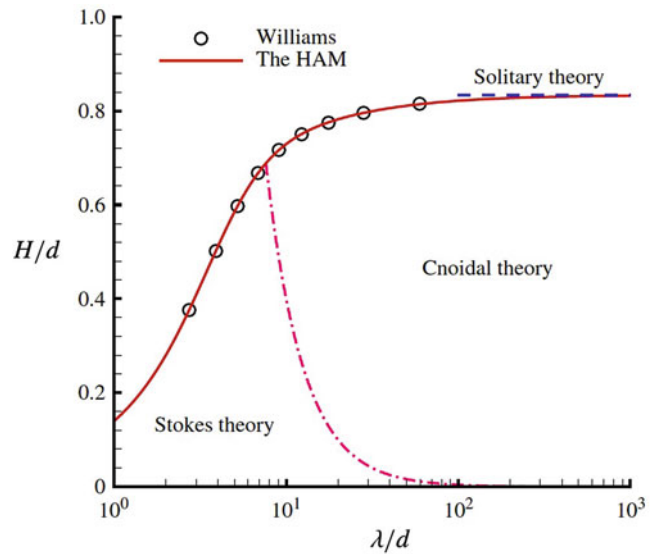
expression of linear high-order approximation equations.

- It provides us a convenient way to guarantee the convergence of approximation series.

In this way, nearly all restrictions and limitations of the traditional perturbation method and non-perturbation methods (such as Lyapunov's artificial small parameter method, the δ -expansion method, the Adomian's decomposition method, and so on) can be overcome by means of the HAM. Besides, it has been generally proved that the Lyapunov's artificial small parameter method, the Adomian's decomposition method and the δ -expansion method are only special cases of the HAM for some specially chosen auxiliary linear operator $\mathcal{L}$ and convergence-control parameter c_0 . Thus, the HAM is more general in theory.

Homotopy Analysis Method,

Fig. 3 Comparison of H/d versus λ/d . Williams (1981): circle; the first-order HAM-based iteration approach: solid line; demarcation line between Stokes and cnoidal theories, the Ursell number (Hedges 1995): dash-dot line; for solitary wave (Hunter and Vanden-Broeck 1983): dashed line



The HAM provides a useful analytic tool to investigate highly nonlinear problems with multiple solutions and singularity in science, finance, and engineering. Based on HAM, we have truly got new solutions that were even missed by numerical methods of some nonlinear problems. It shows us the validity and great potential of HAM and encourages us to apply the HAM to attack some challenging nonlinear problems.

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HPAL

- [Mineral Processing and Metallurgy](#)

HSVA, Hamburg Ship Model Basin

- [Dynamic Positioning in Ice](#)
- [Moored Ship in Ice](#)

Hull Resistance

- [AUV/ROV/HOV Resistance](#)

Hull Structure Design

- [Ship Structural Design](#)

Human Factors

- [Human Factors in the Role of Energy Efficiency](#)

Human Factors in Ship Design

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Synonyms

[Ergonomics](#)

Definition

Human factors is the application of psychological and physiological principles to the engineering and design of products, processes, and systems. The goals of human factors are to reduce human error, increase productivity, and enhance safety and comfort with a specific focus on the interaction between human and things of interest.

In the nineteenth century, Frederick Taylor pioneered the “scientific management” method, which proposed a way to find the optimum method of carrying out a given task. Frank and Lillian expanded Taylor's methods in the early 1900s to develop the “time and motion study.” They aimed to improve efficiency by eliminating unnecessary steps and actions.

Human factors is a broad-based discipline and has always played a role in the design and operation of ship. However, advancing and applying human factors has been an uneven and thorny process throughout history. Knowledge to be permanently embedded in practice has been commonly discarded and must be relearned, often at great human cost.

The dawn of the Information Age has resulted in the related field of human–computer interaction. Likewise, the growing demand for and competition among marine products have resulted in more companies and industries including human factors in their product design.

Scientific Fundamentals

Human factors is a combination of numerous disciplines, such as psychology, sociology, engineering, biomechanics, industrial design, physiology, anthropometry, interaction design, visual design, user experience, and user interface design (Mccormick and Sanders 2002). In essence, it is the study of designing equipment, devices, and processes that fit the human body and its cognitive abilities.

Human factors is employed to fulfill the goals of occupational health and safety and productivity. It is relevant in the design of such things as safe furniture and easy-to-use interfaces to machines and equipment.

Human factors is concerned with the “fit” between the user, equipment, and environment. It accounts for the user's capabilities and limitations in seeking to ensure those tasks, functions, information, and the environment that suit the user.

There are many specializations within these broad categories. Specializations in the field of physical ergonomics may include visual ergonomics. Specializations within the field of cognitive ergonomics may include usability, human–computer interaction, and user experience engineering.

Physical ergonomics is the science of designing user interaction with equipment and workplaces to fit the user. It is concerned with human anatomy and some of the anthropometric, physiological, and biomechanical characteristics as they relate to physical activity.

Cognitive ergonomics is concerned with mental processes, such as perception, memory, reasoning, and motor response, as they affect interactions among humans and other elements of a system.

Organizational ergonomics is concerned with the optimization of sociotechnical systems, including their organizational structures, policies, and processes (Fig. 1).

Key Applications

The primary consideration in the safe design and operation of ship is the “human factors,” the detrimental impact on safety of the ship, its crew, cargo, and the maritime environment.

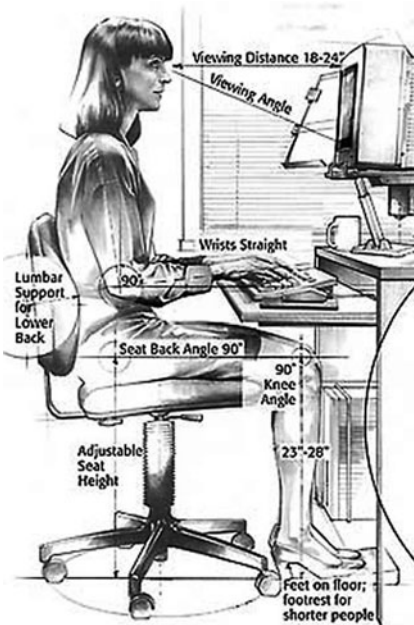
The ship architect, marine engineer, other design specialists, analysts, and operators have influence over a number of important design elements that can enhance human factors in a ship. Examples of these elements include the following: hull form (ship motions), structure, machinery (vibration, noise, condition), general arrangements (accommodation, life at the sea), etc.

Hull Form

Ship includes all ships and craft which move on, under or across the surface of the water. Certain hull types are by their nature more stable than others or more prone to produce motions that

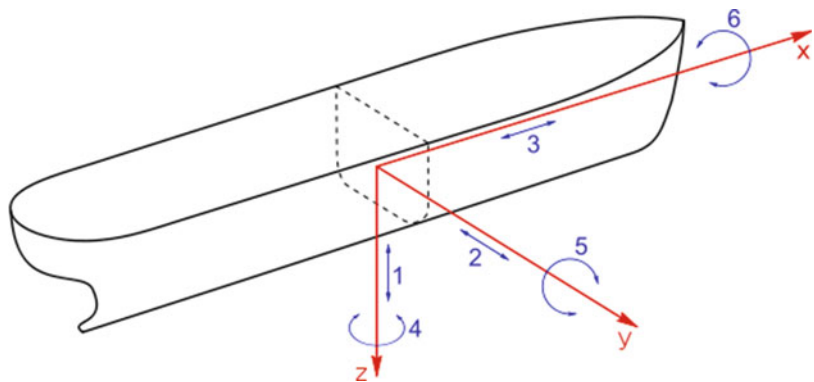
enhance operator performance and embarked personnel comfort.

At the design progresses, there are six disciplines that should be considered: avoid frequencies between 0.125 and 0.250 Hz; avoid high impact (shock); appropriate length, beam (hull and overall), and centers of gravity and rotation; sufficient wet deck clearance; appropriate for different operating configurations; and appropriate for expected sea conditions (Figs. 2 and 3).



Human Factors in Ship Design, Fig. 1 Physical ergonomics: the science of designing user interaction with equipment and workplaces to fit the user (https://en.wikipedia.org/wiki/Human_factors_and_ergonomics)

Human Factors in Ship Design, Fig. 2 Ship's six freedom motion: 1. heave, 2. sway, 3. surge, 4. yaw, 5. pitch, 6. roll (<https://en.wikipedia.org/wiki/Ship>)



Structure

This section provides three guidance materials to help ensure continued consideration of human factors. Examples of these rules include the following: avoid frequencies between 4 and 0 Hz, maintain accelerations below 0.315 m/s^2 , survive worst-case loading.

Machinery

In this part of work, approaches for enhancing human factors are listed below: decreased vibration and airborne noise isolation, condition monitoring, and hearing protection.

Airborne noise often begins as structural vibration caused by diesel engines, and is transmitted directly or through a combination of structure and air into surrounding compartments.

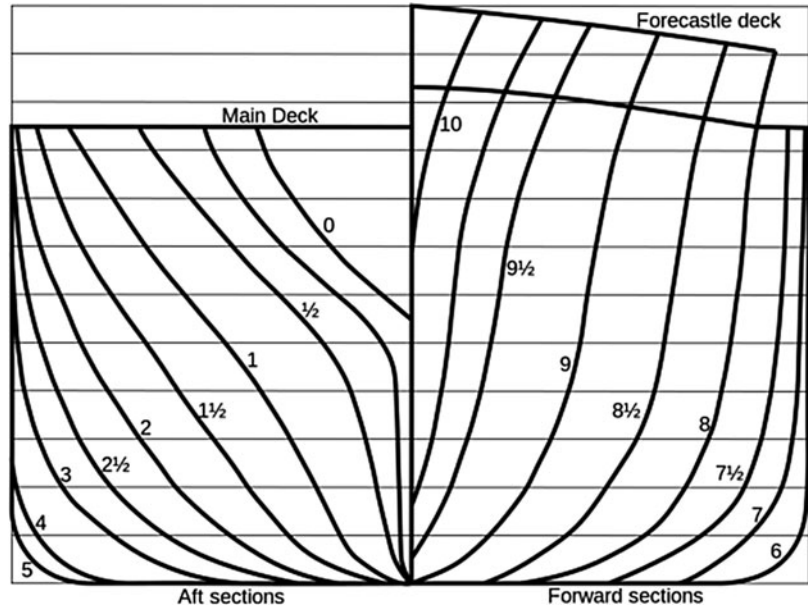
The vibration may be low frequency, such as when a vehicle meets waves, or high frequency, such as that driven by heavy rotating machinery (Fig. 4).

General Arrangement

Ship is limited in their capacity to carry weight or enclose volume and to generate propulsion and service power. Limitations and potential design responses are developed and compared during the early phases of a ship design. These "trade studies" or "analyses of alternatives" consider numerous technical requirements and recently have begun to consider human factors requirements as well.

Accommodations do not include machinery spaces or operational spaces. In designing accommodation spaces, consideration should be given to

Human Factors in Ship Design, Fig. 3 Body plan of a ship showing the hull form (https://en.wikipedia.org/wiki/Ship)



Human Factors in Ship Design, Fig. 4 A ship's engine room: the main source of ship vibration and noise (https://en.wikipedia.org/wiki/Ship)



stateroom deck area, and the size, shape, and fittings associated with doors, stairs, furniture, and corridors. Stairs tend to be steeper, and there is more noise and vibration on ship. Other considerations with regard to accommodations include

ship motion, lighting, temperature, ventilation, and colors (Fig. 5).

Access refers to the means provided for people to move about the ship in vertical and horizontal directions. The designers make trade as



Human Factors in Ship Design, Fig. 5 Accommodations onboard (<https://cn.bing.com/images>)



Human Factors in Ship Design, Fig. 6 Evacuation onboard (<https://cn.bing.com/images>)

whether to maintain watertight integrity with a high skilled, dogged steel door tithe weather deck or easy passage with a nontight swinging door. Safety against the sea, weather, hostile combat action, and the ability to move about the ship unhindered are all design considerations within the realm of access.

Safety at sea is always a concern; designers can enhance safety by numerous means including reinforced structural closures, additional handholds in showers and in dimly lit public areas, decreasing or eliminating deck height changes, marking leading edges of stair steps, ensuring life jackets are easily available, and including stateroom balconies in fire boundaries (Figs. 6 and 7).

Life at Sea is defined by routine. Training exercises encompass fire, flooding, and simulated combat. Some training is grueling and as close to

the mental and physical rigors of combat as the instructors can simulate.

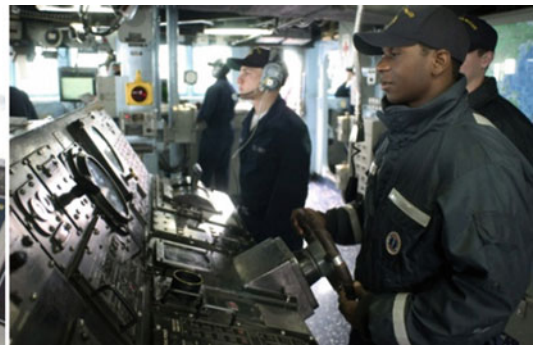
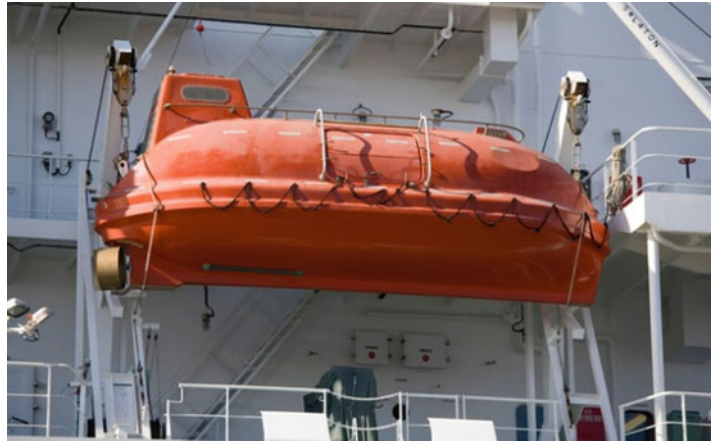
Symptomatic areas of concern are those events or conditions that affect a person's comfort or performance and may be describable only by the person involved in symptomatic areas of concern. There are five symptomatic areas of concern relevant to the design and operation of marine vehicles: motion sickness, spite syndrome, fatigue, sleep loss, and human error.

Human stressors are physically measurable events or stimuli that affect the comfort or performance of a person.

Human–Machine Interface

The human–machine interface is where communication and physical interaction occur between the individual operator or embarked person and a particular component of the ship. Paying attention

Human Factors in Ship Design, Fig. 7 Lifeboat onboard (<http://image.baidu.com/>)



Human Factors in Ship Design, Fig. 8 The human-machine interface (<http://image.baidu.com/>)

to the human-machine interface during the design process will enhance personnel comfort and operational effectiveness, information availability, safety, appropriate automation, and commonality of instruments and control commonality, and decrease fatigue, human error, duplication of effort among crew members, and training requirements (Fig. 8).

Future Trends

Human factors is recognized as an important element of design process, though presently this element receives much more lip service than funded tasking in both the naval and commercial worlds. That said, navies and commercial owners are beginning to increase their funding to human factors as a way to improve operator performance and enhance the comfort of embarked personnel.

Several trends pull human factors to the forefront of marine vehicle design and operation:

Crew sizes: The first of these trends, observed by all in the marine industry, is the move toward reduced crew sizes. This trend is driven by the high cost of qualified personnel and the increasing capability and availability of automation, which can replace crew personnel with remote sensors and controls. The trend is further advanced through enhanced maintenance practices, which focus on less maintenance at sea by the crew, and the change from periodic maintenance at set time intervals to condition-based maintenance and requirement-based maintenance. Interlaced with this trend are issues involving safety and habitability.

Computer capabilities: The second trend is increased computer capabilities, in the field of

design in general, and also in the specialized fields of computer-aided modeling and simulation. Simulation enables the prediction of technical parameters such as speed, power, and ship motions. Simulation also has advanced in the field of human performance, with models addressing task allocation among vehicle operators, as well as personnel performance under stressors such as sleep loss, noise, and temperature extremes. Finally, the capability to link software programs has increased exponentially in recent years, enabling designers to communicate between ship behavior programs and human performance programs.

Safety considerations: Maritime operating organizations have come to realize that the vast majority of incidents and accidents are caused by human errors, either by designers or operators. These organizations are concerned with injuries and deaths, and are beginning to apply human factors techniques to increase safety.

Regulatory changes: Likewise, regulatory agencies have studied the trends, have recognized the role of human factors, and are revising regulations to help prevent incidents and accidents that can cause human and environmental losses.

Human factors research and development are expensive in the larger sense, but acceptable on the incremental level, project by project. Thus, human factors advancement will probably continue to be evolutionary and not revolutionary. The following trends appear reasonable:

- (a) Reduced crew size coupled with increased automation – This overarching trend, motivated by life cycle cost reductions, will likely continue, with more and more human functions replaced by automation.
- (b) Performance predictions – Future research may be aimed at developing techniques to better understand the relationships among motion, fatigue, sleep loss, and spaced syndrome and be better able to predict the resulting drops in operator performance and personnel comfort (Bri2003).
- (c) Enhanced interoperability – Interoperability issues will be addressed, enabling an

increased flow of information among commercial and naval operators (Clark and Moon 2001).

- (d) Operator performance linked to vehicle sea keeping – This integration will enable designers to predict the effect of design changes on operator performance, including cognitive performance, physical performance, and predictions of MSI and MII (Colwell 2006).
- (e) Augmented cognition – The present human–computer interface will evolve to a human–computer symbiosis system that includes artificial intelligence. In this system, extensive information will be available to the human in real time. Distributed, autonomous systems, intelligent agent architectures, and wireless communication will help make the operator aware of critical information upon which to base decisions (Masakowski 2008).
- (f) Predictive models for effective seat design – Improved spinal injury models will become available to better predict the health effects of repeated impact loading in high-speed craft in rough seas (Bass et al. 2005a, b; Peterson et al. 2004).
- (g) Enhancement of bridge controls and arrangement – Human factors considerations with regard to bridge controls and arrangement will continue to advance, with software increasing in sophistication and flexibility, allowing monitors and controls to be more readily adapted on the fly for individual users and specific missions or situations (Bowdler et al. 2005, Widdel and Motz 2000).
- (h) Ship evacuation – Particularly from ships carrying many personnel (e.g., cruise ships, aircraft carriers) – is an area with potential for safety improvement. Areas of investigation could include the effect of lifejackets on personnel mobility (Igloliorte et al. 2006). The results can be used to further validate and enhance programs such as maritime EXODUS (Earl 2002).
- (i) Robotic fire fighters – One idea that could place the human out of harm's way is to employ firefighting robots inside the danger

zone, backed up by human operators a safe distance away (Sheridan 1992).

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Human Factors in the Role of Energy Efficiency

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Synonyms

Awareness; Energy consumption management; Energy efficiency; Environmental impact; Fuel consumption; Human factors; Optimal ship operations

Definition

Energy is an important source for economic and social development in the world. It is even more indispensable in today's fast-growing transportation industry. With the development of transportation in the past few decades, carbon dioxide emissions have increased significantly. The shortage of energy and the emission of waste heat from waste gas have intensified, making the importance and necessity of energy saving and emission reduction of ships particularly remarkable (ABS 2015). According to the main factors affecting ship energy consumption and emission reduction, some methods for energy saving and emission reduction of ships can be divided into ship design, construction, operation, and management (Bailou 2013). In addition to the technical development of those energy efficiency measures, how much benefits of installing an energy efficiency measure can be obtained by a ship strongly depends on the crew members who directly operate these technologies. For maritime transport, more than 60% of accidents are attributed to human-related operations (UNCTAD 2019). Therefore, a good understanding and utilization of the role of human factors in shipping energy efficiency will greatly increase

the working efficiency of those energy efficiency measures to be implemented by ships.

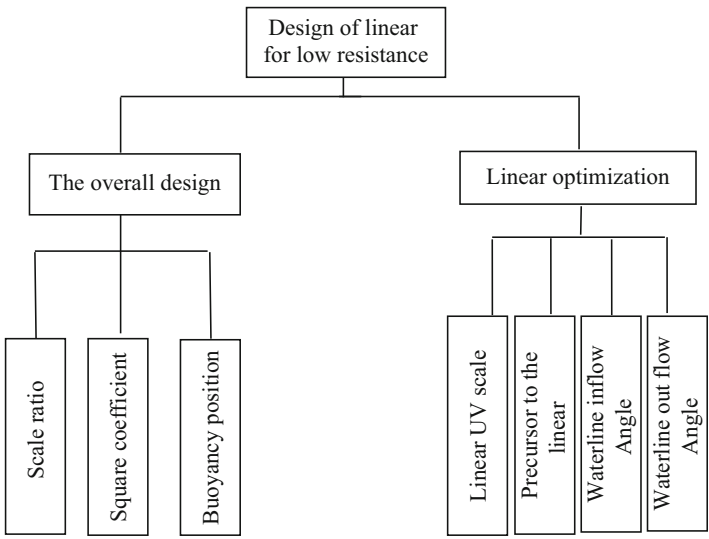
Technical Improvement of Energy Efficiency

From traditional naval architecture perspective, to increase a ship’s energy efficiency is to reduce a ship’s sailing resistance at sea and increase the propulsive efficiency from marine engine and propellers, e.g., Harvald (1983), Lewis (1988), etc. A straightforward approach to increase a ship’s energy efficiency is through better ship design and construction (Bailou 2013). In the design process, a workflow to optimize a ship’s hull geometry is presented in Fig. 1. Linear optimization refers to the design of the ship’s hull line type (U- or V-type), water line inflow, and outflow angles (Holtrop and Mennen 1982). The current design optimization process can use model tests and CFD (computational fluid dynamics) methods to adjust a ship’s hull geometry, repeatedly (Seo et al. 2013; ITTC 2014). Then the final hull geometry of the design ship will be optimized with relative low resistance in a ship’s actual sailing environment. Furthermore, another way to increase a ship’s energy efficiency is to improve the propulsion efficiency

of the ship mainly by improving the tail flow field and the power consumption of the main engine (DNV GL 2014).

The improvement of the tail semi-flow field refers to the installation of the propeller fairing on the ship hull. Moreover, currently existing ship can choose to install high-efficiency and energy-saving power equipment, including main and auxiliary engines, boilers, and drive shaft systems. At present, the diesel engine as the main propulsion device tends to develop in the direction of low speed, long stroke, high pressure, high compression, and electronic injection and can burn inferior fuel oil. Since the diesel engine is working under partial load, its economy is poor, and the fuel consumption rate is increased. When some ships are working under partial load for a long time, the main shaft can be used with the generator, which not only enables the main machine to work in an efficient operation area and improves efficiency but also reduces the number of diesel generator sets. In addition, some electric heating can be installed to a ship’s fuel pipeline to improve the fuel temperature, reduce the viscosity and surface tension of the fuel, improve the atomization performance, and thus improve the fuel combustion and nitrogen oxide emissions. Other technologies related to marine engine systems to reduce air emissions include installation of a

Human Factors in the Role of Energy Efficiency, Fig. 1 Design of linear for low resistance (Lewis 1988)



homogenizer for the fuel system; adding fuel additives to fuel which can improve fuel quality, improve combustion performance, reduce harmful emissions, improve fuel utilization, and reduce harmful emissions and energy; and development of new alternative fuels such as marine fuels, waste heat recovery systems, etc.

Ship Energy Efficient Management Plan

From human factor perspective, the management strategy of a ship's operation is one of the key steps to increase a ship's energy efficiency (Bännstrand et al. 2016). In the management process, captains, together with the traffic management team, are required to rationally design the route plan and strive for the economy on the premise of ensuring safety, by taking full consideration of the meteorological factors of the route and the voyage's waves, surges, and wind flow factors to avoid disastrous wind/wave conditions and optimally utilizing ocean currents in sailing. According to pirate information, avoid high-risk waters where pirates are infested. For example, if a super-large oil tanker is mainly engaged in the transportation of crude oil from the Middle East or Africa to the Far East, it should not only understand the seasonal weather information of the Indian Ocean but also have a good understanding of the information of the pirate's infested waters. The route design should be particularly serious and meticulous. Although it is possible to protect the safety of ships by avoiding pirate-infested waters, it is also necessary to consider designing the best economic routes. Large oil tankers have high freeboards and are equipped with anti-pirate wires and anti-pirate devices from the front to the rear. When there are more than 5 kn winds in the Indian Ocean and more than 1.5 m, the possibility of pirates will be greatly reduced (OECD 2015). When sailing in the waters of pirates, the navigational watch should be strengthened, and the patrol personnel should be arranged to make emergency plans and countermeasures for possible situations.

Implement an Energy Efficiency Management Plan

Develop and implement a ship energy efficiency management plan for energy efficiency targets and indicators on all ships and adopt the best operational plan according to the ship's own characteristics, navigation area, trade, and other related requirements. It has achieved good results in improving the energy conservation awareness of ship and shore management personnel and reducing fuel consumption.

Electromechanical equipment is the most direct consumer of ship oil materials. Grasp the maintenance of electromechanical equipment, ensure that electromechanical equipment is in good working condition, and fully tap energy saving and emission reduction. Maintain the main engine and auxiliary engine (diesel engine) as required; keep the intake and exhaust system, fuel system, lubrication system, and cooling system in good condition; and regularly clean and repair the exhaust turbocharger, injector, and combustion chamber components. Keep the working conditions of the main and auxiliary machines in good condition and the thermal efficiency in best condition.

Maintain boilers and exhaust gas boilers according to requirements, adjust the correct proportion of wind oil, and regularly clean the oil guns to ensure good combustion. Regularly blown and clean surface of the furnace and collector tubes can ensure good heat transfer effect, and regularly reduce water quality. The heating of the cargo oil tank and the fuel oil tank should also be optimally arranged to reduce the fuel consumption of the oil-fired boiler. Ship crew and onshore management team should use the characteristics of cargo oil and berthing unloading plan, to better arrange cargo heating and unloading operations and minimize the time of cargo oil heating and unloading time to save fuel consumption.

A ship's draft and trim should be optimized by ballast water to ensure the efficiency of the propeller and reduce resistance. An optimal ballast tank sediment management plans, flush regularly, and control sediment accumulation can help to reduce unnecessary energy consumption. When

carrying out ballast water operation, the gravity discharge system should be considered to reduce the fuel consumption of the ballast pump. The ballast process for each voyage requires a ballast plan based on the quality of the ballast water at the port of discharge. In addition, ship crews should regularly check the condition of the hull and propeller, evaluate the condition of the hull and the propeller, and find out that the underwater hull is cleaned in time when the hull has big biofouling; check whether the propeller has foreign matter entangled, and if it should be removed as soon as possible, eliminate the navigation of the ship.

Increase Awareness of Energy and Emission Reduction

Ship management team should help to increase the awareness of energy consumption and emission reduction work for ship crews (Bailou 2013; DNV GL 2014). From legislation perspectives, maritime authorities and environmental policy makers should continuously implement regulations and provide financial subsidies to encourage the early elimination of old fashion energy-intensive systems. Further enhancement of environmental regulations on shipping emission control zones can promote the utilization of energy efficiency technologies and new energy sources and increase awareness of pollutions from shipping from whole transport market perspectives. At present, the major incentive of energy saving and emission reduction of current shipping companies is to save costs, but in terms of long-term interests, most enterprises are not very clear about the significance and importance of energy saving and emission reduction. The most important thing for enterprises to do energy consumption and emission reduction is to make enterprises fully aware of the importance of energy conservation and emission reduction. That is to fully raise awareness of energy saving and emission reduction within the company. It is important to make all company members recognize the importance of energy saving and emission reduction. Shipping companies can introduce more indicators, guidelines, policies, etc., to

highlight the significance of energy saving and emission reduction. Through various channels, such as setting up energy-saving emission reduction bulletins and slogans within shipping companies and grassroots, actively carry out publicity and education activities on energy-saving and emission reduction work within the shipping industry, turning energy saving and emission reduction into another work focus besides pursuing economic benefits. Another efficient way is to create good forum for ship operators to communicate and demonstrate the actual benefits of various energy efficiency measures to reduce fuel consumption and air emissions. It can also provide end-users with good communication and training platforms regarding how to implement these measures.

Optimal Ship Operation

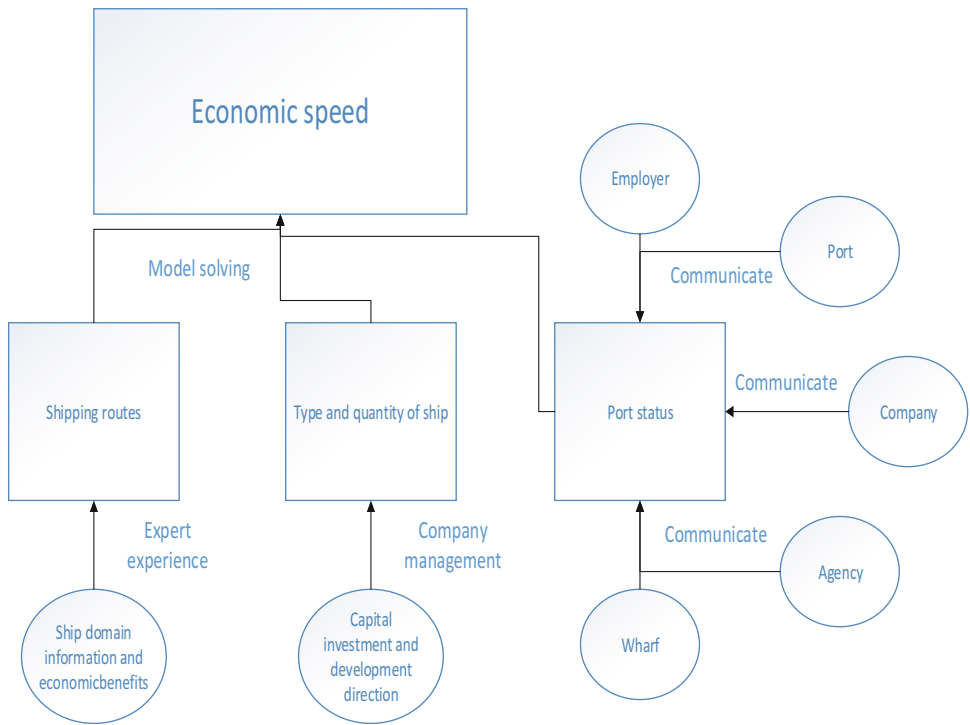
Because of master mariners' lack of capability to continuously adjust a ship's operation profiles (speed/course), ships are not navigated in an energy efficient manner, leading to large waste of fuel cost and emissions (Wang et al. 2017). As the ship size increases, large ships need long time for turning and stalling. Therefore, for an efficient navigation, small steering angles should be used as early as possible to allow the ship speed to drop as much as possible. When sailing in a narrow waterway, the steering is mainly controlled by small adjustment of rudders. When sailing at sea, whenever the steering point is reached, captains usually change to the next heading immediately, leading to significant speed drop after the steering is completed. Therefore, to use the stepwise steering to slowly adjust the ship sailing to the next heading before the arrival of the turning point can also save fuel cost during the sailing at sea. To optimize the operation of cargo loading and unloading can be quite beneficial to reduce the fuel burning and air emissions at the port. The first officer shall be responsible for organizing and formulating a complete plan for loading and unloading cargo according to the relevant loading requirements of the voyage order, ship equipment, personnel, and port status. The deck department

and the engine department should fully communicate with each other to clarify the time for unloading and starting the pump. This management could help to avoid the idling of the cargo pump and make the auxiliary boiler run for a long time and waste fuel. Take a 76,000 DWT tanker, for example. In the case of the same total power load, each additional auxiliary machine will run an additional 1 t of fuel per day. If you stop the unnecessary pumps, oil pumps, and fans during the suspension to save fuel, you can save about 1 t of fuel per day.

Role of Human Factors in Energy Efficiency Measures

For all energy-saving and emission reduction methods, human factors play an indispensable role (IMO 2003). For example, Fig. 2 presents some brief introduction of a ship’s voyage planning process to identify an economic speed for

the navigation. The establishment of the ship economic speed and the route distribution model must include intensively human involvement. Therefore, experts can only rely on their rich knowledge reserves and navigation experience and case analysis of past actual route design to plan and judge the route. It is more conducive to ship navigation and energy saving. In the ship operation process, technical persons are required to carry out data statistics on ship operation information, economic benefits, and the company’s current situation and report it to the management. The management team combines the company’s development policy to discuss, analyze, and try to make a low-cost and high-efficiency plan to benefit the shipping company. The implementation of those planned energy saving, and emission reduction measures requires transparent communications among employees at all different levels. The implementation of energy saving and emission reduction.



Human Factors in the Role of Energy Efficiency, Fig. 2 A brief introduction of a ship’s voyage planning process to determine an economic speed (IMO 2000)

In order to ensure the timeliness of the port information during the voyage, all parties should communicate in a timely manner. Captains are responsible for the communication with the shipping department, optimizing the ship speed according to the loading and unloading plan and rationally formulating the ship replenishment plan, time schedule of parking, loading, unloading, etc. Because of the variability of the actual situations, some plans need to be revised at any time after the multiparty communication gets instant information. The current system is not perfect enough to cover all aspects in the ship operation process. Some modifications are needed even during a ship's sailing stages according to the instant information. In general, it is still necessary to rely on the management of the ship to make plans based on their own professional knowledge and navigation experience. The rationality of the plan largely determines the degree of energy savings.

In the process of managing the inspection and maintenance of a ship's main equipment to achieve energy-saving effect, human participation is of particular importance, such as the maintenance of main engines and the auxiliary engines (diesel engine), the boiler and exhaust gas boilers, regular flushing of ballast water in the optimization of ship ballast to control sediment accumulation, and regular inspection and cleaning of hulls and propellers. During the management process, the crew must be strictly implemented according to the plan, while the responsibility should be assigned to specific crew members rather than any kind of computer systems (Tillig et al. 2017).

In the method of strengthening the awareness of energy saving and emission reduction in shipping, it is the formulation of government policies or the promotion and training of enterprises and various stakeholders of the entire shipping industry. Only by raising the awareness of people can we grow from small details to large implementation of more energy efficiency measures onboard ships. In the worst case, seafarers may not believe (or do not aware the usage of) these energy efficiency measures that are bought by shipping companies to reduce fuel cost and air emissions. The lack of communication between ship operator

management and seafarers can lead to large waste of energy saving potential and investment.

On the other side, if it is promoted properly, the human factors can play a big positive impact to increase a ship's energy saving and reduce emission during her service period. For example, in a ship's design and operation process, simply relying on technical systems cannot fully consider all aspects of ship energy influence factors. Large deviation of a ship's performance characteristics during the design and planning stages can be expected from the ship's actual sailing situations. Manual participation can often be considered more comprehensively, and the proposed scheme can be more practical. Furthermore, information exchange between the onboard crew member and onshore stakeholders such as ship traffic management team, ports, shippers, weather providers, etc. should be more transparent and faster, which can help to adjust a ship's operation plan immediately to reflect the actual sailing status. For example, when the ship is sailing, if needed she can in real time communicate with the port, agent, terminal, cargo owner, and company dispatch and understand the congestion of the port, the time of entry and exit, the berth handling capacity and the plan, etc. All those steps must involve humans from different stakeholders, in order to update original sailing plan according to the real-time situation and improve the navigation and operational efficiency of the ship. In the event of an emergency or an emergency, the worker can respond quickly and in a timely manner.

Furthermore, the involvement of human factors in a ship's navigation process could be enhanced to allow more efficient navigation manner. For example, in today's ocean crossing ships, it is quite common to order weather forecast to guide a ship's navigation. Based on the forecast sea environment conditions, a correct steering mode can be determined to minimize the ship's fuel consumption. In this case, the single auxiliary machine is used as much as possible to reduce the unnecessary power load; the ship should pass the pressure before starting the ship; the water loading regulation of the ship's draft and reasonable trim, to ensure the efficiency of the propeller and

other details of the operation, can only be fully automated, etc.

It can better appeal to the awareness of energy saving and emission reduction in the entire shipping industry. By setting an example for implementing energy-saving and emission-reducing programs and paying attention to energy conservation and emission reduction in detail, it is better to transmit this spirit to others. Starting from the subtleties, the implementation of energy efficiency and emission reduction technologies can be more effectively promoted.

However, if human factors are not properly considered in a ship's operation process, it will lead to negative impact of generating more emissions and cost more fuels during sailing. In the ship design and route planning, even experts may be negligent, because the calculation of some details is not enough for the consideration is not comprehensive. Neglecting some important factors may lead to large deviations, making the result unsatisfactory. When the technicians conduct statistics on ship operation information, economic benefits, and the company's capital status, due to the huge amount of information and complexity, it is more prone to errors, which may lead to the company's management finally getting the wrong information for ship operations. Making wrong decisions leads to a sudden drop in economic benefits and hinders the development of energy efficiency and emission reduction.

Due to the inspection and maintenance schedule of the ship equipment, large amount of human participation is required for the reasonable operation of the ship during navigation. The whole process is prone to errors in management and brings a certain degree of trouble to management planning. Therefore, it is necessary to establish a cumbersome management mechanism system with clear responsibilities to reconcile the artificial relationship between some contradictions. The planning process requires a lot of time and resource investment, while the final inspection and benefit analysis may be also associated with high cost but definitely necessary from end-users to realize the actual benefits. If the process chain is not handled properly, the shipping industry may

become hesitated to invest and implement energy efficiency measures.

Cross-References

- [Impact of Maritime Transport](#)
- [New Technologies in Auxiliary Propulsions](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

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Human Occupied Vehicle (HOV)

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Synonyms

Atmospheric diving suit (ADS); Deep submergence rescue vehicle (DSRV); Manned submersible (MS); Rescue bell

Definition

A **human occupied vehicle** (HOV), also known as **manned submersible** (MS) is one type of **submersibles** which can carry scientists, engineers, and various electronic devices and special equipment to arrive at various deep-sea complex environments quickly and accurately, and conduct efficient exploration, scientific investigation, and other deep sea operations. It is an important technical means to understand the ocean phenomena and to exploit the ocean resources. A typical manned submersible consists of four major components: a pressure hull, propellers (thrusters), buoyant materials, and observational and sampling instruments. Currently there are four types of different HOVs: manned submersible (MS), deep submergence rescue vehicle (DSRV), atmospheric diving suit (ADS), and rescue bell.

Scientific Fundamentals

Historical Development

Human curiosity for the natural world and human exploitation of ocean resources are the first two factors for the progress of deep ocean exploration while man's quest to achieve mobility beneath the surface of the sea for military purposes is the third but the most important factor to speed up the development of deep sea manned submersibles (Nash 1995).

The first modern deep diving by humans, to a depth of 923 m, was achieved in 1934 by William Beebe (1877–1962), an American zoologist, and Otis Burton (1899–1992) using the bathysphere, which means 'deep sphere.' The bathysphere was a small spherical shell made of cast iron, 135 cm of inside diameter designed for two observers. The bathysphere had an entrance hatch and a small glass view port. As the sphere was lowered by a cable and lacked thrusters, it was impossible to maneuver (Hotta et al. 2001). It was the beginning of the modern era of humans entering the deep abyss in a pressure resistant hull. Their discoveries were summarized in two books "Half Mile Down" (Beebe 1935) and "The World Beneath the Sea" (Barton 1953).

Manned observations using the Bathysphere in the deep sea rewarded the researchers with views of live marine life. However, it cannot dive to a very deep depth due to the self-weight of the cable. Auguste Piccard (1884–1962) was the first to design a submersible without the use of steel cables. It occurred after World War II, in 1947. The bathyscaph FNRS II was invented by Auguste Piccard, who had been studying cosmic rays using a manned balloon in Switzerland. The principle of the bathyscaph was the same as that of a balloon. Instead of hydrogen or helium gas, gasoline was used as the buoyant material. During descent, air ballast tanks were filled with sea water, and for ascent, iron shot ballast was released. The pressure hull was made of drop-forged iron hemispheres, 2 m in inside diameter and 90 mm in thickness, allowing for two crew members. It was able to maneuver around the seafloor by thrusters driven by electric motors.

In 1952, he began building the famous bathyscaph *Trieste*. In 1958, *Trieste* was purchased by the Office of Naval Research (ONR) of USA and plans for Project Nekton started. The main purpose of Project Nekton was to reach the bottom of the Challenger Deep. On January 23, 1960 the Bathyscaph *Trieste*, piloted by Jacques Piccard and Don Walsh, made one successful dive to the depth of 10,913 m of the Challenger Deep. However, in considerations of the length of day, sea state, and stirred cloud of sediment, *Trieste* stayed at the bottom for only 20 min. No photographs of the sea floor were taken (Kohnen 2009).

During the same period, the French navy also took great interest in building a deep manned submersible. The objective of *Archimede* was to reach 11,000 m. By 1959, construction and engineering work on *Archimede* were fully engaged. The scientific history of *Archimede* was summarized by Alan Jamieson who suggested that the reports from *Archimede* were far more valuable at the time than was realized (Jamieson 2015). However, in its whole life, it did not fulfill its original mission to reach the Challenger Deep.

The main goal in the 1950s for most deep dives was to set record depths, whereas in the 1960s, after reaching the Challenger Deep in the Mariana Trench, the development of manned submersibles focused more on the solution of certain practical and scientific problems. Both the *Trieste* and *Archimede* submersibles are regarded as the first generation of deep submersibles to distinguish the other operational deep manned submersibles developed later.

In 1964, the second generation of deep manned submersibles began. Alvin was funded by the US Navy under the guidance of the Woods Hole Oceanographic Institution (WHOI). At first, its depth capability was only 1800 m. It was small enough to be able to put on board the R/V Lulu, which became its support ship. Instead of gasoline flotation, syntactic foam was used. Alvin had horizontal and vertical thrusters to maneuver freely in three dimensions. Scientific instruments, including manipulators, cameras, sonar and a navigation system, were installed. Three observation windows were available for the three crew members. In France, the two-person 3000 m-class

submersible Cyana was built. These two vehicles typified submersibles during the 1960s. In 1973, the original steel personnel sphere is replaced with a titanium sphere and more flotation is added, the depth capability of the Alvin has been increased to 4500 m. In the 1980s, 6000 m-class submersibles, such as the Nautil from France, the Sea Cliff of the US Navy, the Mir I and Mir II from Russia, and the Shinkai 6500 from Japan, were built (Kohnen 2009). In 2012, China successfully developed the deepest operational submersible “Jiaolong” (Cui 2013). They were theoretically able to cover more than 98% of the world’s ocean floor.

Hotta et al. (2001) had predicted the possibility of the third generation of deep manned submersibles. They defined that manned submersibles of the third generation would be capable of exceeding 10,000 m depth. Three possibilities were speculated. One possibility is a small and highly maneuverable one- or two-person submersible with a transparent acrylic or ceramic pressure hull. Two projects were funded by private sectors in 2005 and two full ocean depth solo manned submersibles “*Deepsea Challenge*” and “*Deepflight Challenger*” were constructed in 2011. In March 26, 2012, the movie director James Cameron took his privately built one-man submersible “*Deepsea Challenge*” 10,898 m down to the Challenger Deep in the Pacific Ocean’s Mariana Trench (<http://www.deepseachallenge.com/>). However, another project seemed to be a failure. Now USA, Japan, and China are developing full ocean depth three-person operational manned submersibles (Cui et al. 2014; Cui 2018).

Another possibility is a deep submergence laboratory, which would be able to carry several scientists and crew long distances and long durations without the assistance of a mother ship. This would be the realization of the dream like “*Nautilus*” in *20,000 Leagues Under The Sea* by French novelist Jules Verne. Strong scientific and/or social goals would be needed for such a submersible design to be pursued. Ohno et al. (2004) had carried out a design feasibility study, but up to now, there is no actual investment for this concept.

And there is a third possibility that the next generation will be evolutionary upgrades of existing second-generation submersibles. Hawkes (2009) had proposed some ideas on how to dramatically reduce the cost of manned submersibles which is one of the main criticisms to the application of manned submersibles.

Principles of Modern Manned Submersibles

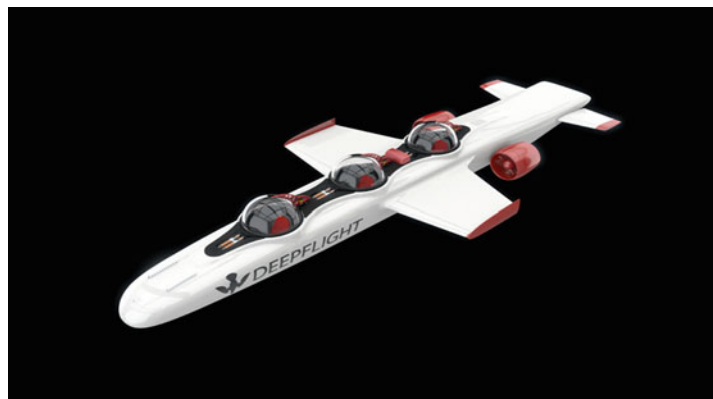
There are several methods to submerge vehicles into the deep sea. The simplest way is to suspend a sphere by a cable, known as a bathysphere. However, mobility is lost and diving depth is greatly limited due to the self-weight of the cable. A second method relies on powerful thrusters to adjust vertical position in relatively shallow water. The submersible *DeepFlight Super Falcon* is a high-speed design which uses thrust power coupled with fins for motion control like the wings of a jet fighter (Hawkes 2009), Fig. 1. It descends and ascends obliquely in the water column at speeds up to 10 knots. Most submersibles employ a third method that, while using weak thrusters to control attitude and horizontal movement, relies principally on an adjustable buoyancy system for descent and ascent, see Fig. 2.

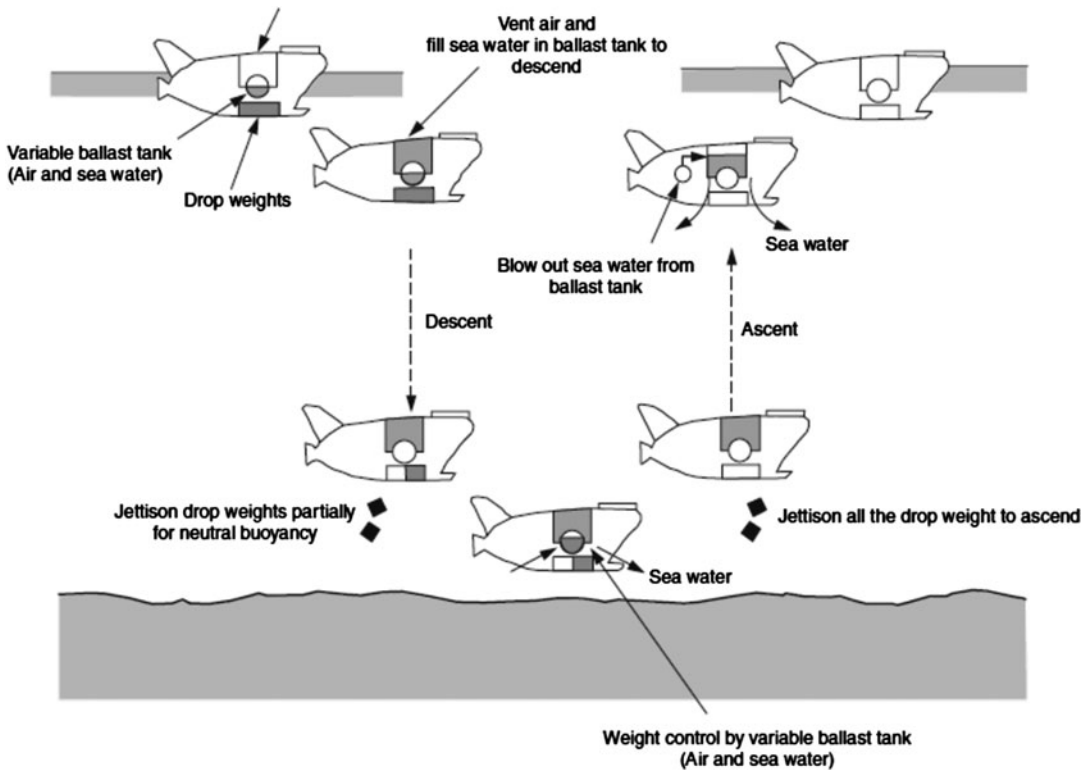
When on the surface, the submersible's air ballast tank is filled with air creating positive buoyancy; hence it floats. When the dive begins, air is vented from the ballast tank and filled with

sea water, thus creating negative buoyancy and sinking the vehicle. As the submersible dives deeper, buoyancy increases modestly due to the increasing water density created by the increasing pressure. Thus the submersible slows slightly as it dives deeper. When the submersible approaches the seafloor (50–100 m in altitude, i.e., height above the bottom), a portion of its ballast (usually lead or some other heavy material) is jettisoned to achieve neutral buoyancy. Perfect neutral buoyancy occurs when the positively buoyant materials (things which tend to float) on the submersible balance the negatively buoyant materials (things which tend to sink). This allows the vehicle to hover weightless in position and move freely about. As perfect neutral buoyancy is difficult to maintain, most submersibles have auxiliary weight-adjusting (trim and ballast) systems. This consists of a sea-water pumping system to draw in or expel water, thus adjusting the buoyancy of the submersible. Upon completing its mission, the remaining ballast is jettisoned and the submersible now with positive buoyancy begins ascending. When resurfaced, air from a high-pressure bottle is blown into the air ballast tank to give enough draft to the submersible for the recovery operation (Hotta et al. 2001).

In the development of the various submersibles, the most challenging equipment is the development of the FOD HOV since almost all of the technology challenges encountered in the development of unmanned submersibles are covered by the HOV. In the entry of submersible, the key technologies involved in the development of

Human Occupied Vehicle (HOV), Fig. 1 The future of manned submersibles could be the elegant *DeepFlight Super Falcon*. (Permission from Shanghai Rainbowfish Ocean Technology Co Ltd)





Human Occupied Vehicle (HOV), Fig. 2 Principle of descent and ascent for a modern deep submersible (Hotta et al. 2001)

the FOD HOV have been introduced and interested readers can read that entry.

Why Manned Submersible

There is no question that the manned submersible has earned its place in history. Much of what are now cataloged as new species were discovered in the last 30 years with the aid of submersibles. The ability to conduct marine science experiments in situ led to the development of intricate precision instruments, sampling devices for delicate invertebrates and gelatinous organisms that previously were only seen in blobs or pieces due to the primitive methods used to collect them (Askew 2001).

After the creation of unmanned submersibles in 1970s and 1980s, people have argued whether manned submersibles should be further developed (NRC 2004). Many people have thought that unmanned submersibles can replace all the functions of manned submersibles. However, after

about 30 years discussion, up to now, still many ocean scientists thought that manned submersibles have their own benefits.

First, the desire for personal accomplishment cannot be achieved in unmanned submersibles. For example, the great honor received by Jacques Piccard and Don Walsh in 1960 and James Cameron in 2012 when they dived to the Challenger Deep cannot be replaced. Second, beyond the desire for personal accomplishment, there are other reasons man should or could be included in an undersea work or exploration system. The poet, Dylan Thomas, has a line which reads "when all my five and country senses see." Man is a sensing creature possessing an integrated, coordinated, active intellect. When a man's trained intellect is part of a system, he is able to repair, reset, adjust, and adapt, and in short, to respond intelligently to the unusual situation. He can perform a variety of tasks because of his general orientation and versatility. The free-swimming diver comes closest to

Human Occupied Vehicle (HOV), Table 1 A comparison of efficiency of bottom time use. (Cited from a lecture PPT of Anatoly Sagalevitch at HAST, 2014)

Vehicle type	Searching of object after the bottoming	Sampling		High quality video	Measurements (physical, chemical)	Visual observation
		Simple (rock, single animal)	Complicated (fluid from smoker, slurp gun probe, etc.)			
HOV	<30 min	5–10 min	<30 min	100% continuous based on direct observation	100% trustworthy	100% direct
ROV	60–90 min	40–60 min	No experience: too complicated	20% of attempts during move of ROV	Open noisy data (thrusters)	30–40% on TV screen

exercising directly his senses in the ocean (primarily seeing, touching, and hearing). The man in the manned submersible, however, is sensing his environment remotely, except for one sense – that of sight. In the unmanned system all sense data is perceived remotely. Thus, this system is “remotely manned,” for man’s intellect and senses are still a part of the overall system, but they are applied remotely to the work site. Therefore, the primary reason for placing man at the scene is to make use of his active, interpretive ability to see (Talkington 1976).

There is no substitute for the autonomous, highly maneuverable submersible that can approach and collect without contact delicate zooplankton, while observing behavior and measuring the levels of bioluminescence, or probing brine pools and cold seep regions in the Gulf of Mexico for specialized collections of biological, geological, and geochemical samples. Tubeworms are routinely marked for growth rate studies and collected individually, along with other biological species that thrive in these chemosynthetic communities. Sediments and methane ice (gas hydrates) are also selectively retrieved for later analysis. Some new vehicles are still being produced, but have limited payloads, which restricts them to specific tasks such as underwater camera platforms or observation. Some are easily transportable but are small and restricted to one occupant; they can be carried by smaller support vessels and are more economical to operate. Man’s desire to explore the lakes,

oceans, and seas has not diminished. New technology will only enable, not reduce, the need for man’s presence in these hostile environments (Askew 2001).

When man is in the system, he must be protected from the hostile environment in a pressure hull. Since the pressure hull is usually made of steel, it becomes the largest, heaviest, and most costly part of a manned submersible. Anatoly Sagalevitch has made a comparison between HOV and ROV in Table 1 and he concluded that ROV is of only about 20% efficiency of bottom time use of the HOV. D.

Key Applications

Application Range of Each Type of Manned Submersibles

Manned submersibles have many uses worldwide, such as oceanography, underwater archaeology, ocean exploration, adventure, equipment testing, maintenance and recovery, and underwater videography. Manned submersibles basically have four types: ordinary manned submersible, DSRV, ADS, and rescue bell. Each type of submersibles have their own application ranges and they are introduced here.

The ordinary HOV can either be used for observations or taking samplings. Observations are made through three small viewports (one in front and one on each side), as well as with a number of video cameras coupled to multiple

in-hull monitors. In addition to a pair of manipulator arms, various scientific payloads may be attached to the front of HOV for collecting samples and performing experiments. Therefore, HOVs are powerful in carrying out all the deep sea observations and operations.

A DSRV is a type of HOV used for rescue of downed submarines and clandestine missions. In addition to the observational and operational capabilities of scientific HOVs, it also has the personnel transferability from a submarine to a DSRV. Rescue is usually accomplished by ferrying rescues from the stranded submarine to the pressure chamber of a DSRV.

Rescue bell is a simplified DSRV which does not have the maneuvering ability, but just ferrying rescues from the stranded submarine to the rescue bell.

An **atmospheric diving suit (ADS)** is a small one-person articulated submersible of anthropomorphic form which resembles a suit of armor, with elaborate pressure joints to allow articulation while maintaining an internal pressure of one atmosphere. The ADS can be used for deep dives of up to 700 m for many hours, and eliminates the majority of physiological dangers associated with deep diving; the occupant need not decompress, there is no need for special gas mixtures, and there is no danger of decompression sickness or nitrogen narcosis. Divers do not even need to be skilled swimmers (Thornton et al. 2012; Kesling 2011).

The ADS is the safest, most versatile deep diving system available for military and submarine rescue intervention use. It can be used for a wide variety of tasks in the full range of depth capability. These include but are not limited to (<http://www.oceanworks.com/our-business/military/military-ads/>): Search; Salvage; Aircraft crash site investigation and recovery; Test weapons Recovery; Submarine Emergency Ventilation and Decompression System (SEVDS) deployment; Emergency Life Support Stores pod posting; Acoustic and degaussing range service and inspection; Harbor and defense infrastructure inspection; DISSUB escape and rescue support; Equipment deployment and recovery; Mine countermeasures support.

Important Events of Manned Submersible Applications

Reaching the Challenger Deep

Manned submersibles reaching the Challenger Deep will always be an important event and up to now, only two manned submersibles have finished that feat. On January 23, 1960, the Bathyscaph *Trieste*, piloted by Jacques Piccard and Don Walsh, made one successful dive to the depth of 10,913 m of the Challenger Deep (Kohnen 2009). The movie director James Cameron funded a secret project to develop the full ocean depth solo manned submersible “*Deepsea Challenge*.” In March 26, 2012, he took his privately built one-man submersible 10,898 m down to the Challenger Deep in the Pacific Ocean’s Mariana Trench (<http://www.deepseachallenge.com/>).

Now, in United States, two full ocean depth manned submersibles (Deepsearch and Triton 36,000/3); in Japan, *Shinkai 12000*; and in China, two full ocean depth manned submersibles are under development (Cui 2018).

New Scientific Discovery

Between 1973 and 1974, project FAMOUS (French–American Mid-Ocean Undersea Study) was conducted in the Mid-Atlantic Ridge off the Azores using the French bathyscaph *Archimede* and the US submersible *Alvin*. The project was the first systematic and successful use of deep submersibles for science. They discovered and sampled fresh pillow lavas and lava flows at 3000 m



Human Occupied Vehicle (HOV), Fig. 3 Extraordinary, unexpected life forms discovered by *Alvin* in the Galápagos Rift in 1977 (www.whoi.edu)

Human Occupied Vehicle (HOV), Fig. 4

The B28RI nuclear bomb, recovered from 2,850 ft (870 m) of water, on the deck of the USS *Petrel*. (Picture from https://en.wikipedia.org/wiki/1966_Palomares_B-52_crash)



H

deep in the rift valley, where the oceanic crusts were being created, providing visual evidence of Plate Tectonics (WHOI 2018). In 1977, *Alvin* discovered a hydrothermal vent and vent animals in the East Pacific Rise off the Galapagos Islands at a depth of 2450 m. Discovery of these chemo-synthetic animals, which were not dependent on photosynthesis, had a profound impact on biology in the twentieth century, Fig. 3 (Corliss et al. 1979).

Salvage of H-Bomb

In 1966, the U.S. Navy called in *Alvin* to help find a hydrogen bomb accidentally dropped into the Mediterranean Sea. *Alvin* locates the bomb and, after the bomb skids down a slope on the first recovery attempt, it finds it again at a depth of 2,900 ft (880 m). Then, an unmanned torpedo recovery vehicle, CURV-I, became entangled in the weapon's parachute while attempting to attach a line to it. A decision was made to raise CURV and the weapon together to a depth of 100 ft (30 m), where divers attached cables to them. The bomb was brought to the surface by USS *Petrel*, Fig. 4.

Cross-References

- ▶ Atmospheric Diving Suit (ADS)
- ▶ Deep Submergence Rescue Vehicle (DSRV)

- ▶ Doppler Velocity Log for Navigation System in Underwater Vehicle
- ▶ Rescue Bell
- ▶ Submarine
- ▶ Submersible

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Human-Occupied Vehicle (HOV)

► AUV/ROV/HOV Propulsion System

Hybrid Remotely Operated Vehicle (HROV)

► Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV)

Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV)

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Synonyms

[Autonomous and remotely operated vehicle \(ARV\)](#); [Hybrid remotely operated vehicle \(HROV\)](#)

Definition

A **hybrid remotely operated vehicle (HROV)**, also known as **autonomous and remotely operated vehicle (ARV)** is a latest innovation of unmanned **submersibles** which can combine the advantages of ROV technology and AUV technology to explore the oceans freely in a relatively large range and photo the seafloor in site with reliable interventions. Compared with traditional ROVs, the vehicle is self-powered while its tether is fiber-optical and finer with less restriction. A typical HROV consists of three major components: underwater vehicle, tether system, and surface support. The fiber-optical tether system and hybrid control architecture are notable features of HROVs.

Scientific Fundamentals

Historical Development

The concept of a light-tethered vehicle for 11,000 m operations appears to have been first proposed by James MacFarlane of International Submarine Engineering Company (ISE) in 1990 (McFarlane 1990). A little later, Japan Agency for Marine-Earth Science and Technology (JAMSTEC) developed first HROV “UROV7K” and a traditional ROV “Kaiko” which dived to

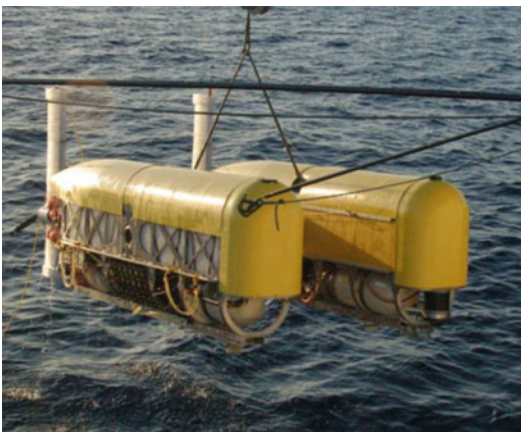
10,000.4 m depth in Mariana Trench in 1995 (Nakajoh et al. 1998).

While the Kaiko was lost for the accidental tether break between the vehicle and tether managing system (TMS) in 2003, new HROV “Nereus” was proposed to perform scientific research for full depth of the ocean (11,000 m) by Woods Hole Oceanographic Institution (WHOI) in USA (Webster and Bowen 2003). Early development is mainly focused on the fiber-optical tether and ceramic structure to insure the survivability in 11,000 m (Cooke 2005; Young et al. 2006). On May 31, 2009, the Nereus successfully completed its first dive to the hadal of Mariana Trench in 10,903 m at 11°22.1'N, 142°35.4'E (Bowen et al. 2010).

The Nereus employed twin free-flooded hulls to accommodate different modes (Fig. 1). In AUV mode, two independently articulated, actively controlled foils (wings) are located between the hulls at the aft and middle sections, respectively. AUV mode propulsion is provided by two 1 kW thrusters fixed on the aft tails and one 1 kW thruster on the articulated mid-foil. AUV mode has no lateral thruster actuation. The vehicle is hydrodynamically stable in pitch and heading when in forward flight. There is a downward-looking survey camera and several LED arrays

on the port hull. Additionally, a scanning sonar can collect high-resolution bathymetry of the seafloor. In ROV mode, the Nereus is configured with a work package containing a robot arm, sampling tools, sample containers, an additional high-resolution digital camera, two utility cameras, and several LED arrays. ROV mode propulsion is provided by two 1 kW thrusters fixed on the aft tails, as used in the AUV mode, with the addition of one lateral 1 kW thruster and one vertical 1 kW thruster (Bowen et al. 2009). In the past decade, many sea trials and applications were conducted to develop the Nereus. However, on May 10, 2014, the Nereus was unfortunately lost with 9,990 m depth in the Kermadec Trench near New Zealand. Then, its fragments appeared on the sea resulting from the ceramic spheres fracturing under high pressure of hadal ocean.

On the technical basis of HROV Nereus, WHOI put forward the solution to develop a new underwater vehicle, Nereid-UI, with the goal of deployment in polar ocean regions since 2011. The vehicle employs a lightweight fiber-optic tether enabling real-time control, high-resolution imaging, and immediate re-tasking over extended distances, thus freeing the vehicle from restrictions imposed by surface ice cover (Bowen et al. 2015).



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Fig. 1 Nereus HROV, shown during its first sea trial in the Pacific in November 2007, is designed to operate in

AUV/ROV modes with 11,000 m depth. Left: the Nereus in AUV mode. Right: the Nereus in ROV mode (Bowen et al. 2009)

In addition, some other research institutes are committed to developing HROVs and the prototypes have been made. In 2010, the French National Oceanographic Institute (IFREMER) launched a development program of HROV for identified needs to explore and perform scientific intervention tasks in marine environments such as the cold water coral reefs, underwater canyons, seamounts, ridges, rift valley walls, and continental margins. IFREMER's new HROV "Ariane" (Fig. 2) targets precise mapping, sampling, and monitoring tasks through the implementation of a hybrid architecture to bridge between canonical ROV and AUV systems. The Ariane has two operational modes, tethered and autonomous, to allow operations from a light vessel without dynamic positioning (DP) capability. In March 2015, the Ariane successfully reached the ocean depth of 2,011 m (Brignone et al. 2015).

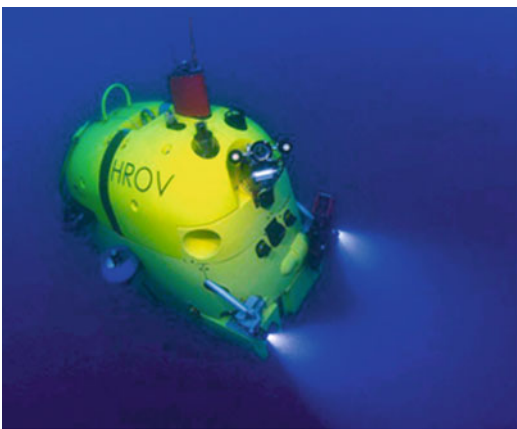
International appellation of HROV reflects innovative extension of traditional ROVs. Considering both AUV/ROV working modes, novel hybrid vehicles are also defined as ARVs. In recent years, China has become an important center for ARV research and development. Shenyang Institute of Automation (SIA) developed the first ARV in China to navigate under the Arctic ice in 2005 and participated in several Arctic scientific expeditions from 2008 to 2014

(Shuo et al. 2011). China Ship Scientific Research Center (CSSRC) developed a prototype of ARV "Seakite" in 2009 and upgraded the ARV for underwater applications in 2012 (Xu 2014). Under the funding of China's National Research and Development Projects, Shenyang Institute of Automation and Shanghai Jiaotong University launched their development of 11,000 m ARVs in 2016 individually, which will provide new available access to the hadal depth (Fig. 3).

Principles of Hybrid Remotely Operated Vehicle

As a self-powered vehicle, HROV can operate in two different modes. For range-free survey, the vehicle can operate untethered as autonomous underwater vehicle (AUV) capable of exploring and mapping the seafloor with sonars and cameras. For precise imaging and sampling, it can be converted at sea to become a tethered remotely operated vehicle (ROV). The ROV configuration incorporates a lightweight fiber-optic tether connected to the surface vessel for high bandwidth and real-time telemetry of video and data, enabling high-quality intervention and remote operation by a human pilot (Bowen et al. 2010).

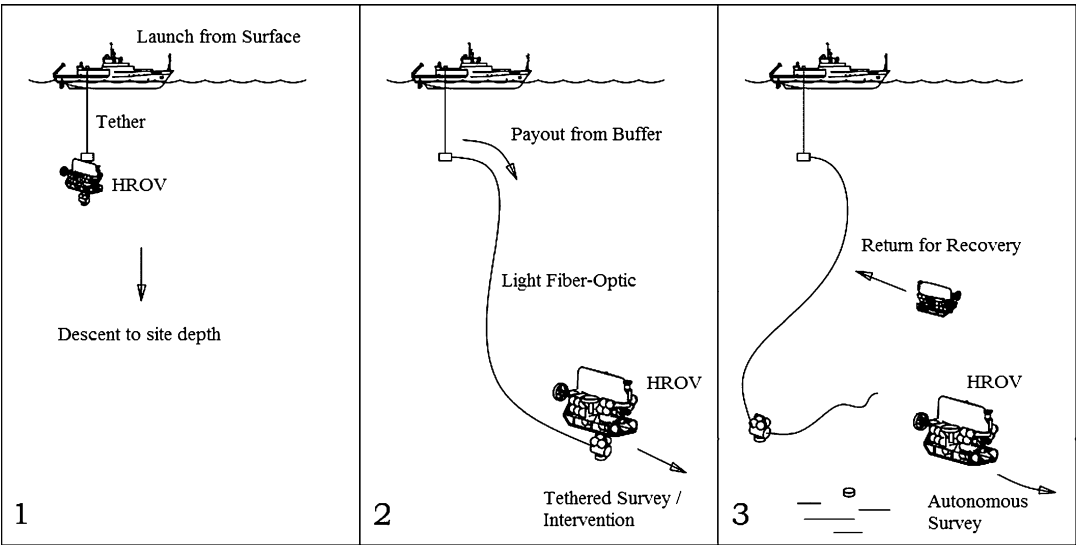
The HROV system is composed of underwater vehicle, surface support module, and tether managing system (TMS). As the first step in an operational deployment, the surface support will launch the HROV and TMS into the water while keeping connection with the vessel through the armored tether. Once the vehicle and TMS have reached the desired depth, the vehicle will be released from the TMS. Then, the vehicle begins a free fall to the seafloor with light fiber-optic payout from the buffer of TMS. Once the vehicle has reached the bottom, the descent weight will be released from the vehicle and the neutrally buoyant HROV will then be free to cruise and conduct its mission. Once the HROV has completed its task on the seafloor, the fiber-optic can be cut off. The vehicle drops its ascent weight and returns autonomously to the surface. Once the vehicle is on the surface, put forward a recovery process (Bowen et al. 2009) (Fig. 4).



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Fig. 2 IFREMER's new HROV Ariane during a technical trial (Brignone et al. 2015)



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV),
Fig. 3 Prototypical ARVs in China. Left: Arctic ARV of SIA (Shuo et al. 2011); Right: Seakite ARV of CSSRC (Xu 2014)



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV),
Fig. 4 HROV typical deployment process: 1-vehicle

dive; 2-tethered survey/intervention; 3-autonomous survey (Bowen et al. 2009)

Why Hybrid Remotely Operated Vehicle

As well known, currently almost all the underwater interventions, especially robotic manipulation, are carried out by ROV technology. ROVs require a lot of support, including a surface vessel, where an experienced pilot (i.e., user expert) is able to operate the ROV by means of an umbilical cable. On the other hand, from the last 20 years, marine explorations in a large region promote AUV technology. The AUVs have some obvious advantages

over ROVs: no vessel, umbilical, or pilot is essential. However, it also presents a critical drawback that any kind of potential manipulation skill is impossible. Thus, HROVs try to join the best of both systems, ROVs and AUVs to explore the oceans freely in relatively large range and photo the seafloor in site with real-time intervention capability. Compared with a traditional ROV, the HROV is self-powered and its fiber-optical tether is finer with weaker restriction. HROVs provide

a flexible choice for marine scientists with real-time intervention (Garcia et al. 2016).

Key Technologies of HROV

Underwater Vehicle

The underwater vehicle is composed of main hull (frame), pressure housing, electronics unit, power supply, propulsion, control boards, environmental sensors, sampling tools, additional buoyancy, and so on. The overall design of HROV configuration is usually considered from several aspects to accommodate the hybrid modes: (1) the descent/floating performance between the surface and seafloor; (2) the maneuverability and low hydrodynamic drag in both operational modes; (3) the functions of environmental sensors and sampling tools. The design of the HROV frame has the characteristics of light weight without reducing rigidity and strength. Therefore, the choice of frame material is very important. The frame structure usually employs a single hull or twin hulls for good stability (Table 1). In addition, the application of buoyancy materials would provide enough buoyancy and pressure-resistance to guarantee safety of HROV in deep sea (Xu 2014).

Power and Propulsion

Power supply is closely related to the endurance of the vehicle. High density and reliability are very important for the power supply in use. Up to now, lithium-ion battery is the most common solution for its advantages of high density, long cycles, no memory effect, environmental pollution-free, etc. The application of lithium-ion batteries will greatly meet the development of deep-sea HROV with large depth, long time, small volume, and no gas precipitation.

Generally, 4 degrees of freedom (4-DOF) motion capability is essential to HROVs for fix-

point operation, including surge, sway, heave, and yaw. According to the motion DOF, fixed combinations of horizontal and vertical thrusters are configured for HROVs to increase vehicle maneuvering. In order to promote the mobility, special HROVs also designed rotary thrusters. The coupling/decoupling hydrodynamics, control of DOFs and rotatory sealing under high pressure are critical problems.

Light Fiber-Optic Tether

The key part of HROV system is a light fiber optic tether used when operated in ROV mode. Although the advantages of HROV including large range and high maneuverability, a light-weight cable is essential for intervention. The fiber optic tether just transmits high bandwidth data, except power supply. The main consideration of a fiber optic tether is the optimization among the reusability, economical efficiency, and feasibility. With the HROV system, the fiber optic is deployed at low tension between the vehicle and launch platform. As the vehicle moves in water, the tether is laid from the storage buffer, which decouples the hydrodynamics between the vehicle and surface. This decreases the drag propulsion of the vehicle and workload for the tether; hence the tether's size and weight can be minimized. The Sanmina/SCI tether was selected as the primary choice for the Nereus HROV and its characteristics are described in Table 2 (Webster and Bowen 2003).

Tether Managing System (TMS)

Compared with traditional ROVs and AUVs, the fiber-optic tether and TMS are obvious features of HROVs in hardware. According to special applications, the design of tether manage system is different. The following aspects must be considered carefully.

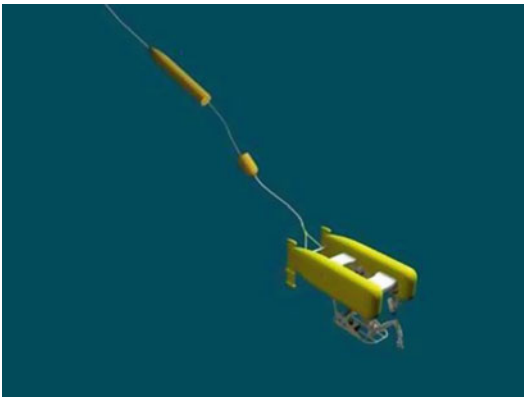
Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Table 1 Performance comparison of HROV configurations

Configuration	Descent/floating	Maneuverability	Sensors & tools
Single hull	★★★★	★★★★	★★
Twin hulls	★★★	★★★★	★★★★

Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Table 2 Sanmina/SCI fiber characteristics

Diameter (mm)	0.25
SG (fresh water)	1.36
Weight of 11 km in water (kg)	0.173
Working strength (N)	8
Breaking strength (N)	108

1. The influence of TMS on the moving flexibility of underwater vehicle should be considered, because it has a direct correlation with the payout of the optic-fiber. Active release by winch is feasible for the optic fiber in short distance. There are relatively few cables in disorder, and the storage of optical fiber is relatively easy. However, the finer release rate must match the speed of HROV. Otherwise, the transient strain between the vehicle and the tether will rip the fiber optic. Besides, the fiber optic should be uniform in diameter and have enough working strength. Another method of the payout of fiber optic is passively released by the drag force of vehicle. In this way, the fiber optic needs to overcome the adhesion stress (usually no more than 0.5 Kg) among the fibers in pack. Therefore, the passive release of fiber optic would not influence the motion of HROV.
2. Another consideration is surface influences of wave, current, and mother-ship movement. In order to minimize these influences, various buffer mechanisms are specially designed in the TMS. To the Nereus, the buffer device involved a snag-resistant depressor and a float pack to house the tether as shown in Fig. 5. The depressor was deployed far away from surface wave and current. The vehicle float pack contains the optical fiber dispenser, brake, fiber counter, and cutter. It floats above the vehicle, which is designed to minimize the drag and snagging possibility of the fiber on either the vehicle or sea floor. The depressor and float pack are mated together during launch process, protecting the fiber during the air-water transition. The depressor, float pack, and vehicle descend together to a designated depth. Once an



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Fig. 5 The float-pack of Nereus separating from the depressor (Bowen et al. 2009)

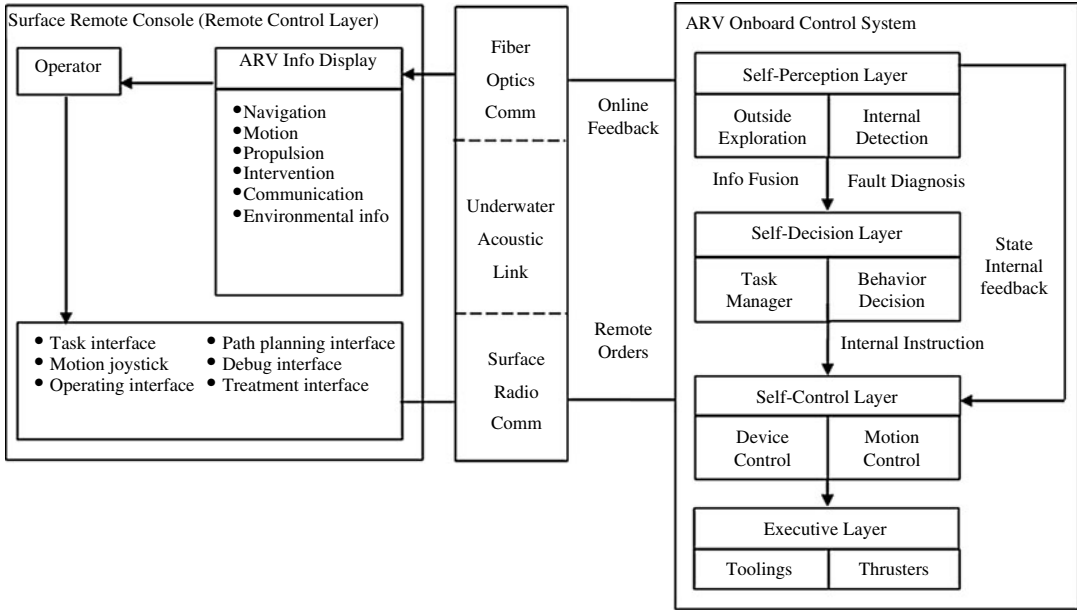
appropriate depth has been reached, the vehicle and float pack separate from the depressor. As the vehicle and float pack descend, optical fiber pays out from both terminals. In other HROVs, a relay station or some repeaters are also used as buffer devices.

HROV Control System

The control of ROV implies a master/slave architecture and a qualified pilot. In recent years, AUVs are becoming more popular without the limitation of tether. However, at the same time, AUVs need more autonomy and the user has little possibility of intervention in case of sudden problems. So, as the combination of ROV and AUV, HROV is presented as a balanced option. The hybrid control architecture is another obvious feature of HROVs in software.

HROV control system has both remote and autonomous modes. The HROV control system can be divided into interconnected layers as followed. Concrete block diagram is shown in Fig. 6. On the surface, there is equivalent of a top-decision layer by operator in loop, which can intervene and monitor the behaviors of HROV online.

- Remote-control layer
Remote-control layer is surface remote monitoring console. HROV send related information to surface through the optical fiber communication, underwater acoustic modem,



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Fig. 6 A kind of HROV control architecture (Xu 2014)

and surface radio as backup. The operators observe and analyze received data, and adjust its strategy online, take over motion control and other higher priority remote manipulation instruction, then send them to HROV through the communication link above.

- Self-perception layer

Self-perception layer mainly depends on various sensors integrated on HROV, such as navigation sensors, inspection sensors, and state monitor sensors. At the same time, perception layer also includes the understanding and processing of sensor data, and can transfer the understanding information to decision layer, then feedback the state information to self-control layer.

- Self-decision layer

Self-decision layer processes environment information output from perception layer, including global task management, extracts rational behaviors from behavior database to plan HROV navigation path, and then send instructions to self-control layer. To a certain context, self-decision layer is similar to preprogrammed AUV control, including simulation of mission programming, mission

implementation, mission download, and simulated running in deck preparation stage. It also takes real-time monitoring during underwater mission, automatic drive, obstacle avoidance, fault diagnosis and emergency treatment, data recording, and post-treatment in diving stage.

- Self-control layer

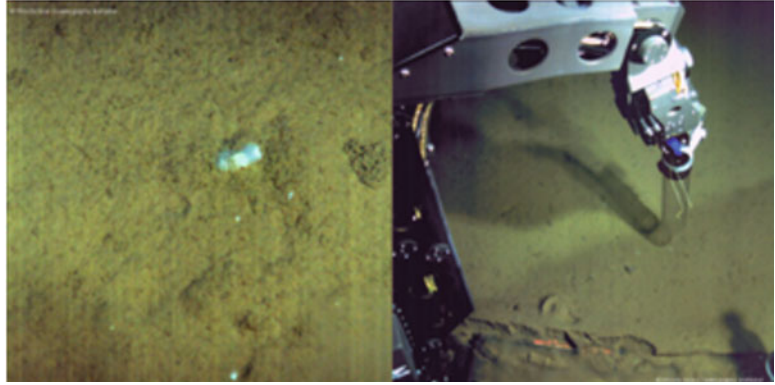
Self-control layer is related to internal state and environmental information measured by self-perception layer, which integrates control algorithms to get expected thrusts of the HROV, and transmits specific thrust instructions to executive layer, so as to achieve the planning objective. When surface remote layer is chosen to control HROV directly through the tether, it is working in ROV mode. When communication link is broken or removed, the vehicle will return to AUV mode automatically, and self-control layer will response independent instructions.

- Executive layer

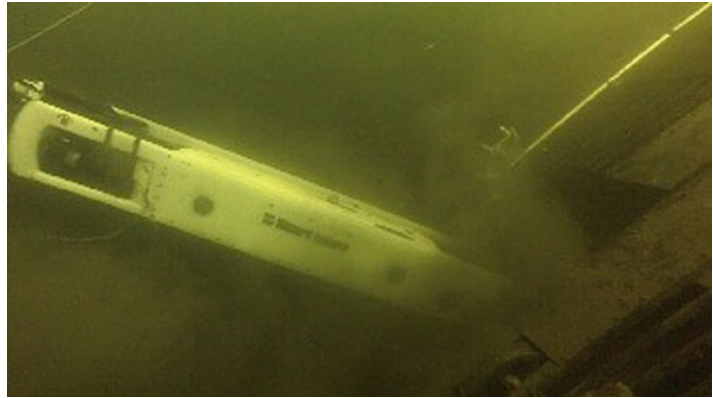
Executive layer will allocate thrust instructions of self-control layer to each actuator, and drive the actuators to generate expected forces/moments needed.

Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV),

Fig. 7 Tube cores and biological samples collected by Nereus (Bowen et al. 2009)



Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV), Fig. 8 The Sabertooth entering a flooded tunnel (www.saabseaeeye.com)



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Key Applications

Reaching the Challenger Deep

The goal of the Nereus is to provide the U.S. oceanographic community with the first capable and cost-effective vehicle for routine scientific survey, sea floor and water-column experimentation, and sampling to the full depth of the ocean in 11,000 m. On May 30, 2009, the Nereus successfully reached a depth of 10,903 m at 11 22.1'N, 142 34.4'E (Fletcher et al. 2009).

Scientific Research

Most of HROVs are used for scientific research. HROVs are superior to the traditional ROVs in economically efficiency, practicability and large range so as to be suitable in exploration and sampling of the seafloor. For example, the Nereus dived near the deepest known spot in the Challenger Deep, explored the edge of the sub

ducting plate, and took rock samples with its manipulator, which was fully operational. The vehicle then moved northwest across the trench floor toward the overriding plate, took tube cores and biological samples for approximately 2 km. Tube cores and biological samples were effectively collected (Fig. 7).

The goal to develop the Nereid-UI vehicle system is enabling real-time, detailed, close-range examination of biological and physical environments in glacial ice-tongues, ice-shelf margins. This approach will promote the existing capabilities for scientific operations under sea-ice, provided by conventionally tethered remotely operated vehicles, and autonomous underwater vehicles (Bowen et al. 2015).

Industry Applications

In addition to scientific research, HROVs can also be used in ocean engineering to solve complex

challenges of power station, water sewer, tunnel, and offshore industries.

Sabertooth HROV, developed by Saab Seaeye, broke a record for the longest tunnel inspection in 2014. During a round-trip of over 24 km, the Sabertooth collected real-time visual data while scanning the tunnel with its multi-beam sonar. The mission is to gather high-density dimensional data, discover open cracks or holes, find debris build-up, detect lining failures, and identify any rock falls. The Sabertooth can operate as either a remotely operated vehicle (ROV) with fiber-optic tether for real-time communication and pilot control, or as an autonomous underwater vehicle (AUV) to give flexibility in various tunnel inspection scenarios (www.saabseaeye.com) (Fig. 8).

Cross-References

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Remotely Operated Vehicle \(ROV\)](#)
- [Submersible](#)

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Hydraulic Pipe

- [Umbilical Cable](#)

Hydrodynamic Coefficient

- [Hydrodynamic Design](#)

Hydrodynamic Coefficients

- [Wind Tunnel Test](#)

Hydrodynamic Design

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Synonyms

Concept design; Hydrodynamic coefficient; Hydrodynamic performance

Definition

Hydrodynamic design of a submersible is the prediction of hydrodynamic performance and shape optimization through theoretical analyses, numerical calculations, or/and physical experiments. The hydrodynamic calculation runs through all design stages of a submersible, but it is of significant importance during the concept design. Hydrodynamic coefficients and propulsion efficiency are the most important factors affecting the hydrodynamic performance which is one of key technical indicators of submersibles.

Scientific Fundamentals

Hydrodynamic Performance

In general, hydrodynamic performance for a submersible includes the performance of speed, maneuverability and seakeeping. They are closely related to the main elements, profile, general layout and navigation control of the submersible, and have an important impact on the safety, reliability, economy, and applicability of the submersible (Allmendinger 1990).

The speed performance is considered good when under the premise of meeting the speed indicators in the design task book, the power consumption of the host is low, or when the host power is selected, the speed that can be achieved is high. The speed performance consists of two

aspects, the resistance and the propulsion. A submersible working under water bears the gravity force, buoyance from fluid static pressure, thrust from propellers, and hydrodynamic force. The resistance is defined as the projection of hydrodynamic force in the opposite direction of the submersible's motion. An excellent shape can significantly reduce the resistance and help to save the energy. The resistance can be categorized into friction resistance, shape resistance, and appendage resistance. The friction resistance is proportional to wet surface area and can be easily calculated by using various empirical formulas of plate friction resistance. The shape resistance, which is also called viscous pressure resistance, is related to the profile of submersible and involves flow separation. It lacks effective approximate evaluation method for a general shape. The appendage resistance comes from the resistance caused by appendages of the submersible. Determining the appendage resistance is usually difficult, because it not only requires accurate resistance calculation of an appendage by itself but also needs to take into account the interaction between the main body and the appendages, and the interaction between appendages themselves.

The propulsion performance relies on the principles of design and application of submersible thrusters. Firstly, submersibles carry out various tasks such as marine exploration, salvage, and life-saving. Therefore, requirements of good maneuverability must be met. Secondly, high propulsion efficiency, not the maximum speed, should be achieved. This is to maximize the endurance and prolong the underwater operation time. The submersible has poor visibility when working in the deep sea, which makes underwater search operations more difficult. Increasing the endurance capability accordingly will undoubtedly increase the success rate of underwater operations. The displacement of submersible is generally designed to be very small, and the battery used as a power source is also very limited. It is important to improve the efficiency of propulsion and make effective use of valuable energy. It should also be further pointed out that the submersible uses a considerable percentage of energy

in the process of diving and floating, so even a slight improvement in propulsion efficiency can achieve a large increase in search area. This effect becomes more obvious as the diving depth increases. Thirdly, the weight of the propulsion should be reduced as much as possible. As mentioned above, the design displacement of the submersible is generally very small. However, the large diving depth requires a sharp increase in the thickness and weight of the pressure shell. Complex deep-sea operations also require the submersible to be equipped with a variety of instruments, which further increases the weight of the submersible. This creates a sharp contradiction. Therefore, it is of great significance to reduce the weight of the propulsion device so as to satisfy strength and mission requirements. The mainstream submersible thrusters are mainly the propeller type and water jet propulsion, and the propeller type propulsion mode is used most extensively. Apart from these types, in recent years the bionic propulsion has received increasing attention because of its high efficiency and low noise.

Maneuverability refers to the performance of a submersible to change or maintain its speed, attitude and direction through its control device. The meaning of maneuverability can be further divided into course retention under small rudder angle, course mobility under middle rudder angle, and emergency avoidance under large rudder angle. The main factors affecting maneuverability include principal scale, profile, position of center of gravity and floating center, and appendages. The analysis of maneuverability is based on the equation of motion, under premise that the hydrodynamic force acting on the submersible is known.

Last but not least, the seakeeping performance means that the structure still has sufficient stability and hull structural strength when it undergoes rocking motions, slamming, stall, or other phenomena due to external forces such as wind and waves, and can maintain a certain speed for safe navigation and operation. The seakeeping performance needs to be considered when the submersible moves or/and works near the free surface like a ship advancing or an offshore structure floating

on the free surface where the influence of waves cannot be ignored.

The hydrodynamic performance is only one of the many factors to be considered in submersible design. Other aspects such as structural analysis and man-machine-environment system also play important roles in the design. The method based on multidisciplinary design optimization (MDO) has become the most potential design method for large-scale complex engineering systems including submersibles. This method puts forward higher requirements for the accuracy and efficiency of its hydrodynamic performance prediction. Therefore, designers must pay attention to the research on the prediction method of hydrodynamic performance of submersibles.

Approaches for Hydrodynamic Forces/Coefficients

The determination of hydrodynamic forces/coefficients is important for predicting loads and induced motion analysis of submersible which will affect the design of submersible. Hydrodynamic forces acting on submersible can be divided into inertial hydrodynamic forces and viscous hydrodynamic forces. When the submersible is in steady motion, only viscous hydrodynamic forces need to be considered. When the submersible is in unsteady motion such as acceleration and deceleration, the influence of inertial forces or added mass should be taken into consideration.

Inertial hydrodynamic forces or added mass are usually estimated by empirical formula or evaluated in the framework of the potential flow theory which is based on the assumptions that the fluid is nonviscous, incompressible, and irrotational (Newman 2018). Following these, the governing equation is the Laplace equation instead of the Navier-Stokes equation. Added mass can also be determined by a planar mechanical machine or vibration tests. According to the potential flow theory, the added mass only depends on the shape of the submersible, having nothing to do with motion states of the structure.

Compared with inertial hydrodynamic forces, viscous hydrodynamic forces are more difficult to calculate. At present, there are four approaches to obtain viscous hydrodynamic forces, which are

constrained or self-propelled model test, system identification, approximate calculation method, and computational fluid dynamics (CFD) method. The model test can be carried out in a tank or wind tunnel and is most reliable. However, it is expensive, takes a long time, and has the problem of scale effect, and therefore not conducive to the development of low-cost submersibles. The method of system identification is suitable for the analysis of actual navigation data and the correction of hydrodynamic coefficients, which is not convenient to guide the maneuverability design of submersibles.

Approximate calculation method mainly refers to making some modifications based on the theoretical calculation formula for simple geometries. It can also refer to the semiempirical formula and charts which are sorted out from a large amount of model tests, rather than analytical solutions of the hydrodynamic model or equation systems. In the concept design stage, this method is effective because of the low requirement of calculation accuracy and is especially suitable for submersibles with a regular shape such as ellipsoids. A basic evaluation of hydrodynamic performance of the design model can be obtained, and it can provide the basis for the design of propulsion system and battery system if there are any. From the relevant research data and experience, although approximate calculation method has advantages of convenient use and fast evaluation, the scope of application is limited. It will produce large errors if it is not used properly.

With the continuous improvement of computer hardware and the rapid development of advanced numerical calculation technology, CFD method (Gao et al. 2016) can be directly applied to the design process of submersibles. Both commercial CFD software such as STARCCM+ and open source packages such as OpenFOAM are used to solve problems related not only to fluid dynamics but also fluid-structure interactions by combining with the finite element method. In the preliminary and detailed design stages, CFD method has been widely applied to evaluate resistance and viscous hydrodynamic coefficients. The accuracy depends on the selection of turbulence model, the mesh generation, and the setting of boundary

conditions. Because of difficulties in meshing and the limitation of computing time, it is still difficult to apply CFD method to shape optimization design.

Though CFD method has been widely used in the design process, there are still many technical problems to be solved. Different turbulence models or grid types often lead to different numerical results. In the absence of experimental data, it is difficult to judge the accuracy of the calculation results. Since the potential flow is solved by Laplace equation, and the Navier Stokes equation is solved by CFD numerically, a hybrid or multi-domain method has been developed for bodies floating or advancing on the free surface. In this method, a control surface is introduced to divide the whole fluid domain into subdomains (Chen and Liang 2016) and apply only potential flow theory solver in the external subdomain in which viscous effects can be ignored. This method may be easily extended to the study of submersibles since there is no need to consider wavy properties of the free surface. A reasonable way is to use these methods comprehensively according to their characteristics, either a coupling of physical experiments and numerical simulations or a coupling of different numerical methods and so on.

Key Applications

It is worth noting that submersibles with different functions have different requirements for hydrodynamic performance. Therefore, hydrodynamic problems to be focused depend on submersible types and missions. The following will describe the hydrodynamic problems in the design of autonomous underwater vehicle (AUV), human occupied vehicle (HOV), remotely operated vehicle (ROV), and autonomous remotely vehicle (ARV). The biologically inspired systems are gradually applied to design of submersibles (Kruusmaa 2018), though corresponding hydrodynamic problems are not illustrated here. The bioinspired submersible may become the submersible of the next generation, with radical designs not only on shape but also on propulsion system, control system, or other systems.

Hydrodynamic Problems in AUV

During the design process of AUV, designers need to study carefully and design its shape and hydrodynamic layout according to its hydrodynamic characteristics. The AUV hull provides space and watertightness for payloads and internal equipment, withstands pressure, and generates buoyancy. Meanwhile, external accessories such as tail fins and propeller are mounted on the hull. Hydrodynamic layout is also called maneuverability design which, through selection and layout design of fins, satisfies requirements of motion characteristics such as balance, motion stability, and maneuverability.

There are numerous AUV profiles, with the common ones shown in Fig. 1. Most AUVs adopt single axisymmetric rotating body shapes, especially the torpedo profile (Liu et al. 2002). From appearance they generally tend to be slender and have larger slenderness ratio and less resistance. The motion control is carried out by the propeller and fins arranged in stern. This kind of AUV has long voyage capability and ability to resist moderate currents. They are applicable to low precision detection tasks within larger regions. By contrast, non-axisymmetric shapes (Ura 2002; Kaiser et al. 2016) and multi-body AUV have better maneuverability and are suitable for a wider range of tasks though they have larger resistance.

An important hydrodynamic problem in AUV design is to investigate the influence of shape on its hydrodynamic performance. Experimental results of a symmetric series of 24 mathematically related streamlined bodies of revolution have even been reported (Gertler 1950), showing how the resistance of these bodies at deep submergence varies with changes in five selected geometrical parameters which are fineness ratio, prismatic coefficient, the nose radius, the tail radius, and the position of the maximum section. Many researchers have studied axisymmetric rotating bodies on above five parameters in order to further explore low drag shapes (Ericksen et al. 2001). Research on hydrodynamic performance and profiles of existing underwater vehicles mainly focus on axisymmetric rotating body shapes, while few open data are available on

non-axisymmetric shapes and multi-body AUV. These calculations or experimental data about axisymmetric rotating bodies mostly belong to submarine, torpedo, and other high-speed navigation bodies. The speed of AUV is lower and corresponding Reynolds number is smaller. Therefore, designers should be careful to use relevant data and conclusions.

With the growing maturity of AUV technology, its application field has been greatly expanded. The task-oriented design becomes the trend in order to adapt to wider demand for use. Submersible shapes must first meet specific tasks usage requirement. Non-axisymmetric or multi-body shapes will occupy an increasing proportion in future AUV designs so it is necessary to pay attention to the hydrodynamic research on these profiles.

Hydrodynamic Problems in HOV

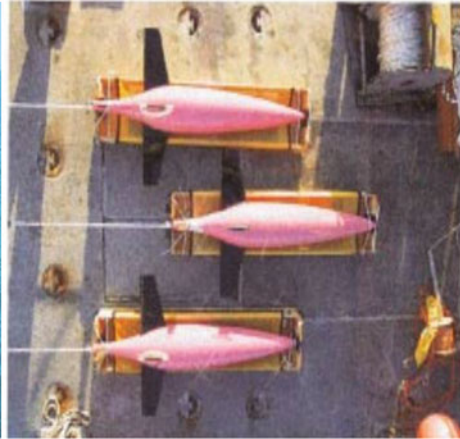
The design of HOV is different from unmanned underwater vehicle (UUV). Its main dimensions and profile elements depend mainly on the sufficient layout space for person and equipment to be carried inside the hull. In the design of HOV, designers firstly start from layout requirement and consider other factors such as carrying capacity of the surface mothership, deployment, and recovery method, so that designers can calculate the main dimensions. Then designers make decisions on reasonable main elements according to the balance of gravity and buoyancy, requirements of resistance, propulsion, stability, and maneuverability.

Therefore, the shape of HOV has the following characteristics:

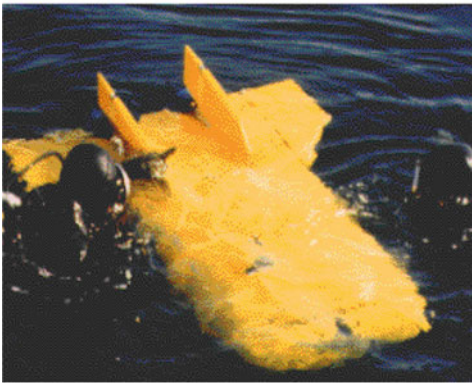
1. The body ship is short and thick, and the length width ratio L/B is small. The value of L/B at large depth is generally between 2 and 3, and the maximum value is not greater than 3.6 which is far lower than that of conventional submarines. The resistance performance of HOV is not good and speed is low.
2. The bow shape of HOV is complex and there are many appendages. In order to meet requirements of observation and operation, it is necessary to arrange sampling basket, operation



CR-02 (torpedo shape)



Seaglider (Low-drag shape)



PTEROA (flat shape)



Sentry (vertical flat shape)

SeaBED (two-body shape, www.whoi.edu)ABE (multi-body shape, www.whoi.edu)

manipulator, camera, and other equipment in the bow of HOV. Appropriate space should be reserved for the observation window of HOV.

3. The bottom of HOV is usually flat. This is not only to meet requirements of sitting on the bottom but also convenient to arrange larger equipment such as battery and ballast.

In the design of HOV profiles, designers can only optimize the local shapes or appendage shape to reduce the resistance and meet the hydrodynamic performance. In order to carry out this work, it is necessary to analyze details of the flow field around the HOV to find the location of flow separation. Then, the local shape or appendage shape which leads to flow separation is modified. Finally, the modified scheme is verified to confirm whether it can improve hydrodynamic performances.

Due to the limitation of weight and volume, the energy carried by HOV at large depth is limited, so it cannot be used in the movement of diving from the sea surface to several thousand meter below or returning to the surface from several thousand meters below. The movement that does not use energy is called unpowered diving and floating. Because the total residence time of HOV is limited, the speed of diving and floating will directly affect the working time of HOV on the seabed. In the process of unpowered diving and floating motion design, it is necessary to ensure both the stability and the speed of unpowered diving and floating (Ma and Cui 2004). The stability of HOV during the unpowered diving and floating is generally guaranteed by stabilizing fins. When designing the stabilizing fins, the stability of the horizontal plane can be improved on the premise of ensuring that of the vertical plane.

In order to ensure the speed of unpowered diving and floating, it is necessary to design a suitable weight of disposable ballast to make the HOV produces negative or positive buoyancy. If the diving speed is too fast, it will easily lead to accidents such as touching the bottom, and if the speed is too slow, the underwater operation time will be affected. Therefore, it is necessary to accurately calculate the weight of the ballast that can

be dropped. It is of importance to establish the hydrodynamic model of HOV to predict its trajectory and attitude under certain conditions.

Hydrodynamic Problems in ROV/ARV

ROV is a kind of underwater vehicle with wired remote control, which is the most widely used type of submersible. ARV synthesizes the advantage of AUV and ROV, and it has its own energy and connects with the mothership through optical fiber cable. It can realize medium range search and fixed-point observation in autonomous, remote control, and monitoring mode in deep-sea environment.

Compared with other types of submersibles, ROV and ARV have cables connected to the surface mothership when they work, so they are not only affected by ocean currents but also disturbed by underwater cables. The surface mothership and the part near the free surface of the cable are affected by wind, waves, and currents; the underwater part of the cable, the submersible, or the relay station are also inevitably affected by the ocean current and internal waves; there are very complex interactions among the surface mothership, the cable, the submersible, or/and the relay station, which constitute a complex dynamic response system. Therefore, in the design process of cable submersible, it is necessary to carry out numerical simulations on the motion of submersibles to understand its dynamic response characteristics. The reliability and safety of equipment can be guaranteed by effective control of cable and submersible.

In the research of ROV and ARV motion simulations, underwater cable modeling is a key work. Because the motion of underwater cable is generally nonlinear and the hydrodynamic force acting on it is time-varying, the dynamic problem of underwater cable can only be solved by numerical approaches. The lumped mass method and finite difference method are the most widely used (Feng and Allen 2004). After the hydrodynamic modeling of cable and submersible is completed, the dynamic motion analysis of the whole system can be carried out, and the maneuverability and control performance of the submersible can be studied.

Cross-References

- [Concept Design](#)
- [Design of Submersibles](#)
- [Detailed Design](#)
- [Multidisciplinary Design Optimization \(MDO\)](#)
- [Preliminary Design](#)

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Hydrodynamics for Subsea Systems

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Synonyms

[Autonomous underwater vehicles \(AUVs\)](#); [Blow-out Preventer \(BOP\)](#); [Computational fluid dynamics \(CFD\)](#); [Keulegan-Carpenter \(KC\) number](#); [Remotely operated vehicles \(ROVs\)](#); [Reynolds number \(Re\)](#); [Vortex-induced vibration \(VIV\)](#)

Definition

Hydrodynamics is the study of liquids in motion and their interaction with a solid object [https://en.wikipedia.org/wiki/Fluid_mechanics]. This subject compares to hydrostatics, which is about fluid at rest. Along with aerodynamics, hydrodynamics is one of several subdisciplines of fluid dynamics.

Subsea systems are submerged-undersea structures, typically without direct impact from the free surface of the ocean (Bai and Bai 2019) except during installation. The subsea systems are enormously diverse. For instance, in offshore oil and gas engineering, subsea systems include the following:

- Structures associated with drilling systems, including wellheads, BOPs, and drilling risers.
- Manifolds and other support structures for containing and protecting valves, tees, flanges, or mechanical connectors. These structures typically have a foundation to enable the seabed soil to support the system, a functional element containing pipes valves or other hydrocarbon-containing components, and a protective frame that prevents damage to the structure from fishing gear or dropped objects.

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- Pipelines, which include flexible flow lines, rigid steel pipelines coated in concrete or polymeric coatings, umbilicals, and cables (Griffiths et al. 2018).

Here, the subsea systems contrast to the topsides above the waterline. More widely, the subsea systems are used to describe, for instance, gliders in oceanography, submarines in maritime engineering, or remotely operated vehicle (ROVs) and communication cables in other domains of ocean engineering [[https://en.wikipedia.org/wiki/TopsidesandSubsea_\(technology\)](https://en.wikipedia.org/wiki/TopsidesandSubsea_(technology))].

Hydrodynamics engineering in subsea systems, underlining practical application, has evolved to focus on understanding water motion to support subsea structural engineering by defining the loads experienced by subsea systems.

Introduction

Hydrodynamic engineers calculate forces and moments induced by water in motion for a safe and efficient subsea system, just like engineers consider wind loads on bridges or aircraft. Challengingly, water is over 830 times denser than air. The loads on the same object generated by moving water can be more significant than moving air at a similar speed. Conversely, objects moving through water or water moving past objects tend to occur at greatly reduced velocities than those dealt with in aeronautics. The comparative nondimensional behavior of these fluids can be understood by considering the Reynolds number of the flow, which defines the ratio of inertial to viscous forces. Reynolds number is defined as $Re = VD/\nu$ where V is the fluid velocity (or object in a still fluid), D is the characteristic length of the object, and ν is the kinematic viscosity. The typical ocean flows around subsea systems produce Reynolds number values in the range 10^4 to 10^6 , which means that the flow and wake produced are often turbulent.

Ocean water also frequently moves back and forth due to wave and tides, and this oscillatory motion may create an additional challenge to the subsea system. These oscillatory flows can be understood using the nondimensional Keulegan-

Carpenter parameter defined as $KC = U_{\max} \cdot T/D$ where in this context $U_{\max}$ is defined as the peak oscillatory velocity and T the period of the oscillation according to $U(t) = U_{\max} \cdot \sin(2\pi t/T)$.

Many subsea systems reside on the seabed once installed, such as oil and gas pipelines and communications/power cables. In contrast to the intention, they may move subject to seabed scour and erosion. The local scour around a subsea system is another area that hydrodynamics has contributed to significantly.

Over the years, the study of subsea system hydrodynamics has received contributions from many mathematician and engineers not necessarily working on subsea systems. Sarpkaya et al. (1982) and Barltrop and Adams (2013) provided excellent documentation of the principals and design methods. Theoretical analysis, field measurements, laboratory experiment, and computational simulation have become three main approaches in hydrodynamics study in recent years. In particular, computational fluid dynamics (CFD) has been widely used in hydrodynamics for subsea systems. In the era of aviation and space exploration (aerodynamics engineering), hydrodynamics has benefited from theoretical and technological advancement in that field, in parallel with the ongoing improvements in computational capabilities following Moore's law [<https://www.investopedia.com/terms/m/moorelaw.asp>].

Dynamic Interaction

The water-induced forces (F) and moments (M) on a subsea system are dependent on the velocity (u) and pressure (p) field of the fluid near the fluid-structure interface. The force can be expressed as

$$\mathbf{F} = \oint_A p d\mathbf{A} + \mu \oint_A \frac{d\mathbf{u}}{dn} dA \quad (1)$$

where μ is the fluid viscosity, du/dn represents the normal velocity gradient on the structure surface, and A is the fluid and structure interface area. Here, the first term is the pressure force, whereas the second term is the friction force. Despite its

simple form, the difficulty in calculating the force arises when the pressure and velocity are not spatiotemporal invariant. Due to the viscosity, water and the subsea system's interaction are a dynamic one in many practical applications. Water viscosity creates a boundary layer adjacent to a solid surface in relative motion. The boundary layer flow is the source of vortex generation and shedding from the solid surface. This vortex shedding process leads to a dynamic change in velocity and pressure adjacent to the solid surface and, therefore, a dynamic force on the solid surface. The following Fig. 1 shows such a dynamic process, where many counterrotating vortices are generated in the wake of an oscillatory cylinder immersed in a steady current.

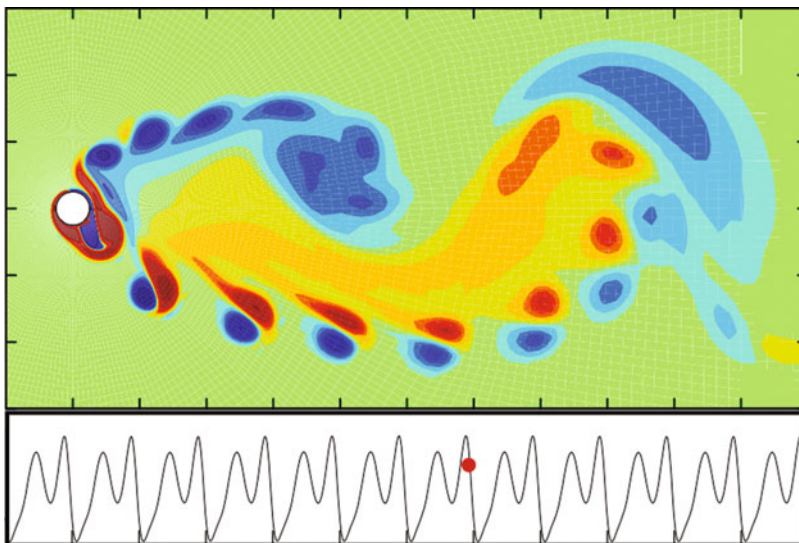
Study Approaches

Since Daniel Bernoulli (1700–1782), mathematical analysis and laboratory experiment have been two major approaches in

hydrodynamics study (Darrigol 2005). Many theories are the foundation of fluid dynamics, including Bernoulli's principle and the Navier-Stokes equations. In particular, the Navier-Stokes equations' derivation is recognized by many, if not all, as the pinnacle of fluid dynamics. The Navier-Stokes equations mathematically describe Newtonian fluids' motion through a series of partial differential equations expressing the conservation of momentum and conservation of mass [https://en.wikipedia.org/wiki/Navier-Stokes_equations].

A very common hydrodynamics model with a great deal of utility at adequate but imperfect accuracy is the set of equations known ubiquitously as Morison's equations, albeit first published in a paper also coauthored by O'Brien, Johnson, and Schaaf (Morison et al. 1950). These equations give the hydrodynamic forces on elongate structures such as piles, oil and gas platform jacket legs and braces, and subsea pipelines as follows:

H



Hydrodynamics for Subsea Systems, Fig. 1 The water flow field adjacent to a solid bluff body can be a dynamic one, which induces a time-dependent force on the cylinder. The top shows the flow field from a computer simulation, with the undisturbed incoming flow in the horizontal direction from left to right while the cylinder oscillates in the

vertical direction. The time-varying signal at the bottom is the drag force on the cylinder (in the same direction of the flow), with the red dot indicating the force induced by the instantaneous flow field on the top (from unpublished work)

$$F_Y(t) = 0.5 \rho \cdot D \cdot C_Y \cdot |U_c + U_w(t)| \cdot (U_c + U_w(t)) \\ + \frac{\pi \cdot D^2}{4} \rho \cdot C_M \cdot a_w(t)$$

$$F_Z(t) = 0.5 \rho \cdot D \cdot C_z \cdot (U_c + U_w(t))^2$$

where $F_Y(t)$ are the time-varying forces orthogonal to the structure axis and in line with the flow, $F_Z(t)$ are the forces orthogonal to the flow and the structure axis (representing the lift force for subsea pipelines), ρ is the liquid density, C_Y is the “drag” coefficient, U_c is the steady component of the liquid flow velocity, $U_w(t)$ is the time-varying wave velocity, C_M is the “inertia” coefficient, $a_w(t)$ is the acceleration of the fluid, and C_z is the “lift” coefficient. Note that this formulation ignores any time-varying forces on the structure due to vortex shedding.

Today, computational fluid dynamics (CFD) has become a valuable tool in almost all fields of fluid dynamics, including hydrodynamics for subsea systems (Tong et al. 2017). One simple reason is mathematicians have not yet found a universal solution to the Navier-Stokes equations. Instead, to obtain the fluid velocity and pressure adjacent to a solid object through Navier-Stokes equations, the only way is to apply a discretization method so that they are solvable by a computer algorithm, i.e., computational fluid dynamics. This tool is potent as it can reveal the flow field at any point of interest in one simulation. The method is highly repeatable, and the construction of a CFD simulation usually is significantly cheaper than the construction of laboratory tests. However, caution should be given to validate the CFD simulations (Fig. 2).

Applications

Many challenges in subsea engineering have a hydrodynamic element. Here, we give some examples.

Vortex-Induced Vibrations

Recall that vortex shedding from a subsea structure may create a time-varying periodic force

(Zhao et al. 2012). When the frequency of vortex shedding matches or is close to that structure’s natural frequency, the so-called vortex-induced vibration (VIV) may occur – a resonant response of the structure to the period force induced by the fluid (Bearman 1984). VIV is a significant part of the design of subsea systems for two reasons: (1) Excessive stress as a result of vibration may directly damage a subsea system, and (2) even without harmful stress, the periodic motion over millions of cycles can cause fatigue to a subsea system.

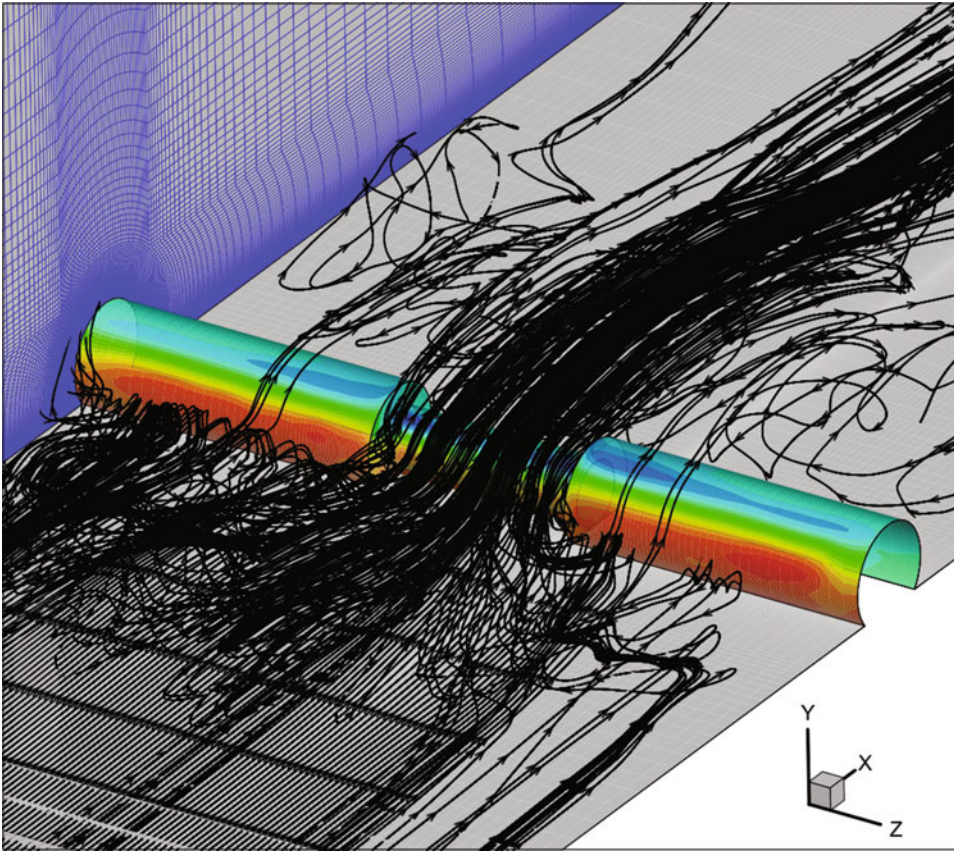
To put this into perspective, assuming a cylindrical steel structure has a diameter D of 1 m subject to 1 m/s ocean current (u) when vortex shedding frequency (f) matches its natural frequency (f_n). If this ocean condition happens 10 days per annum, how many cycles of vortex-induced vibrations the subsea system in a design life of 30 years might experience? First, let’s assume the Strouhal number (S_r) is 0.2 for this subsea system at the metocean condition, so

$$S_r = \frac{fD}{u}. \quad (2)$$

Thus $f = 0.2$ Hz by submitting the velocity and diameter into the equation. The number (N) of vortex-induced vibrations of the structure under the single ocean condition is $N = 30 \text{ years} \times 10 \text{ days/year} \times 86,400 \text{ s/day} \times f = 5.18 \text{ million}$. Here the stress due to a single cycle of vibration may be inconsequential; however, bending a steel pipe back and forth for 5.18 million times certainly is not an impact that can typically be neglected. This is one of the reasons why VIV is a significant consideration in the design of subsea systems.

Flow-Induced Scour

Soil and sediment’s movement in the vicinity of a subsea structure – flow-induced scour – is another traditional area of interest for hydrodynamics (Soulsby 1997; Sumer and Fredsøe 2002). The issue arises that a bed-seated subsea system tends to redistribute fluid velocity around it (Zhao et al. 2010), which would locally increase



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Hydrodynamics for Subsea Systems, Fig. 2 The computational fluid dynamics (CFD) simulation of the flow around a pipeline partially embedded in a seabed. The pipeline is nonuniform, with a smaller diameter in the middle part. The pipeline is colored by the fluid's pressure distribution, where red indicates high pressure and blue

indicates low pressure. The velocity field is shown by streamlines generated uniformly at specific points upstream but has concentrated to the central part on arrival. The mesh used in discretization for the Navier-Stokes equations is shown in blue at the far top-left corner on a plane perpendicularly cutting through the pipeline

the friction force (bed shear stress) on the seabed. Then, depending on the local soil and sediment characteristics, scour may happen around the structure. This process would typically result in a local scour pit due to the loss of soil and sediment, leading to the structure's stability issue. Such a feature is better known in rivers (Soulsby 1997), such as around a bridge pier, where a scour pit could be seen at the foot of a pier after flood water recedes. In the marine environment, tidal current- and wave-induced scour are common in nearshore areas from shallow to deep waters, and hydrodynamics has been a valuable tool in studying and preventing it.

Oscillatory Flow

It is not hard to imagine why water can move in an oscillatory trajectory. Still, this common flow in the ocean behaves distinctively different with that of a unidirectional flow around a subsea structure (Tong et al. 2015b). A periodic varying velocity characterizes oscillatory flow. The oscillatory-flow hydrodynamics depend on both the Reynolds number (Re , defined in a similar way as in unidirectional flow) and an additional parameter – Keulegan-Carpenter (KC) number. The KC number is related to the oscillatory flow period and represents the ratio of orbital distance compared to a subsea structure's characteristic length. For a

classic cylindrical bluff body, the vortex-shedding regimes induced by a sinusoidal oscillatory flow show a great diversity compared to the field generated by the steady incoming flow. Due to the acceleration of the water particles, oscillatory flow introduces a new force term on a subsea structure, which is inertia force. Besides, the oscillatory flow may induce vibrations to subsea systems by itself. When KC is large – the wave in a period travels a much longer distance than the size of a structure – the impact of oscillatory flow, however, is similar to that of the steady directional flow.

Modeling of Hydrodynamic Forces on Complex Structures

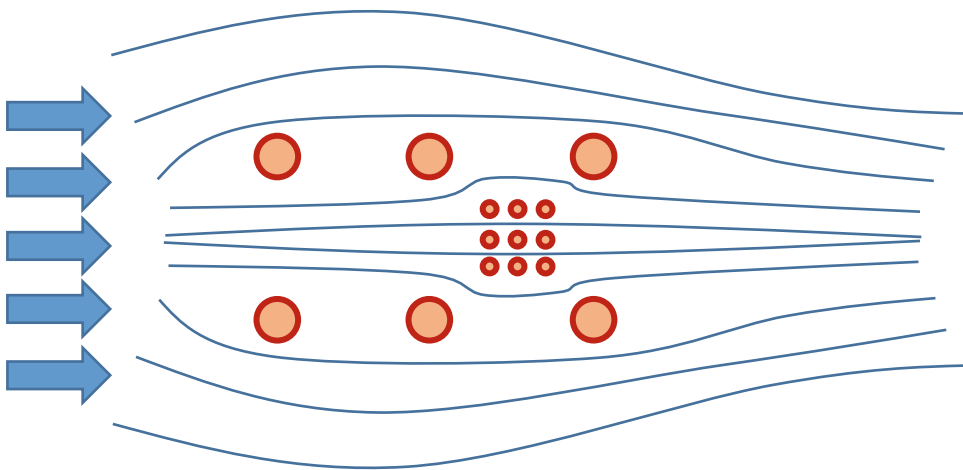
Many subsea structures contain many tubular members, all at different angles and often oriented so that the flow must go past multiple in-line members – as illustrated for a cross section through a typical offshore jacket as shown in Fig. 3. The incoming flow passes through the structure between the members, and some of the flow diverts around the structure due to the increased resistance of flow passing through the structure (Tong et al. 2015a, b). However, this diverted flow means that individual member sees a reduced flow velocity and reduced hydrodynamic forces due to the collective effects of the

group of members. To account for this reduction in local flow, simplified analytical estimates of the local flow reduction have been developed to improve upon the prediction of hydrodynamic forces using Morison's equations (see Taylor et al. (2013)).

These same principles can also be applied to very different scenarios of interest outside the oil and gas industry, for example, to predict the available energy that can be captured from an array of tidal turbines placed in a channel that experiences strong and predictable tidal flows. The influence of the turbines on each other and the overall influence of the farm of turbines on the flow through the channel can be captured using this approach. See, for example, (Adcock et al. 2013, 2021).

Biomimetic Design

A functional subsea system through biomimetic design is still rare, but like biologically inspired designs in other areas, it has a bright future in the subsea systems, especially for unmanned vehicles (Chu et al. 2012), such as remotely operated vehicles (ROVs) and autonomous underwater vehicles (AUVs). Through comparison hydrodynamics studies, the idea is to improve the underwater vehicles' maneuver efficiency to let them behave like aquatic animals.



Hydrodynamics for Subsea Systems, Fig. 3 Sketch of a cross section through a typical offshore jacket showing flow passing through and around the structure as a group of elements

Cross-References

► [Subsea System](#)

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Hydroelasticity

► [Station-Keeping System for VLFS](#)

Hydroelasticity Theory

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Synonyms

[Discrete-modules-based hydroelasticity method](#);
[Three-dimensional hydroelastic analysis method](#)

Definition

Hydroelasticity was first defined by Heller and Abramson (1959), who stated that “it is concerned with the phenomena involving mutual interactions among inertial, hydrodynamic and elastic forces.” Theoretically, the fluid pressure acting on a structure will cause rigid body motion and structural deformation, and rigid body motion and structural deformation will interfere with the surrounding flow field. Hence, fluid-structure interaction problems can be considered hydroelastic problems. However, compared with rigid body motion, the deformation of traditional offshore

structures, such as ships and platforms, is very small. The influence of deformation on the flow field can be ignored. Thus, when calculating the surface pressure, the structure can be regarded as a rigid body. According to structural analysis, the structure is regarded as a deformable body, and structural deformation is caused by fluid pressure. Therefore, hydroelasticity can be decoupled into a hydrodynamic problem and a structural analysis problem (Cui 2007).

Scientific Fundamentals

Historical Development Under Homogeneous Sea Conditions

Linear Hydroelastic Theories

Based on the “dry hull” modal analysis theory and strip theory, Bishop and Price (1979) established the well-known hydroelastic theory of ships. The hydrodynamic force of the ship is calculated by strip theory, as shown in Fig. 1, and the hydrodynamic force and structural damping are considered external effects on the “dry hull.”

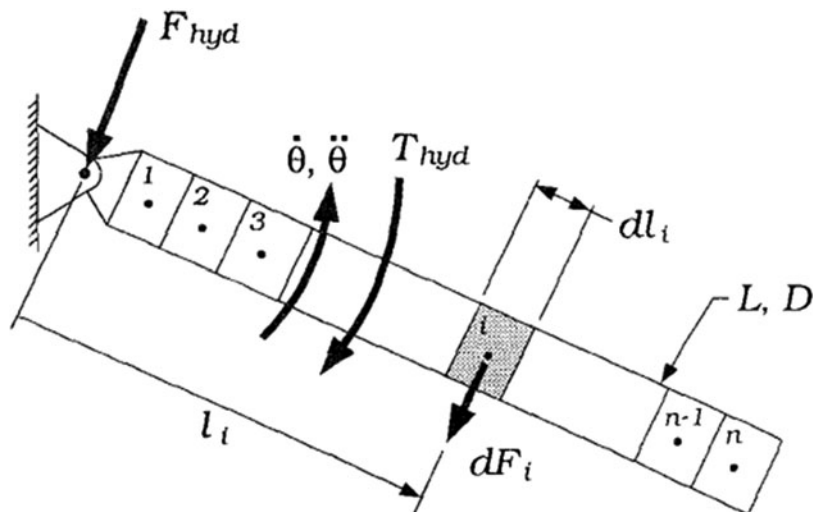
Based on this theory, Bishop et al. (1978) and Belik et al. (1980) simulated the slamming response of elastic hulls in irregular waves. Dong et al. (1989) and Dong and Lin (1992) analyzed the hydroelastic response of a shallow draft ship based on linear hydroelastic theory and

compared the numerical calculation results with the results of sectional ship model tests. Aksu and Bishop (1991) simulated the bending moment and shear response of a ballast oil tanker sailing in irregular long-peak waves in the time domain under three loading conditions. Zhong and Zhao (1998) studied the effects of wavelength and forward speed on the natural frequency and wave-induced responses. Bereznitski (2001) considered the ship as a rigid body and an elastic body and calculated its bottom-slamming response. Through a 2D experiment, Liu et al. (2001) studied the elastic deformation and mooring force of a large-scale floating structure under regular waves.

Although various two-dimensional hydroelastic methods have been widely used in practical engineering problems, their basic assumptions ignore the mutual interference of fluid motion along the length direction; therefore, they cannot solve the hydroelastic problems of marine structures with obvious three-dimensional characteristics. Wu (1984) and Price and Wu (1985) presented a general three-dimensional hydroelastic theory for a flexible body with an arbitrary shape traveling through or stationary in waves. Compared with two-dimensional hydroelastic theory, three-dimensional hydroelastic theory can effectively solve the fluid-structure interaction of arbitrarily shaped objects and can fully reflect the influence of the three-dimensional characteristics of the structure on the flow field.

Hydroelasticity Theory,

Fig. 1 Diagram of a strip-theory implement (McLain and Rock 1996)



Du (1996) established a complete frequency-domain analytical method of the 3D hydroelastic response theory. Wang (1996), who employed the time-domain Green function, developed a linear three-dimensional, time-domain, hydroelastic theory. Wu (1984) and Wu and Price (1985) derived a generalized interface boundary condition of flexible structures. A general linear boundary condition for the hydroelasticity of arbitrarily shaped structures was produced by Xia and Wu (1993). Based on the 2.5-D fluid method (Zhao and Aarsnes 1995), Hermundstad (1995) reported a linear hydroelastic theory for ships moving at a high forward speed. Malenica et al. (2003) analyzed the symmetric hydroelastic response of barges under shock waves and linear waves by using the inhomogeneous Timoshenko beam model and three-dimensional boundary integral method. Chen et al. (2005) analyzed the hydroelastic response of elastic offshore structures in regular waves based on three-dimensional linear hydroelastic theory, as shown in Fig. 2. The predicted vertical displacement along the structure agreed well with the experimental data, as shown in Fig. 3.

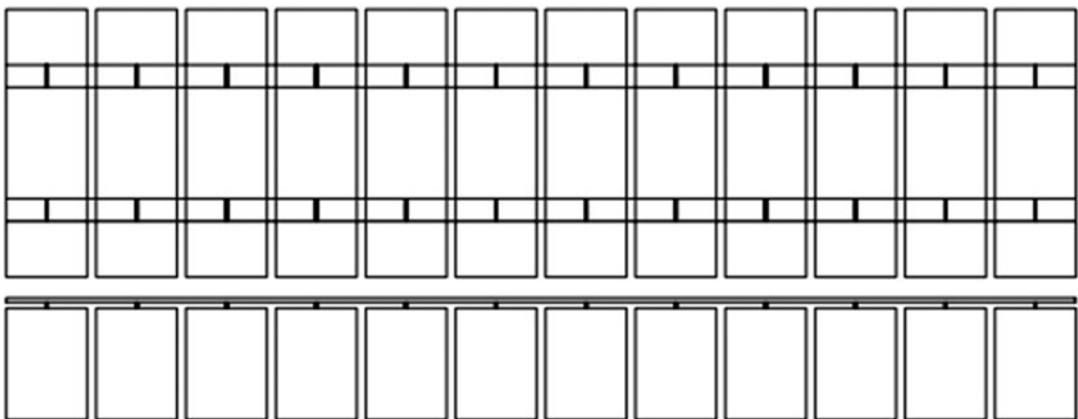
Nonlinear Hydroelastic Theories

The above research on the hydroelastic response is only applicable to the case of microwaves and the case of small amplitude motion and ship deformation. However, when the ship is sailing in

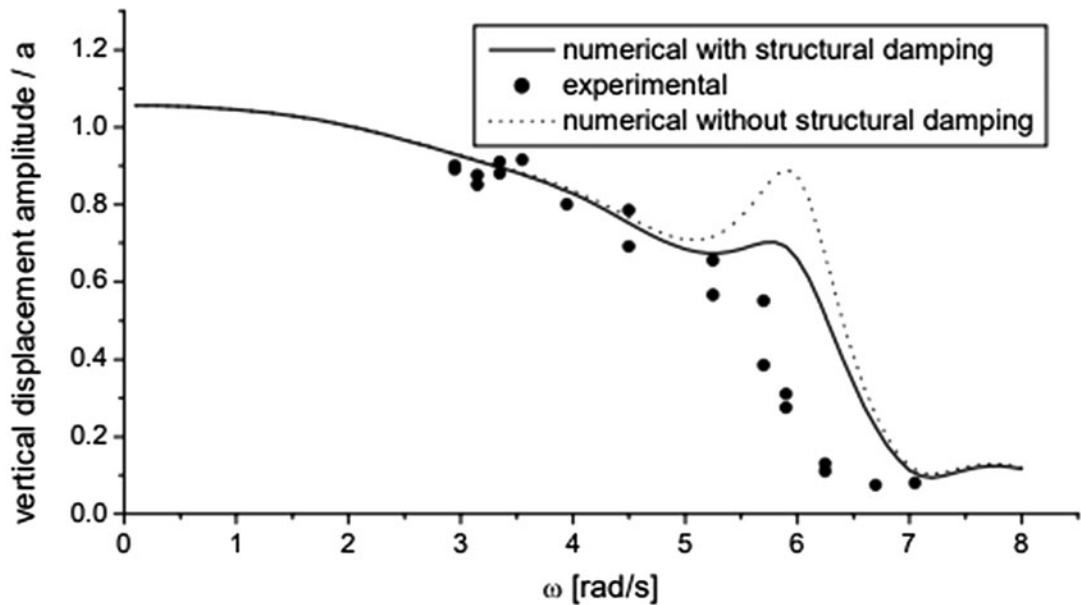
waves, it can be found that the wave load and motion present obvious nonlinear characteristics.

Based on perturbation theory, Jensen (1978) developed the nonlinear quadratic strip theory, which can consider the influence of nonlinear wave excitation forces on the motion response of a floating body. Jensen and Dogliani (1996) extended nonlinear quadratic theory (Jensen 1978) to predict the response of container ships to continuous wave excitation. Their results showed that the nonlinear response caused by the wave-induced vibration was equivalent to the linear response for the wave with a small average zero-crossing period; therefore, the nonlinear effect must be considered; for the long-peak wave, the nonlinear effect cannot be considered.

Based on the higher-order differential relationship between relative velocity and the corresponding hydrodynamic force established by Soding (1982), Xia and Wang (1997) derived a differential form, the hydroelastic equation, which can take into account the nonlinear hydrodynamic effects caused by bow drift, and they predicted the hull response in irregular waves, which was in good agreement with experimental results. Based on the two-dimensional potential flow theory, Xia et al. (1998) derived a two-dimensional, time-domain, strip hydroelastic theory, which can take into account the effects of nonlinear hydrostatic-restoring forces and nonlinear F-K forces. Park and Temarel (2007)



Hydroelasticity Theory, Fig. 2 Schematic outline of a model structure (Chen et al. 2005)



Hydroelasticity Theory, Fig. 3 Vertical displacements along the length under regular waves (Chen et al. 2005)

proposed a two-dimensional, nonlinear hydroelastic analytical method that can simultaneously take into account the nonlinear effects of radiation forces, F-K forces, and hydrostatic-restoring forces.

Wu and Moan (1996) and Wu and Hermundstad (2002) established a nonlinear hydroelastic analytical method for ship hulls under nonlinear vertical loads. In this method, the influence of the nonlinear vertical load is divided into a linear part and a residual part (nonlinear correction term). The linear part can be predicted by 2D, 2.5-D, or 3D potential flow theory; the residual part includes convolution of the time-delay function and nonlinear correction force due to slamming. Based on the fully coupled, three-dimensional, finite-element, structural model and three-dimensional, boundary-element, hydrodynamic model, Malenica and Tuitman (2008) proposed a generalized hydroelastic model that can analyze wave-induced vibrations and flutters of container ships in the frequency and time domains.

For second-order, nonlinear wave loads, Wu et al. (1997) first considered the influence of the nonlinear second-order, hydrodynamic load on

the response of a floating body caused by the rotation of a rigid body and the change in the free instantaneous wet surface and proposed the three-dimensional, nonlinear hydroelastic theory. Based on this theory, Chen (2001) deduced the generalized form of the second-order, hydrodynamic coefficients of the sum frequency and difference frequency of the floating body and the generalized form of the hydrodynamic force of the anchored floating body and established the three-dimensional, nonlinear frequency domain, hydroelastic motion equation of the anchored floating body. Based on the pressure integral method, Maeda et al. (1997) and Ikoma et al. (2000) simplified the floating body as an elastic plate and studied the influence of the second-order, wave load of difference frequency acting on the very large floating structure (VLFS) on the vertical displacement of the floating body by using thin plate hydrodynamics. For a floating body with a large deformation, the influence of the membrane force of the plate on the elastic response of the floating body should be considered. Chen et al. (2003a, b) established a frequency-domain and time-domain hydroelastic method that can consider both fluid and structural

Hydroelasticity Theory, Table 1 Particulars of the plate by Chen et al. (2003a)

Length L	300.0	m
Width B	60.0	m
Depth h	2.0	m
Equivalent plate thickness h^*	0.766	m
Draught d	0.5	m
Young's modulus E	1.19×10^{10}	N/m ²
Poisson's ratio ν	0.13	
Mass density ρ_b	256.25	kg/m ³

nonlinearity. The key parameters of the structure are listed in Table 1. Taking the motion and deformation of a large deformation-floating flat plate in multiple regular waves as an example, the influence of membrane forces on structural deformation and stress was discussed, as shown in Fig. 4.

Analytical Method Under Homogeneous Sea Conditions

Linear Hydroelastic Theories

According to the linear problem, the following assumptions are made:

1. The amplitude of the surface wave is assumed to be small in comparison to its wavelength.
2. The motions and distortions of the structure are assumed to be small and continuous.
3. The material is assumed to be linear.

Based on these assumptions, the linear hydroelastic problem can be solved through the frequency-domain method and time-domain problem, and the time-domain method can be transformed into the frequency-domain method by Fourier transform or Laplace transform.

Frequency-Domain Method The equation of hydroelasticity in the frequency domain can be written as

$$[-\omega^2(a + A) + i\omega(b + B) + (c + C + \Delta C)]p = F_w \quad (1)$$

where the response is assumed to be sinusoidal with the same frequency as the excitation, where

- $\{a\}$ is the symmetric inertial matrix of the dry hull.
- $\{A\}$ is the added-mass matrix.
- $\{b\}$ is the structural-damping matrix.
- $\{B\}$ is the hydrodynamic-damping matrix.
- $\{c\}$ is the diagonal stiffness matrix of the dry hull.
- $\{C\}$ is the fluid-restoring or stiffness matrix.
- $\{\Delta C\}$ is the additional restoring matrix from the linear external actions such as mooring lines.
- $\{p\}$ is the column matrix of the principal coordinates.
- $\{F_w\}$ is the wave excitation matrix including the Froude-Krylov and diffraction forces.

In addition to the elastic deformation of the structure, the largest difference between the seakeeping quality and hydroelasticity is the effect of deformation on the flow field. In potential flow theory, this effect is mainly reflected in the difference in radiation potential.

1. For seakeeping quality, the radiation potential ϕ_R is caused by rigid body motions (six degrees of freedom).

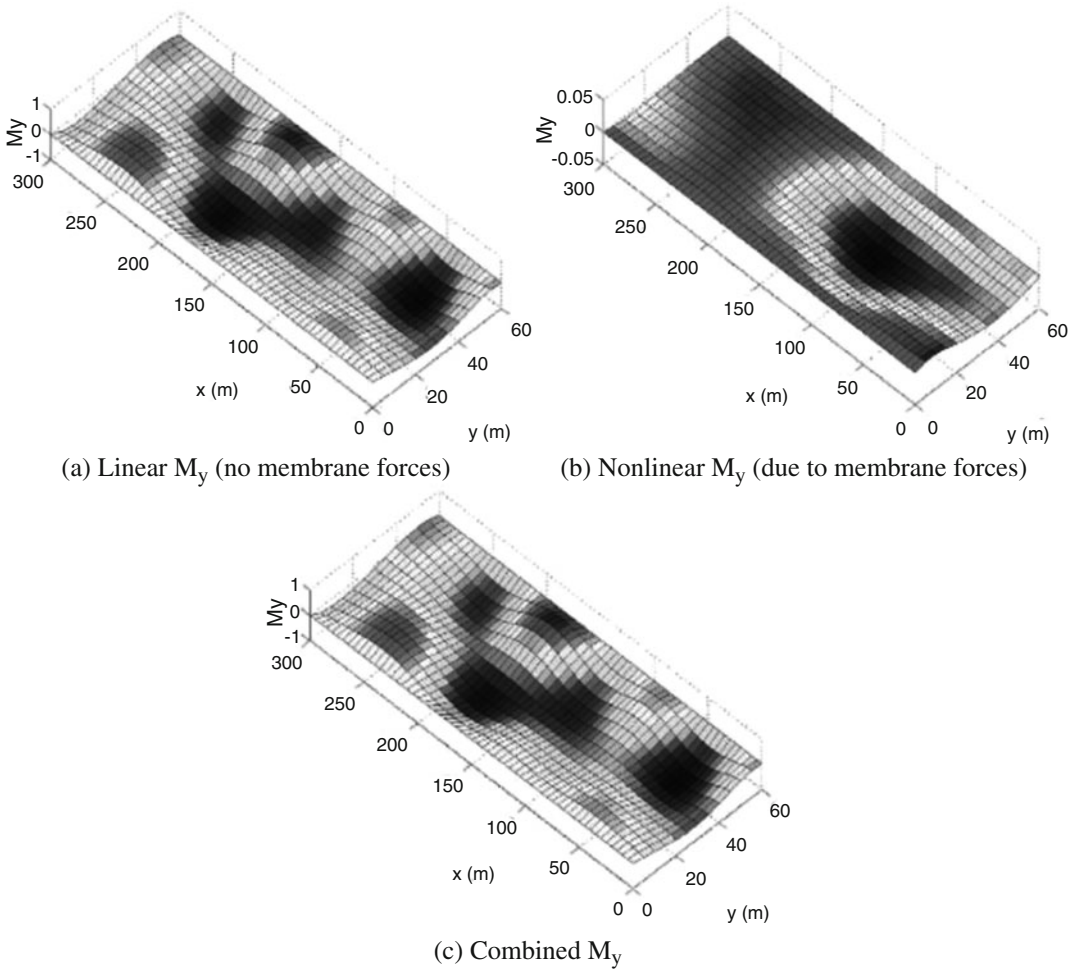
$$\phi_R = \sum_{r=1}^6 \varphi_r \dot{p}_r$$

where φ_r is the normalized velocity potential; and $\dot{p}_r$ is the principal coordinates. The dot sign represents differentiation with respect to time.

2. For the hydroelasticity, the radiation potential ϕ_R is caused by rigid body motion and elastic deformation (6 + N degrees of freedom).

$$\phi_R = \sum_{r=1}^{\infty} \varphi_r \dot{p}_r$$

The radiation force consists of two parts. One is proportional to acceleration, and the proportional coefficient is called the added mass $\{A\}$; and the other is proportional to velocity, and the proportional coefficient is called the damping coefficient $\{B\}$.



Hydroelasticity Theory, Fig. 4 Different contributions of bending moments, M_y in Chen et al. (2003a)

Time-Domain Method For the linear time-domain problem, the superposition method can be used.

$$\begin{aligned}
 (a + A')\ddot{p}(t) + b\dot{p}(t) + \int_0^t H(t - \tau)\dot{p}(\tau)d\tau \\
 + (c + C + \Delta C)p(t) \\
 = F_w + F_c
 \end{aligned} \quad (2)$$

where $A' = A(\omega) + \frac{1}{\omega} \int_{-\infty}^{\infty} H(t) \sin(\omega t) dt$; $H(t) = \frac{2}{\pi} \int_0^{\infty} [B(\omega) \cos(\omega t)] d\omega$; and F_c is the column vector of the generalized concentrated force.

Nonlinear Hydroelastic Theories

Transient phenomena observed in the hydroelasticity of ships and other floating structures can only be studied in the time domain. The nonlinearity of hydroelasticity may come from the nonlinear excitation force and nonlinear structure.

1. Nonlinear Excitation Forces

This may be caused by whipping induced by ship slamming or underwater explosions, dynamic capsizing, collision and grounding, cobblestone oscillation (Cui 2007), moving loads, mooring forces, etc.

2. Nonlinear Structure

This may be caused by nonlinear structural materials, nonlinear mooring systems, nonlinear connectors, etc.

The nonlinear hydroelastic problem can be expressed as

$$[M]\ddot{x} + [C]\dot{x} + [K]x = F \quad (3)$$

where $[M]$, $[C]$, and $[K]$ are the generalized mass, damping coefficient, and stiffness matrices, respectively; F is the generalized force vector, usually consisting of concentrated forces, such as mooring forces and landing forces of aircraft, and distributional fluid forces on the wetted surface of the body; and $\{x\}$ is a vector (column matrix) of generalized coordinates.

Hydroelasticity Under Inhomogeneous Sea Conditions

The above discussions about hydroelasticity are all relevant to homogeneous sea conditions. For an ultralarge floating body, coastline, or island reef constructed near the shore or near the island reef, the seabed topography will change, and the homogeneous seabed assumption may no longer be applicable. To reduce the response of a very large floating body, breakwaters are often set around the floating body. Due to the refraction

effect of the breakwater wall, the location of the very large floating body and the interaction between the breakwater and VLFS should be considered. In addition, due to the large size of the VLFS, which can span several kilometers of sea area, the sea conditions at both ends are very different; therefore, it cannot be described by a single spectrum. The displacement response and load of the structure may be increased under inhomogeneous sea conditions. There are two methods to predict the hydroelastic response under inhomogeneous sea conditions.

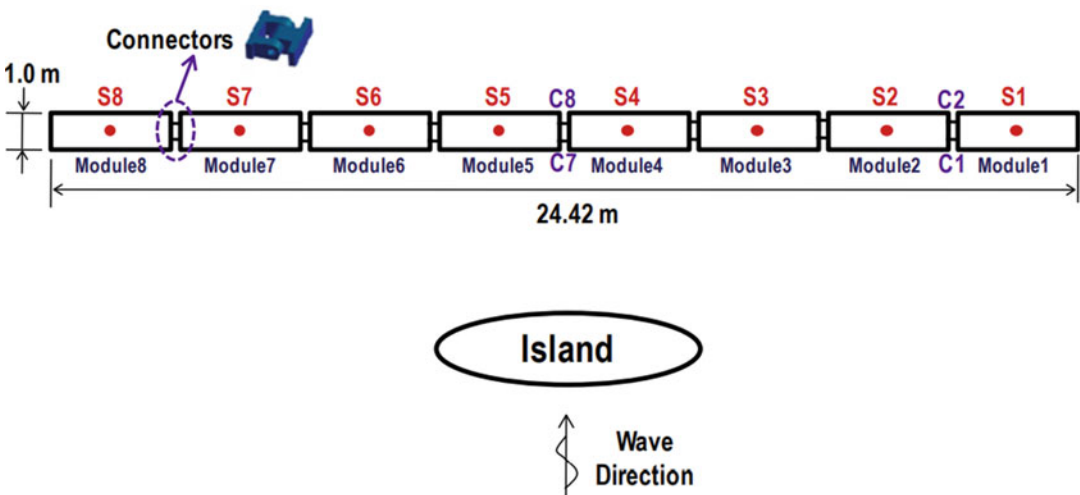
Three-Dimensional Hydroelastic Method

Based on the three-dimensional hydroelastic theory, Ding et al. (2019) developed a simplified method to estimate the hydroelastic response of the VLFS, which can consider the inhomogeneous incident wave angle. Compared with the experimental results, although there are some errors, the simplified model can to some extent predict the hydroelastic response of the VLFS. A schematic of the VLFS model and the comparison between the predictions and experimental results are shown in Figs. 5 and 6.

A Discrete Module-Based Hydroelastic Method

There is a new method to address fluid-structure coupling proposed by the Fu Shixiao group from Shanghai Jiaotong University (Wei et al. 2017,

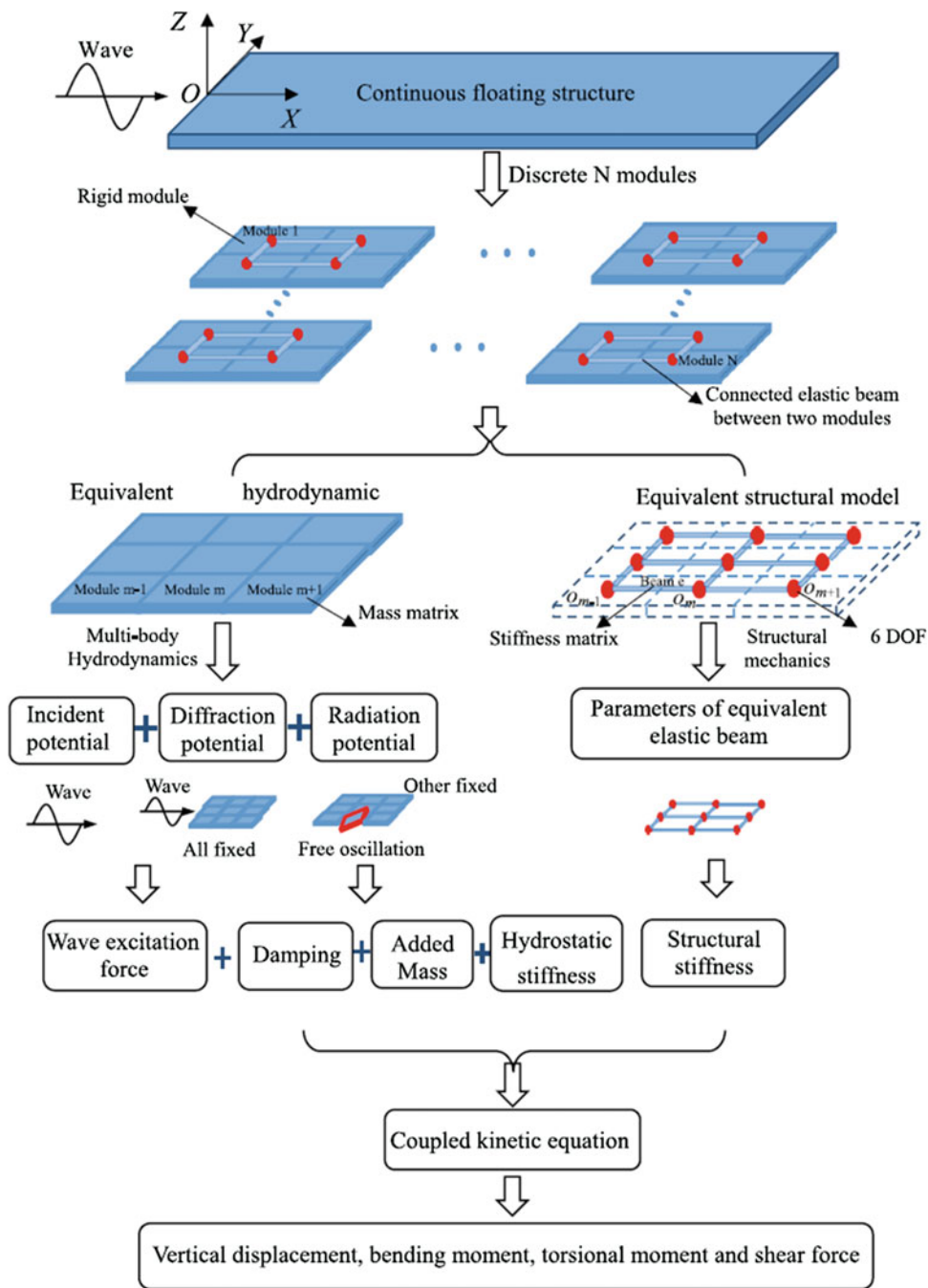
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Hydroelasticity Theory, Fig. 5 The sketch of the 8-module VLFS model and the connectors (Ding et al. 2019)

B1- $H_s = 1.30$ m $T_p = 6.6$ s					B2- $H_s = 3.98$ m $T_p = 8.4$ s				
Exp. (a)					Exp. (a)				
Numerical(b)					Numerical(b)				
(a-b)/a					(a-b)/a				
(a-b)/a					(a-b)/a				
Heave ₁ (m)	0.137	0.090	34.3%	0.286	0.200	30.1%			
Roll ₁ (°)	0.195	0.160	17.9%	0.611	0.420	31.3%			
Pitch ₁ (°)	0.023	0.016	30.4%	0.079	0.065	17.7%			
Heave ₄ (m)	0.098	0.064	34.7%	0.158	0.084	46.8%			
Roll ₄ (°)	0.134	0.078	41.8%	0.489	0.354	27.6%			
Pitch ₄ (°)	0.039	0.027	30.8%	0.083	0.065	21.7%			
B3- $H_s = 4.89$ m $T_p = 10.2$ s					B4- $H_s = 5.54$ m $T_p = 10.7$ s				
Exp. (a)					Exp. (a)				
Numerical(b)					Numerical(b)				
(a-b)/a					(a-b)/a				
(a-b)/a					(a-b)/a				
Heave ₁ (m)	0.925	0.700	24.3%	1.290	1.010	21.7%			
Roll ₁ (°)	1.520	1.300	14.5%	1.736	1.430	17.6%			
Pitch ₁ (°)	0.125	0.080	36.0%	0.172	0.140	18.6%			
Heave ₄ (m)	0.355	0.200	43.7%	0.480	0.280	41.7%			
Roll ₄ (°)	1.272	0.900	29.2%	1.333	1.020	23.5%			
Pitch ₄ (°)	0.271	0.180	33.6%	0.362	0.200	44.8%			

Hydroelasticity Theory, Fig. 6 Significant values of motions of module 1 and 4 from the predictions and model tests (Ding et al. 2019)



Hydroelasticity Theory, Fig. 7 Flow diagram of the numerical simulation (Wei et al. 2017)

2018, 2019; Li et al. 2018). Figure 7 provides an overview of the process performing discrete module-based hydroelastic analysis. The floating structure is first discretized into a set of rigid

modules that are connected by elastic beams. Considering the hydrodynamic interactions between modules, multibody hydrodynamic theory is adopted to obtain the velocity potential of

the flow field and the wave excitation forces, added mass, and damping coefficients of various modules. The motions of each module are affected by the hydrodynamic interactions with the surrounding modules and are restricted by the displacement continuity of adjacent modules. The displacement continuity between modules is guaranteed by establishing an elastic beam with a uniform section stiffness matrix between the equivalent centers of the modules. The displacements can be obtained by solving the coupled kinetic equation. Then, the bending moments, shear forces, and torsional moments of the floating structure are determined based on the theory of structural mechanics.

Cross-References

- [Aquaculture Structures: Numerical Methods](#)
- [Connectors of VLFS](#)
- [Experimental Techniques for VLFS](#)
- [Floating Bridge](#)
- [Mega-Float](#)
- [Station-Keeping System for VLFS](#)
- [Structural Analysis and Design of VLFS](#)
- [Very Large Floating Structures \(VLFS\): Overview](#)

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Hydrogen

- [Alternative Fuel for Ship Propulsion](#)

Hydrostatics Pressure

- [AUV/ROV/HOV Hydrostatics](#)

Ice Breaking Ships

► Ice Breaking Vessel

Ice Breaking Vessel

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Synonyms

Ice breaking ships; Icebreaker

Definition

Ice breaking vessel is also called icebreaker, a special purpose ship or boat designed to move and navigate through ice-covered waters and protect the ships entering and leaving the frozen ports and anchorages or guide the ships to navigate in the ice area. They are divided into river, lake, harbor, or sea ice breaking vessels.

Scientific Fundamentals

Characteristics of Ice Breaking Vessel

In the history of ship design and construction, icebreaker is a relatively late development ship. The earliest ice breaking ship in the world was born in Philadelphia, United States, in 1837, a ship named “City ice boat No. 1.” The first nuclear-powered icebreaker “Lenin” was born in the former Soviet Union in 1959. Ice breaking vessel shows obvious differences from normal ships in its strengthened hull, ice-clearing shape, and power (Hu 2004).

The hull of ice breaking vessel is short and wide, the ratio of length to width is small, the bottom ends are upturned, forward perpendicular is sharp and forward, and the overall strength is high. Stem, stern, and waterline areas are also strengthened with thick steel plates and dense skeletons. Additionally, the hull lines of icebreaker are usually designed to make the flare at the waterline small enough, so as to minimize the ice breaking force. In consequence, the characteristics of ice breaking ships are a sloping or rounded stem with sloping sides and a short parallel midship to enhance maneuverability in ice (Roles et al. 1969). However, poor hydrodynamic efficiency and seakeeping characteristics of the spoon-shaped bow and round hull make the icebreaker easy to be influenced by slamming

(Riska 2011). Thus, the hull of an icebreaker is often at a balance between minimum ice resistance, maneuverability in ice, low hydrodynamic resistance, and adequate open water characteristics (Segercrantz 1989). There is a kind of hull which is wider in bow than in the stern. They are called “reamers,” and they can increase the width of the ice channel; therefore, frictional resistance in the aft ship is reduced, and the ship’s maneuverability in ice is improved. Low-friction paint is often used to reduce friction protecting the ship’s hull from corrosion; besides, making use of an explosion-welded abrasion-resistant stainless steel ice belt can also achieve the purpose. Auxiliary systems are used to reduce friction by forming a lubricating layer between the hull and the ice, for example, powerful water deluges and air bubbling systems. Pumping water between tanks on both sides of the vessel makes continuous rolling to reduce friction and makes progress through the ice easier (Segercrantz 1989). Over the years, to further reduce the ice resistance and create an ice-free channel, experimental bow designs like a cylindrical bow and the flat Thyssen-Waas bow have been tried a lot (Zhang et al. 2017).

Additional structural strengthening is required for icebreakers and other ships which are operating in ice-filled waters because of the existence of various global and local loads resulting from the contact between the hull of the vessel and the surrounding ice. Because ice pressures vary between different regions of the hull, for an ice-going vessel, the most strengthened areas are the bow which experiences the highest ice loads, and around the waterline, the additional reinforcement above and below the waterline forms a continuous ice belt around the ship (Guard 2012).

Short and stubby icebreakers are usually built with transverse framing, in which the shell plating is stiffened and the frames are placed about 400 to 1000 mm, relative to the longitudinal frame used in longer ships. Near the waterline, the frames running in vertical direction distribute the local ice loads on the shell plating to longitudinal girders called stringers, and they are supported by web frames and bulkheads that carry the global hull loads. In older ice-going vessel, the shell plating directly in contact with the ice can reach

50 mm thickness, and the high strength steel is used in the modern ice breaking vessel (Guard 2012). The yield strength is up to 500 MPa. The result is that with the same structural strength, the material thickness is smaller, and the steel weight is lower. Whatever the strength, the steel used in the hull structure of an ice breaking ship must be able to resist brittle fracture in low ambient temperatures and high loading conditions, both of which are typical phenomena in the waters covered with ice (Guard 2012; Norden 1989). From the development of power system, the propulsion capacity of icebreaker has improved significantly from steam power and diesel power to nuclear power. Since steam power has been unable to meet the needs of the current polar strategy, most of the power systems of ice breaking ships in the world have adopted the diesel power system, and more advanced is the nuclear propulsion system. From the propulsion device, both the steam engine and the diesel engine pass the power to the fixed pitch propeller through the deceleration gear. This kind of system is easily damaged by the mechanical damage caused by the propeller hitting the broken ice produced with ice breaking process. At present, most ice breaking vessels have improved to use of electric drive, separating the original motive from the propeller to reduce the damage of ice impact, making the engine room more flexible, and no longer needing the drive shaft and the deceleration gear. At the same time, a nozzle or diversion pipe is added around the propeller, which not only better protects the propeller but also improves hydrodynamic factors around the propeller and increases thrusting power. In addition, the most advanced icebreaker also uses an omnidirectional pod system, a closed device suspended under the hull of a ship, which contains an electric motor directly connected with the propeller. The omnidirectional pod can be rotated at 360°, thus forcing propulsion in any direction. This podded thruster cooperates with the lateral propeller of the bow. It can realize the dynamic positioning of the ship and greatly improve the mobility of the icebreaker in the ice and open waters (Shen and Zhao 2010).

The main features of ice breaking vessel above can be summarized as follows:

1. Compared with hull size, icebreaker has greater weight and can crush sea ice more effectively.
2. The hull under the waterline uses special coatings to reduce the friction between hull and ice at low temperatures.
3. Special steel which has high performance at low temperature is used for extra thickening near the bow, stern, and water line;
4. Have a gentler angle of the bow, making icebreaker more accessible to climb up ice.
5. No anti-roll device and hull outburst.
6. High-power thrusters.
7. The shape of the hull helps protect the rudder and propeller. It prevents the rudder and propeller from being damaged by sea ice when reversing.

Ice Breaking Mode

Ice breaking capability is one of the most important parameters of ice breaking ships, which can be divided into continuous ice breaking capacity and repeated punching ice breaking capacity (Guard 2012). At present, most of the diesel icebreaker has a continuous ice breaking capacity of more than 1.2 m, and more advanced ice-going vessel can break 1.8 m, and nuclear power system ice breaking ships can break the ice of 2–3 m thickness continuously (e.g., Russian “Tamil” continuous voyage in level ice with thickness of 2 m; a more powerful icebreaker “Victory 50th anniversary” can break the ice with thickness of 3 m) (Yu et al. 2017).

In addition to using their own weight, the traditional ice breaking vessel usually needs to coordinate the front and rear pressure tanks to complete the ice breaking operation. For example, when the heading is breaking ice, the back-pressure tank is filled with water, so that the bow is upwards and keeps going, the front half of the hull is pressed on the ice, then the front-pressure tank is filled with water, and the back-pressure tank is drained, and the ice is crushed as the hull center of gravity moves forward; when both the left and right sides are breaking ice, the ballast tanks at left and right are injected water alternately, so that the hull is swaying, thus breaking the ice. However, when ice breaking vessel is

difficult to navigate continuously in ice regions, it can carry out the ice breaking operation by repeated stamping. First, the icebreaker needs to reverse the distance of the 1–2 length of ship, and accelerate forward, then the ice will be broken. The bow of the icebreaker is able to flush the ice easily, with the weight of the hull and the ballast water, so that the ice can be broken into pieces. In thicker ice areas or ice ridges, icebreaker breaks ice through repeated collisions several times.

Generally speaking, the ice breaking thickness of an icebreaker in repeated punching mode is usually two to four times of the breaking ice thickness using continuous mode. For example, Russian “Arctic” can break ice continuously with thickness of 2.25 m, and repeated punching can break ice with 5 m thickness; with American “Polar Star,” the thickness of continuous ice breaking is 1.8 m at the speed of 3 kn, and the thickness of repeated stamping ice breaking can reach 6.4 m.

In addition, it is common to use auxiliary ice breaking equipment/systems including flushing system, bubble system, hull heating equipment, and rapid rollover system. For example, the largest nuclear-powered icebreaker “Arctic” is installed with the tile system and the rapid tilting system. The bubble system can eject highly compressed air from the bottom of the hull, making the ice block on the side of the ship acting on the upward buoyancy and breaking up with the rise of the bubble to reduce the friction of the hull; the rapid lateral inclination system of the ship can change the weight distribution of the hull through a powerful pumping system, so that the hull will be swayed under the ice in tight condition, and the ship body can be shackled away from the ice and snow on both sides. In addition, Russian “Vladislav•Sterinov” icebreaker is installed with the hull heating equipment, which can be heated to prevent the hull from freezing, thus avoiding the toughness of the hull steel due to low temperature.

Special Ice Breaking Vessel

Asymmetric Professional Ice Breaking Ship

Due to special functional requirements and power requirements, the size of professional icebreaker

cannot be too large, especially for the limitation of ship width (usually around 20 m). With the increasing demand for transportation and the development of harbor waterway, the 20 m ship wide icebreaker has been unable to meet the growing actual demand. The traditional method is to break the ice using two ice breaking vessels at the same time, but this greatly improves the cost of breaking the ice. To solve this problem, ship design experts have developed and designed a new type of asymmetric hull ice breaking ship “Baltika,” which is 76.4 m in length and 20.5 m in width and has a draught of 6.3 m in summer. The ship can sail diagonally. When the thickness of ice is 60 cm, a channel with width of 50 m can be opened at 2 kn. The average hull width of the icebreaker in the Baltic Sea is 25 m, and the width of an Apollo type tanker is more than 40 m, so with the help of “Baltika” icebreaker, it can replace two ordinary icebreakers. In addition, it is more efficient to navigate at a speed of 3 kn in ice with 1 m thickness and navigate at a speed of 6 kn in ice of 60 cm thickness, and the maximum speed in open water is 14 kn. With the integration of this unique ship type and full rotary propulsion technology, this icebreaker also has good flexibility, versatility, and maneuverability.

Bi-Directional Ice Breaking Vessel

Bi-directional ice breaking vessel, or called double acting ice breaking vessel, is a new type of ice breaking vessel that supplements and strengthens ice breaking functions on the basis of dual power technology, such as the first double-powered icebreaker “Vasily Dinkov.” The stem and stern direction of the bi-directional icebreaker can carry out continuous ice breaking operation. It is suitable for navigation in open waters when the bow is forward, and it can be suitable for ice breaking operation when the ship stern goes forward, so as to balance the combination properties both in ice breaking and open waters. It is important to note that the operation of a bi-directional icebreaker at the stern of the icebreaker does not simply strike the ice, but flexibly turns the full rotary thruster back and forth, using powerful water to flush, move, and crush the sea ice, so that it eventually collapses and reaches the

purpose of breaking the ice (especially ice ridge ice), and turning the stern of the ship as well as widen the channel. However, this process is slow relatively.

Ice Breaking Hovercraft

Hovercraft is also used to break ice. There are two different models for air cushion ice breaking mechanism: low speed mode and high speed mode (Buck and Pritchett 1978; Muller 1979).

In low speed operating mode, the air cushion pressure causes the surface of the ice to fall down. The simplified model of the ice is a cantilever beam model without the support of water, so the ice is broken when the stress of the ice is large enough.

In high speed mode, ice breaking occurs when the velocity of energy transmission in ice is equal to the velocity of the air cushion icebreaker. The energy of the ice cannot be radiated and accumulates continuously, so that the amplitude of the fluctuation of the ice is increasing, and the tensile stress in the ice exceeds the yield limit of the ice. Therefore, the key to ice breaking at high speed is to find the speed of ice-water energy transmission, that is, critical speed. Besides, air cushion icebreaker parameters such as air cushion pressure and air cushion height also affect the ice breaking process.

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Ice Fracture

- Numerical Simulation Floe Ice–Sloping Structure Interactions

Ice Loads

- Offshore Structure Design Under Ice Loads

Ice Management in Offshore Operations

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Synonyms

Ice management system; Iceberg management; Physical ice management; Sea ice management

Definition

Ice management (IM) is defined in ISO 35104 as the sum of all activities, carried out with the objective to mitigate hazardous situations by reducing or avoiding actions from any kind of ice (sea ice or glacial ice). This includes, but is not limited to:

1. Ice detection, including tracking and forecasting

2. Threat evaluation, including procedures to identify potentially hazardous ice features or ice situations
3. Physical ice management, including methods to physically manage hazardous ice features or ice situations (e.g., icebreaking, ice clearing, iceberg towing)
4. Ice alert procedures designed to initiate appropriate operational reactions to ice hazards or adverse ice situations in a timely manner
5. Procedures associated with the safe shutdown of floating structures (moored or dynamic positioning), both active (move off and ice management) and semi passive (ice management, but no move off)
6. Procedures associated with the safe shutdown of bottom-founded structures, both active (with ice management and move-off capability) and passive (fixed with ice management)

Typical components of an ice management system are illustrated in Fig. 1.

Scientific Fundamentals

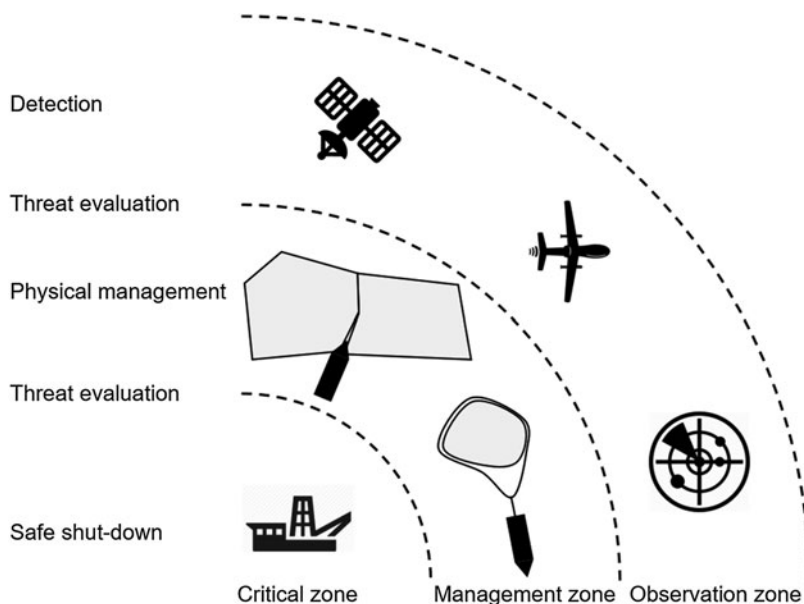
Ice management involves various operational procedures that can be used to reduce global and local design ice actions. General requirements of an ice management system and recommendations and guidance for the design of Arctic offshore structures can be found in ISO 19906. There is also a plenty of literature on ice management and associated systems. Some highlights of the relevant methods and techniques are presented below.

Ice Detection

Typically, surface scouting tools such as satellites, fixed wing aircrafts, helicopters, marine radars, drift buoys, and visual observations from icebreakers have been used for ice detection. By studying previous experiences with ice management, Eik (2008) stated that no single tool or method was able to detect all kind of ice features. Use of unmanned aerial vehicle (UAV), autonomous underwater vehicle (AUV), and multibeam sonar may be considered as possible future supplements to existing ice detection tools (Eik and

Ice Management in Offshore Operations,

Fig. 1 Typical components of an ice management system. (Figure based on ISO 19906)



Løset 2009). There is also a need to develop tools for gathering, analyzing, and presenting all information in a quick and efficient manner.

Threat Evaluation

According to ISO 19906, the methods used for threat evaluation should consider the information provided by the ice detection system; forecast movements and changes to any ice threats and their expected time of arrival; characterize ice events in terms of potential consequences to the installation for the various operations and phases of the development; identify circumstances when active intervention in the form of physical ice management activities is required; and identify the thresholds where installation-specific operational response actions (e.g., production shut-down, disconnect, personnel evacuation) are triggered and the advance warning time required for the response actions to be carried out.

Ice Alert Procedures

Ice alert procedures provide an indication of the severity of ice conditions affecting an offshore operation, identify hazardous ice features or situations to it, and outline a series of pre-defined response actions, including physical ice management. An effective ice alert procedure is critical to

operations and is designed to ensure that personnel on the installation, its support vessels, and the management ashore understand the ice situation, so they can immediately recognize and address changing circumstances. A typical ice alert system uses a color (or number) code to indicate the level of threat. To ensure that appropriate actions are taken by the installation and its support vessels in various ice conditions, alert levels can be defined by the proximity of potentially hazardous ice conditions to the operations area and/or the observed responses of the installation (Dunderdale and Wright 2005).

Physical Ice Management

Physical ice management refers to the actual vessel operations that are intended to break up and clear sea ice and/or specific features within an ice cover and deflect icebergs and small glacial ice masses (ISO 19906). Specific approaches and techniques have been developed for both sea ice and iceberg management.

Sea Ice Management

Dunderdale and Wright (2005) described five basic sea ice management techniques used to reduce ice load levels and defend floating platforms that are station-keeping on a mooring or by

dynamic positioning. Three of the five basic techniques involve particular icebreaking patterns, including the linear, sector, and circular techniques. The remaining two are ice clearing procedures that include pushing heavy ice floes and the use of high-power propeller wash to break thin ice and clear brash ice and small floes from around the station-keeping unit.

1. **Linear technique:** The linear technique is an icebreaking pattern used by a support vessel to break pack ice up-drift of a floating platform in straight lines, parallel to the direction of ice drift. This icebreaking pattern is typically used when the ice drift speed is fast and the ice drift direction remains reasonably constant.
2. **Sector technique:** The sector technique is an icebreaking pattern that provides wide managed pack ice coverage around the approaching ice drift direction. This technique requires the support vessel to steam back and forth across the drift line between two designated bearings that create the sector. This pattern is typically used when ice drift speed is slow and/or when the drift direction is variable or changing rapidly.
3. **Circular technique:** The circular icebreaking technique is a procedure that requires the support vessel to steam in a circular pattern up-drift of the platform location. The diameter of the circles will vary with the speed of the ice drift and the maneuverability and speed of the support vessel. This pattern is typically used in high concentrations of thin ice or small diameter thick ice floes and when the ice drift direction is variable. A circular pattern is also made completely around a platform as an effective method to relieve ice pressure.
4. **Ice pushing technique:** The ice pushing technique is an effective way of removing medium and large ice floes from the drift line. The pushing direction is usually at right angles to the approaching ice. The benefit of pushing a large floe instead of breaking it is that the threat to the platform is removed from the drift line, whereas if the ice is broken up-drift, the broken remnants may still pose a threat. Care must be taken to properly forecast any change to the ice

drift to ensure that the ice will not become a threat at a later time. To allow full power pushing, the bow strength of the vessel used must be appropriate. At least two vessels are often used to prevent floe rotations.

5. **Propeller washing technique:** Propeller washing of small pieces of thick ice, even if present in high concentrations, can be very effective to reduce or prevent ice accumulation against the platform. This technique is particularly effective when used by vessels fitted with azimuth thrusters. Such a system allows the support vessel to remain almost stationary up-drift of the platform on the drift line, with the propellers angled outward and using high power to wash ice to each side of the platform. There are sometimes restrictions to the use of this approach, for example, if there is only one vessel and poor visibility which prevents knowing what ice may be coming from further up drift.

In addition to these techniques, there are a number of variations and combinations that are effective in certain ice situations. When more than one ice management vessel is operating up-drift of an installation, each can use a different technique that is most suitable for its operating capability.

Iceberg Management

McClintock et al. (2007) summarized the iceberg towing and deflection techniques used on the Grand Banks.

1. **Single vessel iceberg towing procedure:** The standard method for deflecting an iceberg posing a threat to offshore facilities has been to encircle the iceberg with a heavy floating rope which can act as a bridle from which the iceberg may be towed by an offshore supply vessel. For larger icebergs, the objective is to deflect the ice mass by a few degrees from its naturally preferred route. The technique has good reliability for medium to large icebergs but is ineffective for small iceberg.
2. **Dual vessel towing:** Towing is conducted with two vessels to deflect large or unstable ice masses. The approach requires two vessels, a

single wire rope, and one steel hawser for each vessel. The approach is practical and can produce a significantly greater tow force; however, there can be difficulties in having a balanced vessel thrust, a tendency to see-saw around the iceberg while towing, and difficulty in maintaining depth control over the tow wire.

3. **Water Cannon:** Water cannons are actually fire monitors or nozzles mounted on the superstructure of the support vessels which can direct a high-pressure stream of sea water to deflect small ice masses. While their primary function is firefighting, tests and experience have proven them highly effective in managing small ice masses which might not be towable.
4. **Propeller washing:** Propeller washing can be a generally successful technique for small ice mass deflection, typically used when a bergy bit or growler is close to facilities and its drift path can be accurately predicted. The tow vessel repeatedly backs up to the piece of ice and accelerates away, thus moving the ice in the desired direction with the backward thrust of the water from the propellers.

Station-Keeping Considerations

There are a wide range of platforms and vessels that have been used in almost any kind of ice regime (Eik 2008). Some representative systems and associated station-keeping techniques were viewed by Dunderdale and Wright (2005).

1. **Gravity-based platforms** (e.g., the Hibernia GBS) are typically designed with the ability to withstand the loads from large iceberg impacts and hence have an inherent capacity to withstand pack ice interactions. As a result, ice management is not required to support them, from the standpoint of keeping ice load levels within acceptable levels. However, ice management will, at times, be required to support some of the operations that may be carried out around a GBS platform. For example, ice management would be needed for the safe operation of tanker loading system.
2. **Turret moored FPSOs, DP vessels, and tankers** that are moored at single point loading facilities have a distinct advantage over floating systems anchored with fixed alignments, since they can vane into moving pack ice, thereby presenting a much reduced profile to any ice interactions. In most cases, ice management support will be needed to keep load levels and any other adverse ice effects on these vessels within acceptable levels, at least in heavy pack ice conditions. While station-keeping, vessels fitted with multiple thrusters can use strategically directed propulsion wash to help reduce ice accumulations and some ice impacts and enhance the smooth flow of ice and its clearance around them. These types of capabilities and maneuvers would help to reduce the amount of pack ice that needs to be managed up-drift by support vessels. Also, turret moored and dynamically positioned vessels can move off a location on very short notice, once their production or loading operations have been suspended. However, a turret moored FPSO may require considerable time to reconnect to a released mooring system. Ice capable support vessels can also play a role in escorting such vessels through ice and in helping them reconnect.
3. **Semi-submersible rigs and drillships** are typically anchored with either 8 or 12 anchor lines. They are generally used to drill exploration and production wells and are not “permanent” platforms like a GBS or FPSO. When these types of structures are used for operations in ice, consideration would have to be given to the effectiveness of ice management support in reducing the global ice actions. Ice management would also be required to prevent local ice loads on their hulls, which are often only nominally ice strengthened. Another consideration of particular importance for the operation of a semi-submersible in pack ice is the effect of any ice interaction with the risers which, on most rigs, are not enclosed within a leg but are exposed. If pack ice conditions become threatening and a decision made for a semi-submersible or drillship to vacate location, consideration must be given to the possibility that anchor systems will need to be retrieved in the existing pack ice conditions. In some circumstances, ice management may be required by each of the anchor

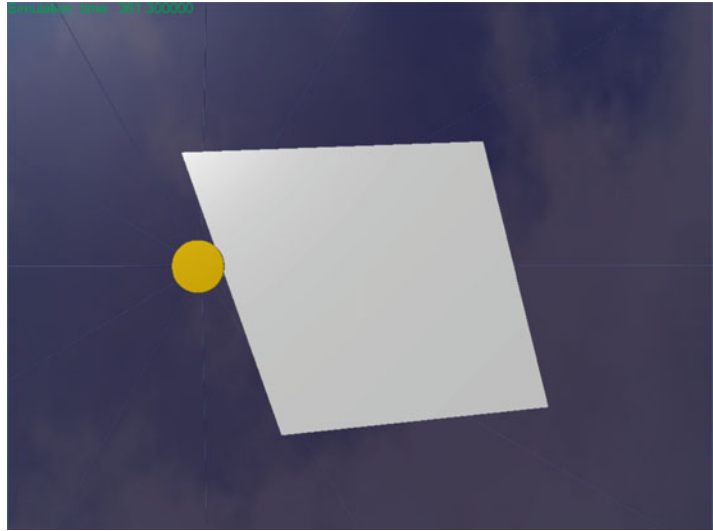
handling vessels during the recovery and return of anchor systems to the rig. As a contingency, all moorings used by anchored vessels operating in ice are fitted with some type of emergency quick release system. Various methods are used to disconnect mooring chains from the rig such as a chain “shear link” or an acoustic release mechanism between the mooring wire and anchor. Issues associated with a “quick rig release” include careful positioning of the mooring where the ends to be dropped stay clear of any sub-sea assets and the considerable amount of time and work necessary to retrieve and restore anchor systems back onto the rig. Once a rig moves off location and is underway, additional consideration must be given to towage. In this regard, ice escort may be necessary to move the rig from the location to open water.

Evaluation of Ice Management Capabilities

1. **Evaluation of sea ice management:** Eik (2011) presented a method for including sea ice management in a structural reliability analysis. The approach focused on how icebreakers or offshore installation disconnection capabilities might alter ice load distributions and thereby reduce the probability of failure. To demonstrate the approach, a moored symmetrical structure (i.e., “Kulluk-type”) in the central Barents Sea was considered. Realistic ice data from a probabilistic ice drift simulator was used as a basis, and ice loads were calculated for a 1000-year period. The load calculations were based on the transformation of all developing ice conditions into a standardized equivalent ice thickness. Empirical formulations for ice resistance on icebreaking vessels were combined with “max-to-mean ratios” from physical tank-model tests to estimate peak loads. The demonstration revealed a need for more relevant ice load models that are valid both for managed and unmanaged ice. In a severe ice environment, it was found that the icebreaker’s ability to reduce ice loads was insufficient to reduce the most extreme loads. The ability to disconnect moorings and risers and escape the site prior to abnormal ice events would ensure that the ice loads were within an acceptable range, conditional that the disconnection capabilities were sufficiently reliable. A fault-tree analysis including failure probabilities in ice management operations and/or disconnection operations was considered useful for estimating the frequency of specified accidents.
2. **Evaluation of iceberg management:** A method for the systematic evaluation of an iceberg management system and its impact on the design of offshore structures was also introduced by Eik and Gudmestad (2010). The approach was based on the combination of a numerical iceberg drift model and a probabilistic analysis with the event tree model. Experiences from the Canadian iceberg detection studies and iceberg deflection operations were also incorporated into the model. For a selected site in the Barents Sea, it was found that the maximum impact load corresponding to a 10,000-year event was 85 MJ for a concept without any iceberg management capabilities. An alternative system with iceberg detection, iceberg deflection, and disconnection capabilities including emergency disconnect indicated a corresponding abnormal load of about 1.8 MJ.
3. **Evaluation of mooring load in managed ice:** Monitoring of ice conditions and measurements of ice forces have provided a comprehensive and unique set of data from the full-scale experience with the Kulluk structure (a conical drilling unit used in the Beaufort Sea). The monitoring included records of ice cover conditions as well as descriptions of ice management operations. The ice forces were measured using the tension values on the mooring lines. Wright (1999) carried out a comprehensive analysis of the observations and measurements, in which the ice loading episodes were summarized in a manner that clearly linked ice forces to ice conditions, ice management activities, and the mode of ice interaction with the structure. Su et al. (2019) carried out a numerical study of the Kulluk structure in managed ice as well as some “extreme events” reported by Wright (1999). The simulation shown in Fig. 2 refers to one of these “extreme events”: a 500 m diameter

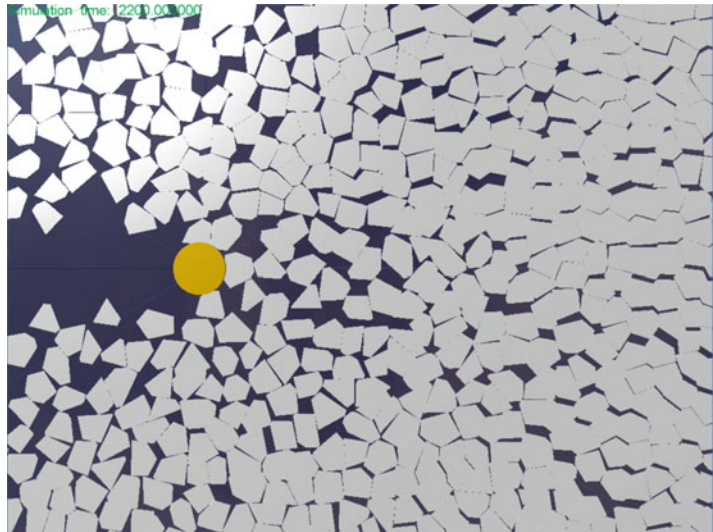
Ice Management in Offshore Operations,

Fig. 2 Numerical simulation of an “extreme event” reported by Wright (1999)



Ice Management in Offshore Operations,

Fig. 3 Numerical simulation of the Kulluk structure in managed ice



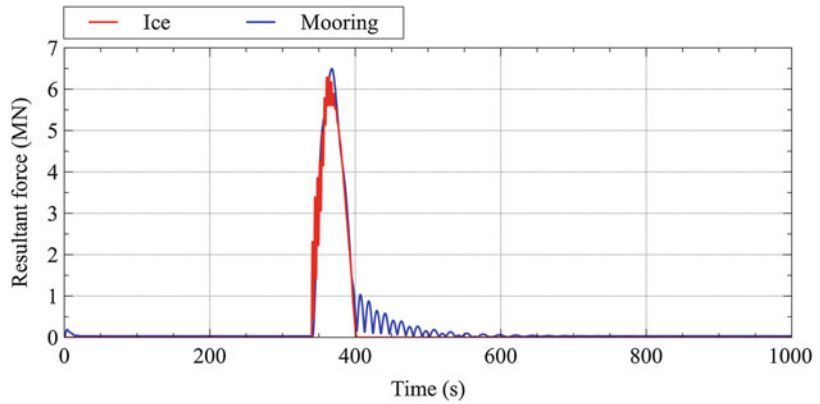
and nearly 2.5-m-thick ice floe impacted the Kulluk structure (with a mooring stiffness of 350 t/m) at 0.3 m/s and produced a peak load of about 600 t (5.88 MN), before it rotated around (Wright 1999). The simulation shown in Fig. 3 refers to a managed ice condition (ice concentration, 90%; ice thickness, 2.5–2.7 m; floe size, 30–50 m; drift speed, 0.3 m/s). The simulated time series of ice and mooring loads are given in Figs. 4 and 5, respectively. These two simulated cases directly show that ice

management can contribute to a significant reduction of ice load level (from about 6 MN to 1 MN), which has also been concluded by assessing the full-scale data (Wright 1999).

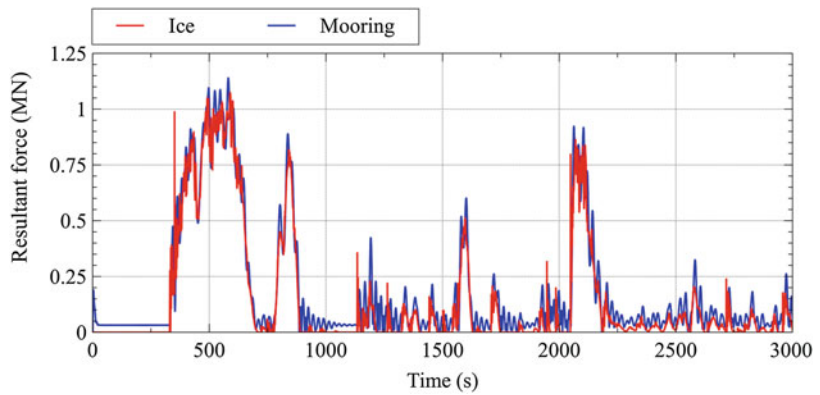
Key Applications

Ice management is most commonly used to support floating systems in both sea ice and glacial ice environments and can significantly influence the

Ice Management in Offshore Operations,
Fig. 4 Simulated ice and mooring loads under the “extreme event”



Ice Management in Offshore Operations,
Fig. 5 Simulated ice and mooring loads in managed ice



design philosophy that is adopted for them. It can also be used to mitigate the risk of deep draught ice features interacting with sea floor facilities. In certain cases, ice management can be used as a means of modifying ice actions on fixed structures, although this approach is not common. It is also a relevant consideration in terms of supporting other in-ice activities, such as Evacuation, Escape and Rescue (EER) systems and tanker offloading operations.

The oil and gas industry have already been involved in different ice management activities for a long time (more than 40 years) and in several geographical areas (Eik 2008):

1. **Beaufort Sea:** Sophisticated ice management systems were developed for the exploratory drilling in the Beaufort Sea, where the conventional drillships (since the 1970s) and a
2. **East coast of Canada – Grand Banks:** The occurrence of drifting icebergs is the dominating threat for the petroleum production on the Grand Banks. During the years with

conical drilling unit – Kulluk (since the 1980s) – were used. During operations the Kulluk structure experienced thick first-year ice, large pressure ridges, heavy rubble, and high concentrations of drifting multi-year ice. The experiences with the Kulluk structure provide the best source of data for most considerations related to moored structure in various pack ice conditions with ice management activities (Wright 1999; Dunderdale and Wright 2005). In 2009, a comprehensive series of ice management trials were conducted in the Fram Strait to evaluate the performance of the icebreakers in mainly second-year ice (Maddock et al. 2011).

- activities on the Grand Banks, experiences from physical iceberg management operations (since the 1970s) have been collected, reported, and presented in a publicly available database (PERD Comprehensive Iceberg Management Database, Rudkin et al. 2005).
3. **West coast of Greenland – Fyllas Banke:** Over a period of 10 weeks in 2000, exploration drilling was carried out at a water depth of 1150 m offshore central west Greenland in the Fylla field. During the drilling, valuable information regarding icebergs as well as practical experience regarding iceberg management was gained (McClintock et al. 2002).
 4. **The Greenland Sea:** In order to evaluate and test various ice management technologies, full-scale trials offshore north-east Greenland were performed in 2012 and 2013 within the framework of the Oden Arctic Technology Research Cruise (OATRC) program (Scibilia et al. 2014).
 5. **The Arctic Ocean:** As a part of the International Ocean Drilling Program (IODP), a multiple vessel expedition was sent into the Arctic Ocean in 2004 to drill and recover deeply buried sediments at the Lomonosov Ridge. Two icebreakers worked up-drift of a drillship in a two-stage ice management configuration, and the drillship was able to remain on station for up to 9 days in 1100–1300 m water using manual dynamic positioning control (Moran et al. 2006). A research cruise (OATRC2015) was performed in the Arctic Ocean in 2015. One of the main objectives of this cruise was to collect full-scale data necessary to build, calibrate, and validate theoretical models for ice management operations (Lubbad et al. 2018).
 6. **Barents Sea:** During a data collection expedition in the northeastern part of the Barents Sea, an attempt to tow an iceberg frozen into sea ice was carried out (Eik 2008).
 7. **Pechora Sea:** In 2000, an offshore oil-loading facility was built at Varandey in the Pechora Sea. At least one icebreaker was used to break and clear the ice around a single loading hose.
 8. **Sakhalin:** In order to produce oil from the Piltun-Astokhskoye field (Sakhalin 2), the Vityaz Production Complex was developed, and ice management operations were required. With respect to new technology, it has been reported by Keinonen (2007) that the introduction of icebreaking support vessels, equipped with azimuth propeller systems, has significantly improved the ice management capabilities.
 9. **North Caspian Sea:** In addition to icebreaker support (Arpiainen et al. 1999), physical ice barriers (e.g., steel piles and caissons) have also been used as passive ice management protection during exploration drilling in the North Caspian Sea.
 10. **Bohai Sea:** In the Bohai Sea, which is a typical area of marginal oil field, most of the offshore oil and gas platforms are in a seasonally frozen region. Specific sea ice management systems have been established based on years of field monitoring and research (Zhang et al. 2017).

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Ice Management System

► Ice Management in Offshore Operations

Ice Tank Test

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Synonyms

[Coastal and River Engineering \(NRC-OCRE\)](#); [Cold Regions Research and Engineering Laboratory \(CRREL\)](#); [International Towing Tank Conference \(ITTC\)](#); [Krylov State Research Center \(KSRC\)](#); [National Research Council Canada-Ocean](#); [National Maritime Research Institute of Japan \(NMRI\)](#); [Maritime Ocean Engineering Research Institute \(KRISO\)](#); [The Hamburg Ship Model Basin \(HSVA\)](#)

Definition

Ice tank test is used to investigate various ice–structure interactions in an ice tank under scaled conditions. An ***ice tank*** is a model basin whose purpose is to provide a physical modeling environment for the interaction of ship, structures, or sea floor with both ice and water.

General

Ice tank test can be used to investigate various ice–structure interactions and offers the advantage that relatively complex problems (that can be difficult to analyze using other methods) can be studied and visualized at a small scale (i.e., smaller than full scale). An ice tank is a model basin or tank whose purpose is to provide a physical modeling environment for the interaction of ship, structures, or sea floor with both ice and water. Ice tanks may take the form of either a towing tank or maneuvering basin.

Generally, ice model tests have been used to investigate global ice actions resulting from

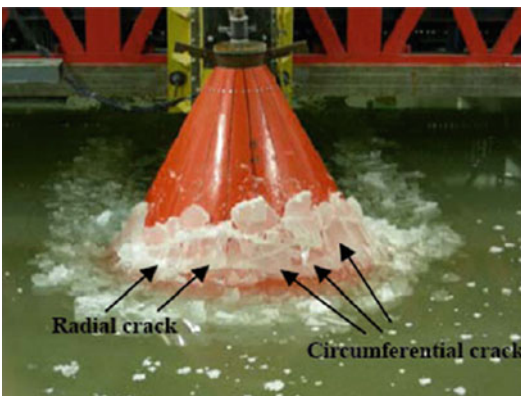
moving ice, such as the expected ice actions and/or the expected ice interaction modes (e.g., ice failure modes, ice ride up, the buildup of ice rubble around a structure, ice clearing behavior around a structure, ice blockage, etc.). Ice tank test is a flexible tool that has been applied to many ice–structure interaction cases such as:

- (a) Various structure types; the types tested include:
 - Large vertical or multifaceted caissons (e.g., Cornett and Timco 1998)
 - Upward breaking and downward breaking cones (e.g., Huang 2010; see Fig. 1)
 - Narrow versus wide structures (e.g., Huang et al. 2017).
 - Multi-leg structures (e.g., Huang et al. 2013; Li et al. 2017; see Fig. 2)
 - Berms
 - Moored versus bottom-founded structures (e.g., Evers and Jochmann 2011; see Fig. 3)
 - Satellite structures placed near a drilling structure to provide ice protection, and access for EER craft or protection for quay areas
- (b) Various ice features interacting with a structure, such as:
 - Sheet ice
 - First-year ridges
 - Multi-year ridges
 - Broken ice, and floe ice conditions

Table 1 lists common ice processes treated by means of physical modeling and indicates the important variables influencing them. Model reliability depends on the accuracy with which the model replicates those variables. Note that some modeling situations can involve a combination of processes. Consequently, the variables can be grouped differently as shown in Table 1 (Zufelt and Ettema 1996).

Only a limited number of comparisons have been made between model scale and full-scale results for structures, although several correlation studies have been done for icebreaking ships (e.g., Wang and Jones 2008). Ice model tests have been conducted at different scales for fixed icebreaking conical structures. Comparisons are made for the Kulluk (a large downward breaking floating conical structure) between model tests conducted at different ice tanks and field data collected while it was on station in the Beaufort Sea (Comfort et al. 1999).

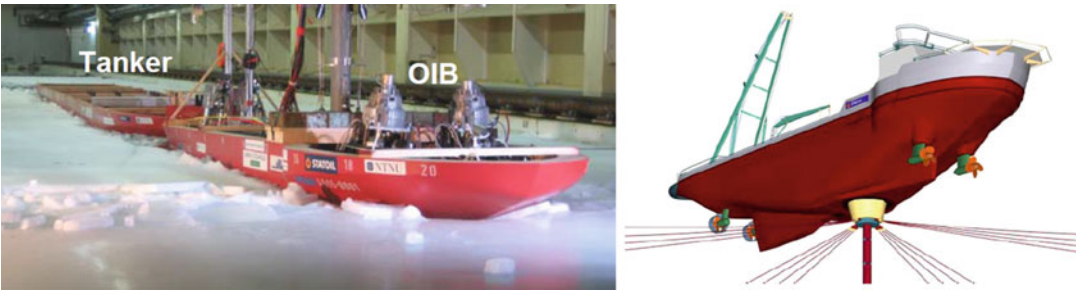
A problem of major concern in planning a model test program is the type of ice. Not all ice problems can be investigated reliably using present-day ice tank modeling techniques. An ice model test program should be targeted to investigate the specific ice interaction problem of interest. Sometimes, it is preferable to test a simplified, basic problem in the ice tank, and then use these results to assess an overall problem that is more complex.



Ice Tank Test, Fig. 1 Model ice failure in an ice-breaking cone test. (Huang 2010)



Ice Tank Test, Fig. 2 Model ice failure in front of a multi-leg structure. (Li et al. 2017)



Ice Tank Test, Fig. 3 Moored tanker in ice tank with underwater illustration. (Evers and Jochmann 2011)

Ice Tank Test, Table 1 Ice processes and variables that it is important to model

Ice process	Variables of importance
Ice sheet deflection	Ice flexural strength
	Modulus of elasticity
	Loading rate
	Ice thickness
Ice ridging	Velocity of parent ice sheet
	Ice thickness of parent ice sheet
	Ice–ice friction
	Ice sheet strength (flexural, crushing, and shear)
	Keel depth
	Sail height
	Consolidated layer
Ice–structure interaction	Structure stiffness
	Dynamic characteristics of structure/mooring system
	Velocity of ice sheet
	Ice thickness
	Ice sheet strength (crushing, flexural, and shear)
	Ice–ice and ice–structure friction
	Mode of breaking (crushing, flexure, and shear)
	Ridge properties and dimensions
Icebreaker modeling	Vessel speed
	Ice–ice and ice–hull friction
	Ice thickness
	Ice sheet strength (crushing, flexural, and shear)
	Ice piece size
	Piece movement around hull and propeller
	Mode of breaking (crushing, flexure, and shear)
	Ridge properties and dimensions

Previous investigations of ice ride up and ridge building are examples where this approach has been used. Initially, simple model tests were

done to investigate ice ride up and ridge building processes. These basic data are part of the information set that is used to address more complex issues such as

- Designing islands and structures to avoid ice ride up problems
- Quantifying pack ice driving actions and developing satellite structures designed to form protective rubble around drilling structures in the Beaufort Sea

Scaling uncertainties and modeling artifacts are another important issue that should be considered in planning an ice model test program (von Bock und Polach and Molyneux 2017). These can arise from imperfections in the ice modeling material being used. Some model ices are better able to simulate some types of ice interactions than others.

Scaling

When model testing is used for ice–structure interactions, appropriate scaling relationships should be selected to represent the mechanisms or processes that dominate the ice actions or response to ice actions. In the physical modeling of ice and structure interactions, Froude similarity (the ratio of inertial to gravitational load) and Cauchy similarity (the ratio of inertial to elastic forces) are maintained between the prototype and the model (Timco 1984). Considerable effort has been expended on the development of model ice with mechanical properties scaled to prototype ice and

with analogous failure behavior. There can also be rigorous modeling of the structure shape, stiffness, and surface characteristics. As long as the failure modes expected in the prototype are correctly simulated, model tests can provide an important input into the design process. Model ice can provide optimum simulation of ice interactions only over a certain range of scale factors, generally within about 10 to 50. Tests done at scale factors that are too high are likely to produce results that are subject to modeling distortions.

Scaling is accomplished by observing the Froude scaling law, as given in Eq. (1), and the Cauchy scaling law, as given in Eq. (2):

$$Fr = V / \sqrt{gL} \quad (1)$$

$$Ca = \rho_w V^2 / E \quad (2)$$

where:

Fr is the Froude number, the ratio of inertial to gravitational forces.

Ca is the Cauchy number, the ratio of inertial to elastic forces

V is the speed, expressed in meters per second

g is the acceleration of gravity, expressed in meters per second squared

L is the length dimension, expressed in meters

ρ_w is the density of water, expressed in kilograms per cubic meter

E is the modulus of elasticity of ice (Young's modulus), expressed in megapascals

Since inertial and gravitational forces dominate free-surface flow, the Froude number is generally used as the similitude criterion for ice basin modeling. The Cauchy number, then, provides the necessary scaling for material parameters. It is not possible in model tests to simultaneously satisfy the requirements of Reynolds law, which governs viscous effects, with those of Froude/Cauchy scaling. Since viscous effects are relatively small for ice model tests at low speeds, Reynolds law is usually ignored.

In order to satisfy the Froude and Cauchy laws, the various geometrical and physical quantities of the tested objects should be scaled according to Table 2.

Ice Tank Test, Table 2 Scale relations

Physical values	Model scale	Full scale
Ice thickness	h'	$h = \lambda h'$
Flexural strength of ice	σ_f'	$\sigma_f = \lambda \sigma_f'$
Elastic modulus of ice	E'	$E = \lambda E'$
Density of water	ρ_w'	$\rho_w = \rho_w'$
Density of ice	ρ_i'	$\rho_i = \rho_i'$
Time	t'	$t = \lambda^{1/2} t'$
Velocity	V'	$V = \lambda^{1/2} V'$
Force	F'	$F = \lambda^3 F'$
Friction coefficient	μ'	$\mu = \mu'$

NOTE: λ is the geometric scale factor

Test Methods

Model Ice Types

Natural sea ice behaves as a brittle or ductile material depending primarily on its temperature, salinity, and strain rate. Basic mechanical properties of the natural ice, such as strength and deformation, also depend mainly on the strain rate, ice temperature, and salinity content in the ice. Flexural strength is conventionally considered independent from strain rate (Timco and Weeks 2010). Ideally, model ice should behave in the same manner when applying Froude scaling law, i.e., the strengths should be related by a linear scale factor, λ .

In the past decades, significant effort was made in finding the ideal material for model ice. Different approaches used to produce model ice include synthetic ice, hybrid ice, and refrigerated ice. Currently, refrigerated ice is the most widely used model ice. To scale the ice properties correctly, the ice is doped by sodium chloride or other chemical dopants, such as ethylene glycol and detergent. A wet seeding technique can be used to produce a fine, crystalline, columnar structured ice (see, e.g., Fig. 4). A fine-grained ice can be produced by spraying a saline solution during the entire ice growth process, resulting in a granular structured ice similar to snow ice (see, e.g., Fig. 5). Further model ice development includes the injection into the water of air bubbles that are entrained into the growing ice crystals to increase brittleness and decrease ice density.



Ice Tank Test, Fig. 4 Columnar structured model ice. (Sun and Huang 2020)



Ice Tank Test, Fig. 5 Granular structured model ice. (von bock und Polach 2016)

It is important to note that a standard “model ice” has not yet been established, and judgment is required to ensure that the model ice used is appropriate for the ice–structure interaction of interest, in combination with other factors such as the size and capabilities of the ice tank.

Testing Technique

In general, two testing options are used in ice model tanks. The first method is to push or pull the model through a stationary ice sheet; the second is to have a moving ice sheet pushed against a fixed or moored model. The selected option should be consistent with the test

objectives. Parameters measured and the configuration of the measurement systems should be primarily dictated by the test objectives and the characteristics of the structure and its operational conditions.

Differences in the dimensions and layout of the ice tank facility can have some influence on the tests. An important factor is the effect of the basin size (length and width) on the number of tests that can be carried out with one ice sheet. The required actual test length depends on the type of test. As an example, for ice rubble formation tests or ice pileup tests, the model should be able to proceed until a steady state is reached. The other factor that can restrict the test program is the proximity of the basin walls. For the case when a model is pulled through ice, the level ice sheet can be considered to have infinite extent if the shortest distance from the point of application of any loads to the nearest tank wall is more than three times the characteristic length of the ice sheet (ITTC 2017).

The choice of testing technique depends strongly on the type of test. In particular, it should be appreciated that when pushing the ice sheet against a model of a fixed or moored structure, unrealistic cracks can often propagate from the model towards the sidewalls of the tank. In addition, the ice sheet can fail along the pushing plate instead of on the model; this happens especially for thin and brittle ice. As well as for the testing technique option, the most practical way to minimize negative consequences of the boundary condition deviation is to conduct tests with an ice sheet sufficiently wide relative to the model width.

Ice Conditions

Model tests can be performed in ice conditions such as level ice, rafted ice, a newly broken channel, brash ice (old channel), broken ice, rubble ice fields, and ridges. The proper documentation of the tests should include the method by which the ice conditions are simulated. It is recommended to document the ice conditions prior to and after the test by still photographs and video records, preferably from some elevation. The number, time, and location of ice thickness and property measurements should also be documented. Visible cracks in the ice extending from the broken

channel towards the basin walls or towards open water areas should be included in the report.

The thickness of the model ice should be measured after each level ice test and the number of measurements should be sufficient to adequately evaluate variations in ice thickness along the track of the model through the ice. In cases where test runs are performed close to the sidewalls of the ice tank, measurements should be taken on both sides of the broken channel.

In an old channel or rubble field, the thickness should be measured along two or more lines within the model track. Underwater sonar for scanning the lower surface of model ice and laser profiles for evaluation of ice upper surface or similar remote sensing devices is particularly advantageous for tests with ice rubbles/ridges or brash ice. Such measuring systems allow for the nondestructive acquisition of the ice thickness profiles or a 3D picture before running the test.

Ice ridges and rubble can contribute, in many cases, to the design actions on a structure, and conducting model tests for such conditions is of particular importance. Producing model ridges and rubble in compliance with the requirements of similitude theory is one of the most challenging tasks in ice–structure interaction modeling. The conventional method of creating a first-year ice ridge/rubble is to grow level ice with a thickness equal to the thickness of ice blocks constituting a model ridge/rubble, break this ice sheet into irregular pieces, and then collect these pieces into a pile. The resulting pile of ice is normally left exposed for some period of time to negative ambient air temperature in order to create a consolidated layer of the ridge/rubble.

In the course of ice ridge/rubble modeling, particular attention should be paid to modeling the effects of the interaction within the nonconsolidated part of the ice feature and to the mechanical properties of its refrozen part. The mechanical properties of the consolidated layer can be evaluated in the same way as for level ice, assuming that both sail and keel are carefully removed. For the measurements of the mechanical

properties of the nonconsolidated part, pull-up, punch, and/or shear tests can be employed.

Model Ice Properties

General

Different model testing facilities use different types of model ice materials. None of the existing model ice materials is absolutely perfect. Thus, ice properties measured for one type of material cannot be directly compared with another material. The geometrical parameters and mechanical properties of the model ice should be determined accurately, regardless of the material composition.

To maintain reliable results, it is recommended to perform property measurements in situ in the tank and, whenever possible, without lifting the samples out of the natural environment. The timing and location of the measurements are important. The measurements should be done as close as possible to the actual test area and test time. All measurement procedures should be as simple as possible and measure the desired parameter directly. The test procedures and measurements should be performed by qualified personnel and should be clearly documented. The equipment used in all measurements should be calibrated at the ambient temperatures of the ice tank.

Properties of first-year and multi-year ridges are very difficult to control in ice basins. Samples should be taken from parts of the ridge to document both consolidation and other internal properties. The ice conditions adjacent to the ridge should be documented. For instance, broken level ice surrounding a ridge reduces the actions substantially compared to intact level ice.

Flexural Strength and Modulus of Elasticity

The flexural strength and the modulus of elasticity are frequently determined by cantilever beam tests. A floating cantilever beam having length L and width B is cut in situ. The recommended dimensions of a beam ($L \times B \times h$) are $6h \times 2h \times h$, where h is the ice thickness. The tip of the beam is

loaded at a constant speed until the beam fails. It is recommended to use an electrically driven loading device. The applied load as well as the deflection of the beam should be recorded. The determination of the deflection is required only for the determination of the elastic modulus. Strength measurements should be carried out several times during the tempering phase and once after the ice test has been performed. Whenever possible, the strength should be measured at different locations distributed over the tank length. The flexural strength and the modulus of elasticity in bending can be calculated from the failure load and the free-end deflection measurements as given in Eq. (3) and (4), all in consistent SI units:

$$\sigma_f = \frac{6F_b L}{Bh^2} \quad (3)$$

$$E_f = \frac{4FL^3}{Bh^3\Delta} \quad (4)$$

where:

σ_f is the flexural strength in bending

E_f is the elastic modulus in bending

F is the load applied to the cantilever free end

F_b is the failure load

L is the length of the beam (load point to support)

B is the width of the beam (at the support)

h is the thickness of the ice

Δ is the deflection of the beam (cantilever free end)

An alternative method to measure the modulus of elasticity in bending is to measure the deflection of an infinite floating ice sheet under a given load. In this case, the elastic modulus is defined as given in Eq. (5):

$$E_f = \frac{3(1 - \nu^2)}{16\rho_w g h^3} \left(\frac{F}{\Delta}\right)^2 \quad (5)$$

where

ν is the Poisson ratio for ice

ρ_w is water density

g is acceleration due to gravity

Other designations are as for Eq. (4).

Compressive Ice Strength

The compressive strength of model ice can be measured using any of the following three methods:

- Uniaxial test is performed in the water
- Ice block is cut and lifted from the water to perform a uniaxial test
- Indenter is used to load the ice sheet horizontally

The compressive strength is the most difficult to measure with high accuracy and the structure of the model ice does not always allow the use of all of the above methods.

Friction Coefficient

The dynamic friction coefficient is determined through measurements of the tangential action when either a piece of model ice is slid along a plate, painted identically to the model structure, or such a plate slides over a piece of model ice. In both cases, the ice and the plate should be pressed together by a given normal action. The friction between two ice pieces can be determined using the same testing arrangements. The contact surfaces of the ice and the plate should be moistened, since wet and dry friction coefficients differ significantly. The friction tests should be performed at a speed corresponding to the ice–structure interaction velocity expected in the tests. The friction coefficient, μ , is calculated according to Coulomb's friction law as given in Eq. (6):

$$\mu = F_T / F_N \quad (6)$$

where

F_T is the mean measured tangential (friction) load during steady motion

F_N is the normal load on the contact surface

List of Ice Tanks (Table 3)

Ice Tank Test, Table 3 List of ice tanks

Ice tank facility	Location	Dimensions	Grain structure	Chemical composition
Aalto University	Finland	40 m × 40 m × 2.8 m	Fine grained	Ethanol
Aker Arctic Technology Inc	Finland	75 m × 8 m × 2.1 m	Fine grained	Sodium chloride
Krylov State Research Center (KSRC)	Russia	80 m × 10 m × 2 m	Columnar and fine grained	Sodium chloride
National Research Council Canada, Ocean, Coastal and River Engineering (NRC-OCRE)	Canada	76 m × 12 m × 3 m	Columnar	EG/AD/S
Cold Regions Research and Engineering Laboratory (CRREL)	USA	37 m × 8 m × 2.4 m	Columnar	Urea
Maritime Ocean Engineering Research Institute (KRISO)	Korea	32 m × 32 m × 2.5 m	Columnar	EG/AD
National Maritime Research Institute of Japan (NMRI)	Japan	35 m × 6 m × 1.8 m	Columnar	Polypropylene glycol
The Hamburg Ship Model Basin (HSVA)	Germany	78 m × 10 m × 2.5 m	Columnar	Sodium chloride
Tianjin University	China	40 m × 6 m × 2 m	Columnar	Urea

Note: *EG* ethylene glycol, *AD* aliphatic detergent, *S* sugar

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Iceberg

- ▶ [Iceberg Scouring](#)
- ▶ [Ship-Iceberg Interactions](#)

Iceberg Management

- ▶ [Ice Management in Offshore Operations](#)

Iceberg Scouring

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Synonyms

[Burry depth](#); [Gouging](#); [Iceberg](#); [Scouring](#); [Subsea pipeline](#)

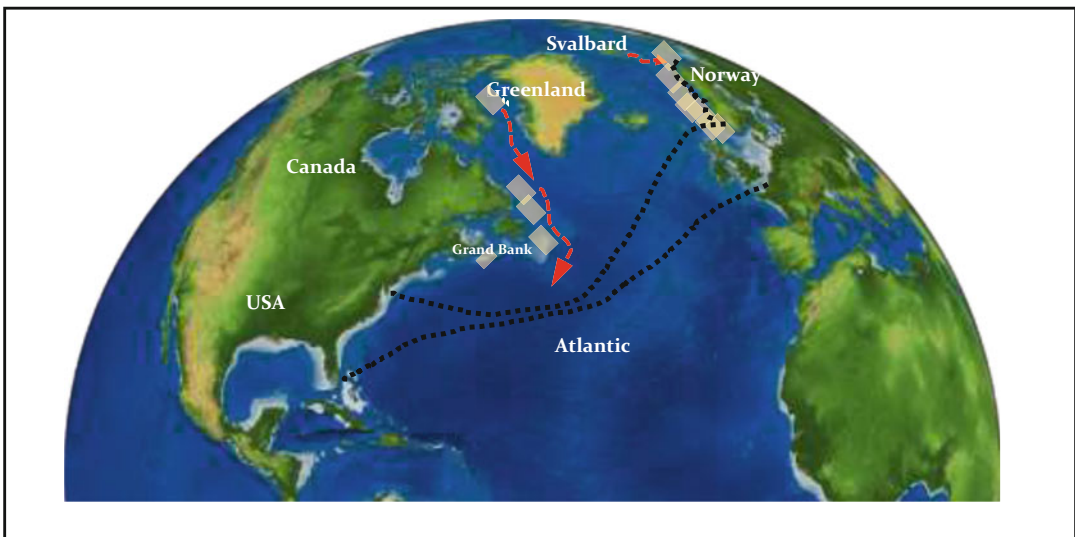
Definition

Iceberg scouring (also called *iceberg gouging*) is a phenomenon in which the bottom of an iceberg interacts with the seabed. It happens normally in shallow water areas due to the limited depth of icebergs. The drifting force of iceberg is the main driving factor in such interaction. If there are subsea pipelines or other structures installed in the same area, the *iceberg scouring* may present a potential hazard to the integrity of the subsea pipelines or structures. **Burry depth (BD)** is used to describe how deep a pipeline shall be buried against the iceberg scouring loads. To trench the seabed and burry the pipeline to a certain depth (*so-called burry depth*) is one way to lower the risk, but it significantly increases the project cost.

Scientific Fundamentals

Iceberg Drifting

Icebergs originated from the source glaciers are drifting on the sea surface under the combination of wind, gravity, wave, and current. The largest source of iceberg is the glacier located at Greenland (Denmark), see Fig. 1.



Iceberg Scouring, Fig. 1 Iceberg falling off from Greenland (Denmark) and Svalbard (Norway), drifting to shipping lanes and oil and gas fields in North Atlantic/Barents Sea (for illustration only)

The basic equation for the horizontal motion of an iceberg is as following (Bigg et al. 1997):

$$M \frac{d\mathbf{V}}{dt} = -Mf\mathbf{k} \times \mathbf{V} + \mathbf{F}_a + \mathbf{F}_w + \mathbf{F}_r + \mathbf{F}_s + \mathbf{F}_p \quad (1)$$

where M is the iceberg mass, $\mathbf{V}$ is the iceberg velocity, f is the Coriolis parameter, $\mathbf{k}$ is the unit vector directed vertical upwards, $\mathbf{F}_a$ is the air form drag, $\mathbf{F}_w$ is the water drag, $\mathbf{F}_r$ is the wave radiation force, $\mathbf{F}_s$ is the sea-ice drag (where applicable), and $\mathbf{F}_p$ is the horizontal pressure gradient force exerted by the water on the volume that the iceberg displaces. Besides the horizontal motion, the iceberg has potential off-horizontal plane rotation. The most stable iceberg is tabular and blocky iceberg. The domed, wedged, and pinnacled iceberg is not stable. They may roll completely over in seconds without any apparent reason; see Fig. 2 for a drifting pinnacled iceberg.

In the meanwhile, iceberg's mass and shape will be changed due to its melting and thus influence the drifting model. Løset (1993) presented a detailed numerical model simulating the iceberg melting. A rough estimation might be good

enough to be included in the drifting model considering the large amount of iceberg numbers and the long simulation time.

Iceberg Scouring

As shown in Fig. 3, the bottom of iceberg may have the chance to get contact with the seabed. The contact/interaction will continue until the balance of the driving force and the resistance from seabed soil. The scouring depth can reach 4 m or more. According to Palmer and Croasdale (2013), the iceberg scouring force is much higher than the anchor impact (approx. 1 MN or more). It could be imagined that if the pipeline is subjected iceberg scouring, significant damage might be obtained. Thus, it is a challenge to design a pipeline system in which the iceberg scouring is a potential threat. To rock dump or to trench the pipeline are two alternatives; however, both of them are not economical optimized solutions, especially considering the difficulties on predicting the necessary rock dump height or trench depth.

As shown in Fig. 1, the main potential iceberg risk areas are located at the eastern edge of the Grand Banks of Newfoundland (also called iceberg valley, the iceberg sank Titanic in 1912 came



Iceberg Scouring, Fig. 2 Highly eroded iceberg in front of a towing ship, photo provided by Mark

Iceberg Scouring,

Fig. 3 An illustration of iceberg scouring, with courtesy of SURF, Aker Solutions AS

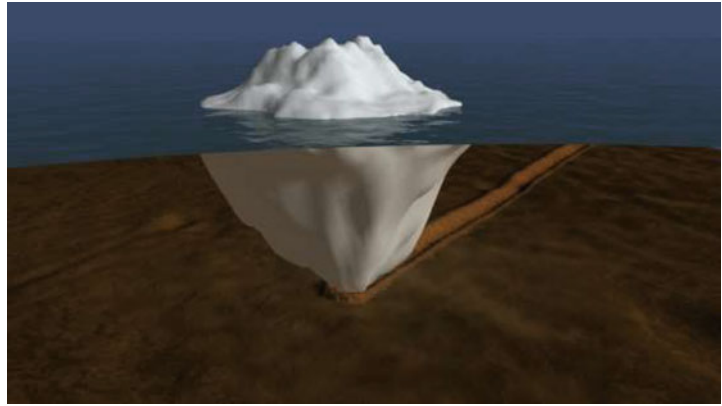
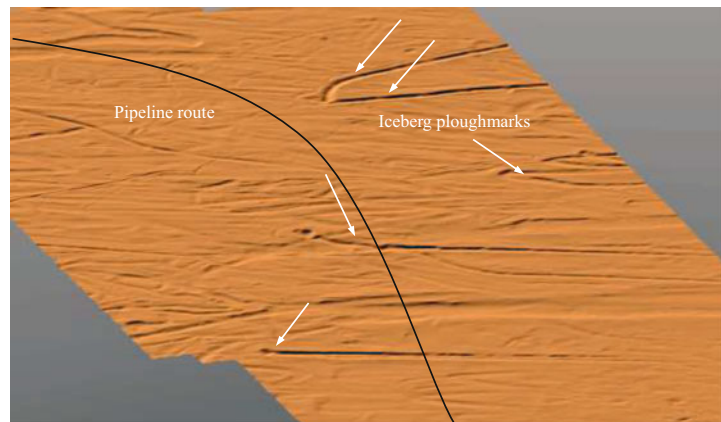
**Iceberg Scouring,**

Fig. 4 Iceberg ploughmarks (where arrow points) from previous glaciation at Snøhvit oil and gas field of Norway, photo rendered by Thomas Hauglien at SURF, Aker Solutions AS



from here) and the Barents Sea in Arctic (North to Norway). The offshore subsea facilities located on the iceberg drifting routes are subjected to the iceberg scouring problem. An example of this is shown in Fig. 4, which is from the seabed survey results of Snøhvit gas field. It lies in the Barents Sea, about 140 km northwest of Hammerfest in northern Norway. The fields were discovered in 1984 in 250–345 m of water and extend across seven production licenses. The iceberg ploughmarks are clearly seen from the survey, even though these marks are probably from previous glaciation age. However, it gives an intuitive feeling about the risk of iceberg scouring to the subsea pipelines.

Subsea Pipeline Design

Subsea pipeline is one of the main components of a subsea production system, see Fig. 5. It connects

production templates and transports the content (oil or gas) to floaters or onshore facilities. Following aspects shall be considered during the design process of a subsea pipeline system:

- Wall thickness
- Pipeline route selection
- Bottom roughness
- On-bottom stability
- Free spanning
- Expansion and buckling
- Tie-in system
- Cathodic protection
- Seabed intervention

The load and resistance factor design format (LRFD) is the preferred pipeline design comparing to traditional allowable stress design, which is depending on the safety class level

Iceberg Scouring,

Fig. 5 Subsea production system, with courtesy of SURF, Aker Solutions AS



(DNVGL 2017b). For other structural components, the allowable stress design is also used, such as the bend, valve, connector, etc. The principle of the LRFD method is to ensure that the design loads do not exceed design resistances for any of the considered failure modes in any load scenarios.

The pipeline integrity shall be checked against the iceberg scouring if the hazard does exist in the lifetime of pipeline in question. On the other hand, a risk assessment for the iceberg scouring is recommended, in which the probability for the pipeline exposed to iceberg scouring is considered. The combined risk is the product of exposure probability and the corresponding consequence. Comparing to the LRFD method, the risk assessment may give more detailed information about the iceberg scouring loads. An acceptable risk level has to be defined. The acceptable risk level can only be obtained with the approval from the oil and gas field operator. Thorough seabed mapping and detailed iceberg drifting data are required for making a reasonable risk level.

Seabed Types

There are quite a lot of ways to category the seabed soil types, such as ASTM D2487-17, 2017; BS 5930:2015 2015; and ISO 14688-2:2017 2017. Typical soil types are listed as follows:

- Clay
- Silt
- Sand
- Gravel

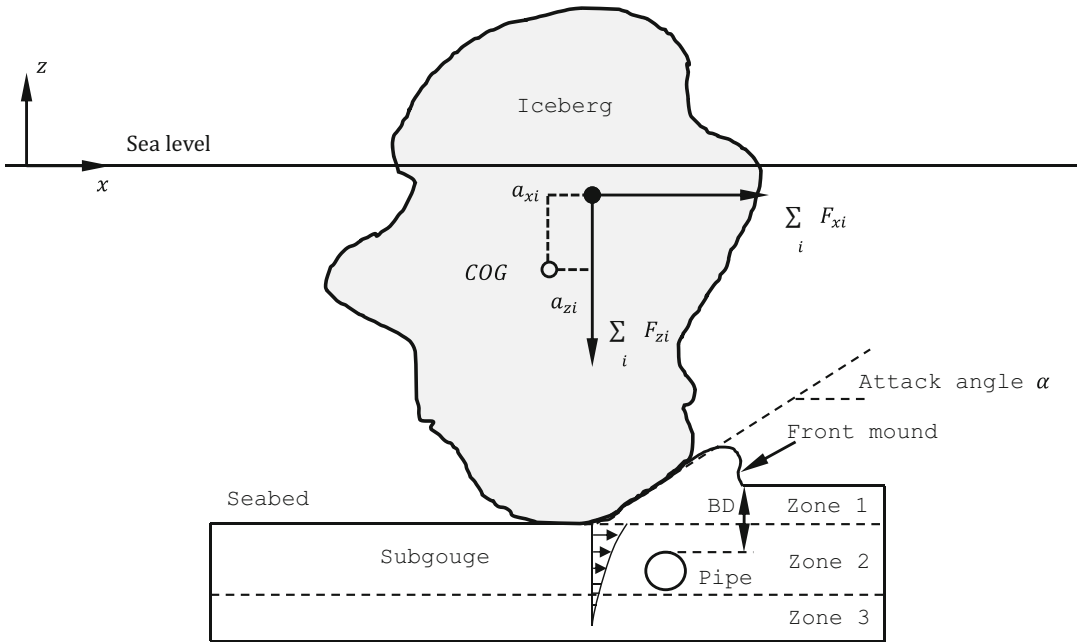
- Pebble
- Cobble
- Boulder

Among these types listed above, “sand,” “gravel,” “pebble,” “cobble,” and “shoulder” may behave in a similar fashion, while “clay” behaves quite differently. The property of “silt” depends on the particle size distribution. It can behave like “sand” or “clay.” Two main types of seabed, “sand” and “clay” are widely used for subsea engineering, see DNVGL (2017a). The subcategories for “sand” are “loose,” “medium,” and “dense.” The subcategories for “clay” are “very soft,” “soft,” “firm,” “stiff,” “very stiff,” “hard.” The simplified stiffness and the shear strength of each type are correspondingly defined. The soil types and stiffness are normally documented in geotechnical report of a subsea field development project.

The seabed soil in the Grand Banks offshore area is mainly composed of cohesionless soil, while the soil from Beaufort Sea is normally a cohesive soil. Nevertheless, the saturation of soil shall be considered (Barrette 2011).

Iceberg Scouring Mechanism

As mentioned earlier, the iceberg scouring becomes only interesting if it presents a hazard to the integrity of subsea structures, such as pipeline, protection structure, or template, etc. Consequently, the interaction involves at



Iceberg Scouring, Fig. 6 Iceberg scouring mechanism

least three major components: the iceberg, the seabed soil, and the subsea structure. Error! Reference source not found. Illustrates an ice keel scouring along the seabed soil and is about to in contact with the backfill of a subsea pipeline. The force generated by iceberg gouging the seabed could be significantly high. Palmer and Croasdale (2013) estimated that the force may reach 32 MN to cut a gouging 4 m deep and 40 m longer for a typical medium clay soil (shear strength 50 kPa). The force magnitude is much larger than the other design force. In order to protect the flowline, two methods are available. The first method is to trench seabed and bury pipeline with rocks (as shown in Fig. 6), and the other method is to shield the pipeline with protection structure, which could be used to absorb the impact energy from iceberg. Both methods are very expensive.

For the iceberg cutting the seabed soil, it is under combined loads from environmental loads and contact force either from seabed soil or subsea structure, see illustration in Fig. 6, the rigid body motion equations in 2D could be expressed as follows:

$$(1 + A_{xx})M\dot{V}_x + B_{xx}V_x = \sum_i F_{xi} \quad (2)$$

$$(1 + A_{zz})M\dot{V}_z + B_{zz}V_z = \sum_i F_{zi} \quad (3)$$

$$(1 + A_{xz})M\dot{\omega}_y + B_{yy}\omega_y = \sum_i F_{xi}a_{xi} + \sum_i F_{zi}a_{zi} \quad (4)$$

where A_{xx} , A_{zz} , A_{xz} are added mass factors on xx, zz, and xz direction, respectively. B_{xx} , B_{zz} , B_{xz} are damping factors on xx, zz, and xz direction, respectively. V_x , V_z , ω_y are iceberg velocity on x, z directions and rotation along y direction. F_{xi} are external forces on x direction which could due to current, wave, air, contact force or drag force from other ice features, etc. F_{zi} are external forces on z direction which could due to buoyancy, gravity, and contact force, etc. The soil can normally sustain very large strains and soil deformation, especially at the front mound (zone 1, the uppermost zone), as seen from Fig. 6. The calculation of the contact force between iceberg bottom and soil is

not straightforward, and the important parameters for calculating the contact force are the attack angle, the shear strength of the ice bottom, and soil. Croasdale et al. (2005) presented an analytical method by using the lower bound plasticity theory to calculate the contact force (vertical and horizontal). Mohr-Coulomb failure criterion is adopted for soil mass. Except for the analytical solution, reliable numerical simulation may give more detailed results for iceberg scouring (see next section).

The extent of soil deformation in Zone 2, the intermediate zone, is still uncertain. Further, the existence of subsea pipeline makes the problem more complicated, see Fig. 6. For the buried pipeline case, the bury depth (BD) is the most important parameter.

Design of BD by RMRS

As mentioned in previous section, the design bury depth (BD) is crucial. A large value of BD will definitely increase the cost of the project, for example, if the pipeline is buried in Zone 3, it may experience negligible soil deformation; however, the large BD makes it is not achievable at all. Pipeline in nowadays is often designed to be placed in Zone 2, where subgouging deformation exists. An optimized BD value can only be obtained by considering the soil and ice deformation into account, which is indeed challenging.

The rule by Russian Maritime Register of Shipping presents a thorough discussion of the subsea pipeline design under iceberg scouring loads (RMRS 2012). It requires that the pipeline is to be buried if there is evidence of ice gouging. Minimum BD must be set a distance of 1 m below the gouge depth. Safety factor (from 1.0 to 1.3) shall be multiplied with this distance, see Eq. 5.

$$B_D \geq G_D + Dk_0 \quad (5)$$

where B_D is bury depth, G_D is the design gouge depth, and D is a given “margin” (equal to 1 m), which can be decreased upon adequate justification. k_0 is a safety factor (from 1.0 to 1.3) based on the pipeline classification in the code (RMRS 2012). The value of design gouge depth could be approached from following methods:

- From seabed gouge data
- From statistical analyses
- From bathymetry and ice regime

Numerical Simulation of Iceberg Scouring

The numerical simulation of iceberg scouring is complicated due to following facts:

- The fluid structure interaction
- The soil and ice interaction
- The soil and structure interaction
- The large deformation of soil and ice
- Validation

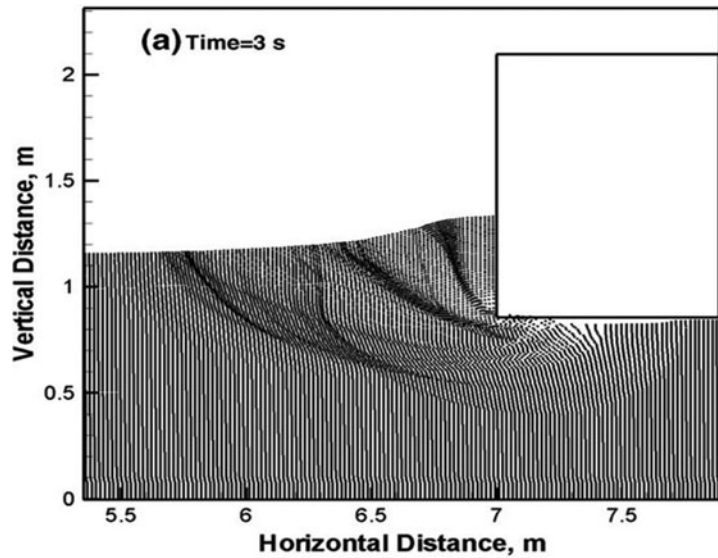
The traditional Lagrange elements are not suitable for simulating the iceberg scouring problem. The main reason is that large deformation of soil is expected in the front mound area. Konuk and Gracie (2004) used so-called arbitrary Lagrangian-Eulerian (ALE) method to simulate the iceberg scouring, the buried pipeline was included in the model. A CAP plasticity constitute model was used to model the soil material. Nobahar et al. (2007) approached the iceberg scouring simulation by using the Lagrangian adaptive meshing technique which overcomes the large element distortion problem. A comparison with the conventional Winkler-type structural analysis was performed. Sayed and Timco (2009) presented a numerical model by using the particle-in-cell (PIC) advection scheme. It was a 2D model in which the iceberg was idealized as rigid indenter, see Fig. 7. Fadaifard and Tassoulas (2014) presented a new numerical model for the coupled seabed scour problem and its interaction with buried pipelines. The seabed was modeled as a viscous non-Newtonian fluid. The interface is achieved by level-set function. Large deformations are allowed. In all the numerical simulations mentioned above, the iceberg is assumed to be rigid, no failure of iceberg ice is allowed. This may produce over conservative scouring loads.

Mitigation Methods

If the iceberg scouring is indeed a design load, following mitigation methods for lowering the required bury depth can be used. In practical, all

Iceberg Scouring,

Fig. 7 Numerical scouring model by Sayed and Timco (2009), with permission of Elsevier



the mitigation methods shall be evaluated together with the economic cost.

following reports/studies found after the initial proposal.

- **Iceberg management**

The idea of iceberg management is to move the potential iceberg scouring risk away from the pipeline route. The monitoring of iceberg tracks and offshore towing vessels are needed.

- **Strong barriers besides pipeline**

The strong barriers can either be rock embankments or sheet piles. These barriers are used to stop the iceberg moving towards the pipeline and thus get rid of scouring. The barriers shall be strong enough to resist the ice features even if the icebergs may be accumulated over time.

- **Weak layer concept**

Palmer and Croasdale (2013) mentioned the so-called “weak layer” concept, in which a specially made layer is laid above the buried pipeline. It will deform significantly if an iceberg across the pipeline, but it will not transfer much shear stress downward. Thus, the pipeline is protected.

- **Cheese wire concept**

Din and Hafezparast (2015) proposed a “cheese wire” concept for protecting the pipeline against the iceberg scouring. It is designed to cut the iceberg bottom by an installed wire system besides the pipeline. However, no

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Icebreaker

► Ice Breaking Vessel

Ice-Induced Vibration and Noise of Ships

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Synonyms

Vibration and noise of icebreakers; Vibration and noise of ships under ice collision

Ice-Induced Vibration and Noise of Ships,
Fig. 1 The large-scale oil drilling caisson platform Molikpaq (<https://www.auseenco.com/en/molikpaq-spacer-and-modifications>)

Ice-Induced Vibration

Definition and Influence of Ice-Induced Vibration

When a ship sails through the ice area, it will inevitably collide with the ice-water mixture. The alternating load generated by crushing the moving ice and its impact on the structure causes the hull vibration, which is called ice-induced vibration (Huang 2004). For a long time, ice-induced vibration was considered a unique phenomenon of slender structures. However, in 1986, the large-scale oil drilling caisson platform Molikpaq was almost destroyed by ice-induced vibration in the Beaufort Sea in 1986, making the dynamic effect of ice on the structure attracting the attention of researchers (Fig. 1).

Ice load is the main environmental load that affects the safe operation of marine structures in ice areas. The ice-induced vibration caused by ice load inevitably causes damage to the hull structure. At first, continuous ice-induced vibration aggravates the fatigue damage to the structure and seriously affects the safety of the hull structure. Secondly, severe ice-induced vibration damages the internal equipment of the hull, such as breaking the upper and lower pipelines and loosening up the flanges. Moreover, it will also influence the comfort and health of the staff.

Research Methods of Ice-Induced Vibration

At present, the ice-induced vibration of the structure is mainly studied through field test,



indoor experiment, and numerical simulation (Wang 2019).

Field Test

The scope of ice-induced vibration research depends on the collection and analysis of field data to a great extent.

In the winter of 1971–1972, the Norstromsgrend lighthouse was observed to undergo violent vibrations after the interaction of sea ice (Eivind 2014). The staff carried out monitoring work on the vibration of the lighthouse. The maximum vibration acceleration generated by the lighthouse was 0.1 g, and the frequency was about 2.8 Hz. In 1998, Jefferies and Wrigh published an article based on the ice vibration phenomenon of the Molikpaq platform. The article stated that the ice-induced vibration caused the fatigue of the sand core and reduced the structure's ability to withstand ice loads. After that, marine structures deployment in ice areas usually needs to be equipped with related monitoring systems. For example, the Bohai JZ20-2 MUQ cone jacket platform is equipped with a complete ice-induced vibration three-dimensional monitoring system. The monitoring system mainly includes sea ice image monitoring, radar image monitoring, load plate and vibration acceleration sensors, etc. It monitors the sea ice parameters such as ice speed, ice thickness, ice direction, sea ice density, and structural ice load and vibration acceleration for long time which provides a large amount of field measured data for ice-induced vibration research.

Indoor Model Experiments

Indoor model test is also an important method for studying structural ice-induced vibration. As compared to the field tests, during indoor experiments, there is flexibility to change sea ice parameters and to obtain accurate measurement results. The Norwegian University of Science and Technology (NTNU) carried out an ice shock steady-state model test at the Hamburg Ice Pool (HSVA) in Germany. The ice speed range that may cause steady-state vibrations is determined by employing a mobile device to change the ice speed, moving frequency and amplitude (Hendrikse et al. 2012). Tianjin University has built the only indoor ice pool in China, which can be used to test the model response under different ice directions and speeds and to measure the propulsion resistance of ships in the ice area so as to study the characteristics of ice-induced vibration and the propulsion resistance of ships (Fig. 2).

Numerical Simulation

Due to the complexity of the sea ice-ocean structure interaction process, it is difficult to analyze the impact of different parameters on the vibration of the ocean structure through field tests and indoor experiments. Numerical simulation methods can accurately change the calculation parameters and have the advantages of low cost and high efficiency for understanding and analyzing the sea ice-structure interaction mechanism.

Based on the peculiarity of ice loads, new numerical methods are used to study the

Ice-Induced Vibration and Noise of Ships,

Fig. 2 Tianjin University indoor ice pool (<http://image.baidu.com/>)



interaction between sea ice and structure, including discrete element (DEM), bonded element (CEM), near-field dynamics (PD), and several other methods. Since sea ice mostly floats on the sea surface in the form of discrete fragments which has a strong discontinuity, therefore, when the sea ice collides with each other or interacts with the ocean platform structure, it will further break. At present, the sea ice model, based on DEM, is developing rapidly which can effectively simulate the breaking phenomenon of sea ice and can reasonably reflect the ice load characteristics of different structures. During analysis process, the hull at the waterline and the ice at the edge are regarded as discrete, and various load values at the contact point are calculated according to empirical formulas. It was found by Su et al. (2011) that the variation in simulated local ice loads was noteworthy even if the ice properties (ice thickness and ice strength) were fixed. Thereafter the variation in the contact and icebreaking patterns was taken as an internal source of the variation in local ice loads, than the external ice conditions.

Types of Ice-Induced Vibration

Dynamic properties of sea ice load are related to factors such as the strength characteristics and breaking characteristics of sea ice. The physical mechanism of ice-induced vibration has not yet been fully identified, and it mainly includes the

following two categories: forced vibration theory and self-excited vibration theory.

Forced Vibration Theory

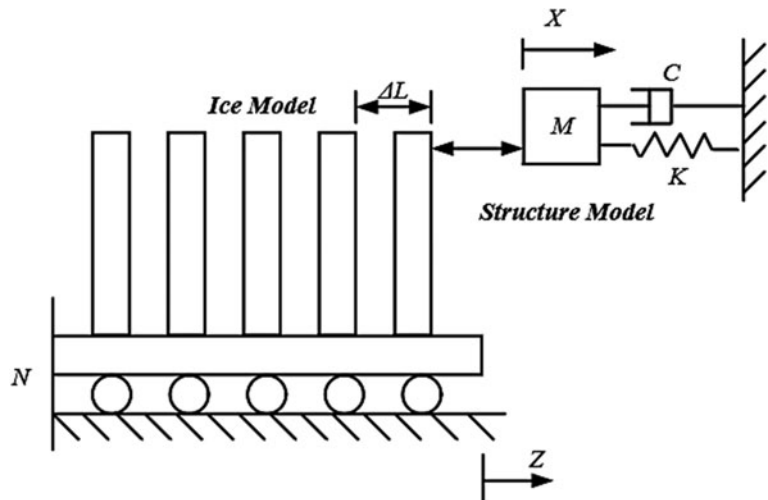
The forced vibration theory of ice-induced vibration was first proposed by Peyton. It is believed that dynamic ice force has inherent frequency spectrum characteristics, and the change of ice force has nothing to do with structural conditions. Based on the field tests results in Cook Bay, Matlock et al. (1971) proposed a calculation model to predict the vibration of flexible structures caused by ice sheets. Starting from the displacement and elastic deformation conditions of ice sheets and structures, the interaction between the two was considered during study. The sea ice is regarded as elastic beams with equal spacing, and the beam spacing is the fracture length of the sea ice. The stepwise integration method is used to solve the equation of motion of the system under the action of ice force (Fig. 3).

Self-Excited Vibration Theory

The theory of self-excited vibration started when Blenkarm was monitoring the ice load on the Cook Bay oil and gas platform and found that the stress rate of sea ice increased with decrease in strength. This will lead to “negative damping,” which in turn causes self-excited vibration of the structure (Sodhi 1988). Self-excited vibration refers to a special vibration phenomenon that

Ice-Induced Vibration and Noise of Ships,

Fig. 3 Matlock model schematic



continuously accumulates energy inside the structure and increases amplitude. The occurrence of self-excited vibration depends on the strength of sea ice itself, and the movement of the structure is also an important prerequisite. When the self-excited vibration theory is used to describe the structural vibration caused by ice, it describes the relationship between the response of the structure and the change process of change in ice force and the relationship between the uniaxial compressive strength of ice and the rate of acceleration.

Characteristics of Ice-Induced Vibration of Marine Structures

According to the different characteristics of structural response at different ice speeds, ice-induced vibration can be divided into the following three types (Li 2010): the quasi-static response, the steady-state vibration response, and the random vibration response. The probability of steady-state vibration is small, but the structural vibration is very violent when it occurs, which is very harmful to marine structures. The amplitude of random vibration response at fast ice speed is not as high as that of steady-state vibration, but it is the most common type. This type of vibration should be considered during fatigue analysis of ice-induced vibration, whereas the quasi-static response does not cause the dynamic response of the platform.

The Quasi-Static Response Mode

The quasi-static mode occurs when the ice speed is very slow (generally less than 0.02 m/s) and the ice surface is relatively flat. In this case, the loading of ice force can be regarded as a static force, and the dynamic effect can be ignored.

The Random Vibration Response Mode

When the ice speed is fast, the sea ice acts on the ocean structure to compress and fracture it quickly and continuously. The sea ice is compressed into pieces of different sizes. The irregularity of the sizes of the sea ice causes the randomness of the force change period. Therefore, marine structures experience random vibrations in dynamic response.

The Steady-State Vibration Mode

Steady-state vibration is a type of severe vibration with constant period and constant amplitude. The occurrence of steady-state vibration needs the conditions of a fixed ice speed. From the field data, the steady-state vibration mostly occurs at the flat tide level. At this time, the ice speed is generally 0.02–0.04 m/s. From the analysis of the ice and platform interaction process, the cause of steady-state vibration is found to be due to the equal structural vibration speed and the ice speed. This causes the structure and ice to form a cyclic loading in a vibration period of the structure, thereby controlling the change in period of the compressive ice force, and the change in frequency of the ice force is controlled at the natural frequency of the structure. When the steady-state vibration occurs, the structural response is extremely violent, which will not only cause damage to the marine structure but also cause damage to the pipelines, flanges, and other equipment on the upper part of the marine structure in a very short period of time.

Control Methods of Ice-Induced Vibration

Studies have shown that sea ice is the main load of navigational objects in ice areas, especially in the construction of marine structures around the development of oil and gas resources in ice areas. This is the reason suppression of ice-induced vibration has become a new research topic. The traditional method of increasing the overall rigidity of the hull to strengthen the anti-icing vibration ability is often ineffective. Because of the huge investment in shipbuilding engineering, this method is not economical and difficult to implement (Zhang 2008). Structural vibration control technology has become a choice to solve this problem. The main strategies are divided into three categories: changing the characteristics of the vibration source, top isolation, and dynamic vibration absorption technology.

Changing the Characteristics of the Vibration Source

In comparison with the action mode of wave, wind load, and seismic load on the structure, position of the ice load is relatively fixed which only acts on the structure at the waterline. In

addition, the alternating ice load is formed by the ice destruction process during the interaction between ice and the structure. Therefore, different ice destruction forms will lead to different ice forces. Based on the form of the interface between the ice load and the structure, special devices can be set up to change the form of ice destruction and then change the force of ice acting on the structure.

Top Isolation

Top isolation technology can also be called base vibration isolation technology. Its vibration reduction principle is to isolate the energy that causes structural vibration by installing a vibration isolation support between the supporting member of structure and the upper system, so that the input energy is consumed on the vibration isolation layer before reaching the upper systems. In order to achieve a good vibration reduction effect, the vibration isolation device must have high strength (carrying capacity) and low rigidity (large deformation) mechanical properties.

Dynamic Vibration Absorption Technology

Dynamic vibration absorber has been proved to be one of the most efficient means of vibration reduction for a long time. Its basic design principle has been studied by many scholars, and it has been

widely applied in engineering practices. To solve the vibration problem of the anti-ice platform under the action of ice load, Wang Lingyu et al. (1996) studied the vibration reduction control of the platform by using a tuned liquid damper (TLD) and carried out field tests on the JZ20-2 MUQ platform and obtained very meaningful results (Fig. 4).

Ice-Induced Noise

Definition of Ice-Induced Noise

Ships sailing in the ice area will inevitably collide and impact with floating ice. The impact load of floating ice will excite the vibration of the hull and generate structural noise and air noise at the same time. The noise generated by the impact load of the floating ice is called ice-induced noise.

Influence of Ice-Induced Noise

Due to low temperature in winter, the sea surface of the harbor will be frozen. Generally, ships with icebreaking function will be used to assist icebreaking and berthing in the port area. To ensure the safety of merchant ships entering and leaving the port, tugboats are to sail in the ice area to ensure the smoothness of the channel. When a ship sails in through a floating ice area or breaks

Ice-Induced Vibration and Noise of Ships,

Fig. 4 Ice-resistant oil drilling platform (<http://image.baidu.com/>)



ice in an intact ice area, the hull is subjected to the impact load of the ice layer, and as a result it vibrates violently. At the same time, structural and air noises are generated which are the transient noises caused by the impact load of the floating ice. The noise is transmitted to the cabins along the hull structure and air, causing the noise of some cabins to increase sharply and seriously affecting the comfort of the ship and its crew. The research on transient noise of ships sailing in ice regions is of great significance for the comfort of ships planned to be developed in the future for polar regions.

Control Methods of Ice-Induced Noise

For reduction of ice-induced noise caused by transient excitation when colliding with ice floes, adding various noise reduction measures in the cabins can increase the comfort of the ship; however, it is not an economical option (Wu 2019). Therefore, keeping in view the ice-induced vibration, a centralized noise control technology is proposed to provide references for the noise control of ships sailing in polar ice regions.

Increase the Thickness of the Bow Plate

Increasing thickness of the bow plate can not only increase the strength of the bow but also efficiently resist the impact of floating ice. To a certain extent, it can suppress the transient noise generated during ice collision.

Arrange Stiffeners Orthogonally on the Bow

The simulation results prove that from the perspective of restraining the impact on the bow, stiffeners are arranged orthogonally, and the longitudinal direction is better than the transverse direction. From the perspective of controlling the transient noise of the cabin, the stiffeners are also arranged orthogonally, but the transverse direction is better than the vertical direction.

Laying Damping Materials

Laying damping materials can obviously suppress the transient high-frequency vibration which can control the bow impact response and cabin

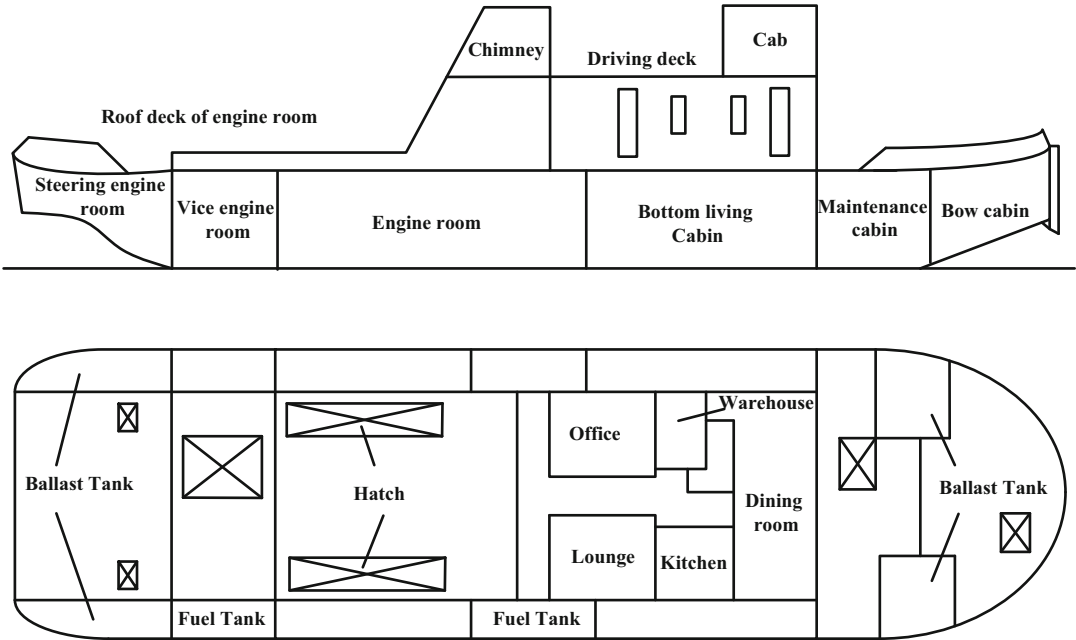
transient noise. With the increase of the damping loss factor, the bow impact response and cabin transient noise under ice impact will gradually decrease.

The Rules of Ice-Induced Vibration and Noise

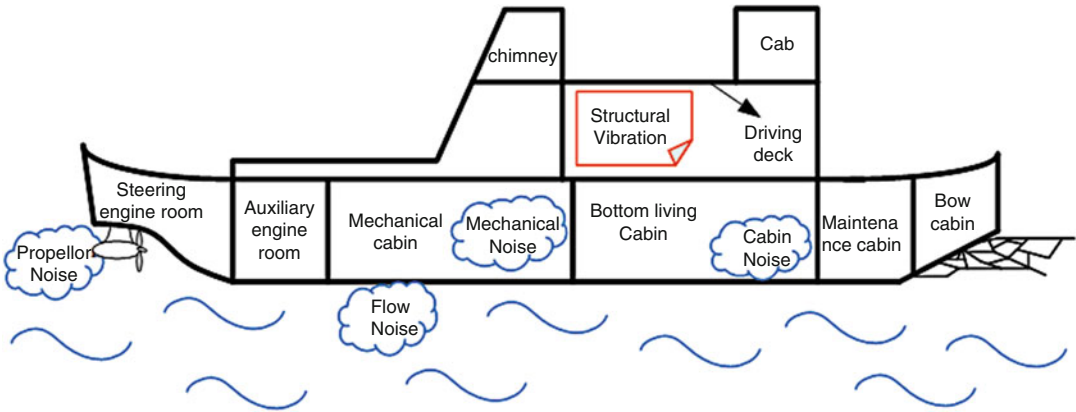
In order to obtain the rules of ship ice-induced vibration and noise, Pang Fuzhen (Pang et al. 2017) team carried out an icebreaker's vibration and noise measurement. The experimental ship type is the towed icebreaker work ship provided by Harbin Waterway Bureau. The test site was the Harbin section of Songhua River in Heilongjiang Province. The test temperature was $-2\sim 5\text{ }^{\circ}\text{C}$ and the ice thickness was measured as 40~50 cm, and the background noise was 45~50 dB (Figs. 5 and 6).

The experimental results obtained are as follows:

1. The ship trail test signifies that ice-induced load not only causes the low-order vertical vibration in the hull structure but also causes the local vibration in the ship.
2. During icebreaking operations, the vibration acceleration level in each area increased. Among them, the vibration acceleration along the length of the ship shows that it significantly increases at the bow, whereas at the midship it comparatively increases less. Increase at the stern is the lowest. In general, the ship's vibration response during icebreaking operations shows that the vibration at bow and midships is less, whereas in the engine room and stern area is more. The vibration response along the ascending direction at 1/4 of the length of the ship increases the most in the main deck area. The vibration in upper pilot area increases less, and in the cabin floor area, it increases the least. In general, the vibration response is the largest in the main deck area, slightly less in the driving deck area, and the lowest in the cabin floor area.
3. Icebreaking operations will have a certain impact on the habitability of various areas on



Ice-Induced Vibration and Noise of Ships, Fig. 5 Layout of the experimental ship cabin



Ice-Induced Vibration and Noise of Ships, Fig. 6 Ice-induced vibration and noise of ships

the ship. Among them, the vibration environment of the living area on the main deck, the upper pilot deck area, and the main deck fore and aft area are within the acceptable range of the human body. Only the vibration environment of the engine room floor area exceeds the acceptable range of the human body which becomes harmful.

- 4. During icebreaking operations, the noise of each cabin of the ship increases, but the noise of the cabins in different positions is different.
- 5. When the ship sails through ice area, noise of cabin of fore peak tank differs significantly along longitudinal direction. The noise change of living quarters of midship and steering

engine room in the stern region is less. When the ship navigates in open area, the noise of engine room is high; as a result, the change of noise of engine room is small when it sails in through ice area. For the cabin along draft direction, the noise change of cabin on the main deck is greatest, change of noise in cabin on the driving deck is lesser, and change of noise in living quarter of hull bottom is the least. In a word, when the ship sails through ice area, the cabin noise of engine room is still the most.

6. During icebreaking operations, the noise in some cabins exceeds the standard limit value, which affects the comfort of the ship. Therefore, in order to obtain a more comfortable experience, certain noise protection measures must be taken to reduce noise harmful effects.

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ICP – Imperial College Pile

- [Pile Capacity](#)

IFR (Incremental Filling Ratio)

- [Offshore Pile Driving](#)

IMO (International Maritime Organization)

- [Energy Efficiency Regulations from IMO](#)
- [Impact of Maritime Transport](#)
- [Ship Propulsion System](#)

Impact of Maritime Transport

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Synonyms

[Air emissions](#); [Automatic identification system](#); [Energy efficiency](#); [Environmental impact](#); [Fuel consumption](#); [IMO \(International Maritime Organization\)](#); [Shipping](#)

Definition

Maritime transport is considered as one of the most important human activities throughout history. It is one of the foundations of globalization, communication, international standardization, and trade liberalization (Kumar and Hoffmann 2002). Countries can be barely isolated from the economic activities of other countries due to the technological and economic impact. In fact, countries nowadays are seeking for the opportunity for economic growth by opening their borders and markets to attract foreign investment and trade. The globalization is irresistible.

The globalization has greatly increased the demand for goods movement so as the maritime transport. As the consequence, the energy consumption, environmental consequences, as well as the social impact that maritime transport brings to the world are unneglectable. They will be discussed in the following entries.

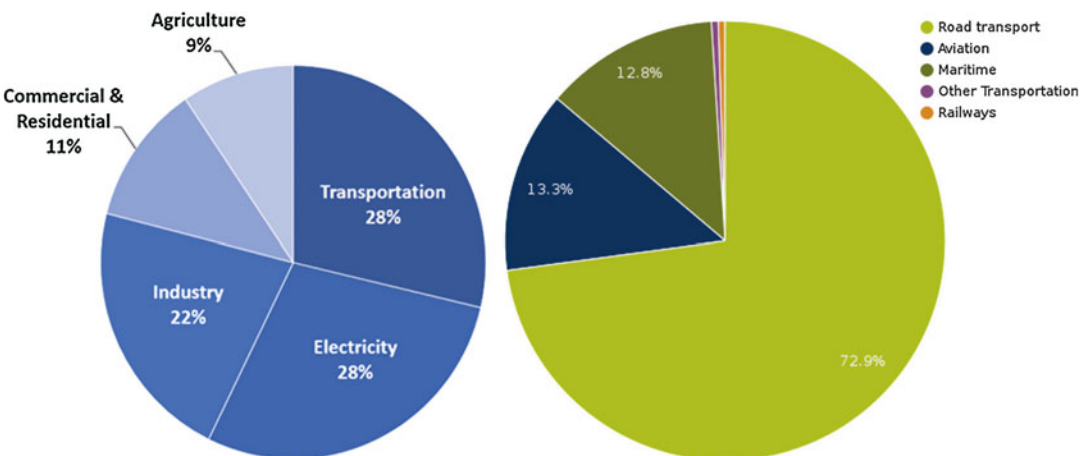
Economic Impact of Maritime Transport

Globalization addresses more demands of good transportation, and the goods movement has a number of energy and environmental impact. For example, the transportation sector generates the largest share of greenhouse gas emission,

which is nearly 28.5% of 2016 greenhouse gas emissions (Environmental Protection Agency 2018). Furthermore, the survey conducted by European Environment Agency in 2017 indicates the maritime transportation is responsible for 12.8% of total greenhouse gases (GHG) emission.

Shipping is considered as the most energy-efficient one, and the international shipping industry is responsible for the carriage of around 90% of the world trade. However, the fuel costs represent as much as 50–60% of total ship operating costs depending on the type of ship and service. There are over 40,000 active vessels that transport cargo around the world. According to the survey of International Energy Agency, those global shipping fleets are responsible for around 2% of the world's annual fossil fuel consumption (Fig. 1). Thus, there is a strong economic motivation to employ technologies that can reduce the fuel consumption as well as the GHG emission.

To reduce the energy consumption from maritime transportation, not only many regulations have come into force, for example, Ship Energy Efficiency Management Plan (SEEMP) and Energy Efficiency Design Index (EEDI), but also more sea areas are being declared as Emission Control Areas (ECA). Furthermore, the requirement of passing through ECAs gets harsher over time. Besides the regulations, shipping markets have employed more energy-saving measures.



Impact of Maritime Transport, Fig. 1 Source of GHG and share of transport GHG emissions (European Environment Agency 2018)

According to the shipping market survey conducted by DNV-GL (2015), the shipping companies selecting measures for energy saving mainly base on financial considerations. Nine simple energy-saving measures have been commonly addressed such as hull and propeller cleaning, slow steaming, voyage planning optimization, etc. Still, many of them do not realize the expected saving. Most of the shipping companies underestimate the share of fuel costs in total shipping cost. Cost is the key element for efficiency.

Environmental Impacts of Maritime Activities

The environment impacts from international shipping are several such as greenhouse gas emission and acoustic and oil pollution (Wikipedia 2018), and they can be summarized in different contexts. For example, considering the sources of pollution from shipping, they can be categorized into liquids such as oil spills, solids such as garbage, and air such as greenhouse gas.

The air pollution from ships is the best understood, which is evidenced by the international conventions and national regulations, and reducing the air pollutant emissions from shipping is the key to avoid the most catastrophic impacts of climate change. The CO₂ and other climate pollutant emissions are still increasing. According to the survey conducted by ICCT (2017), the total shipping CO₂ emissions increased 2.4% from 2013 to 2015. Three ship classes accounted for 55% of the total shipping CO₂ emissions, container ships (23%), bulk carriers (19%), and oil tankers (13%). CO₂ is not the only pollutant from ships. Other pollutants such as sulfur dioxide (SO₂), nitrogen dioxide (NO_x), as well as particulate matter (PM_{2.5}) are also considered to contribute to climate change. Air pollution from international shipping accounts for approximately 50,000 premature deaths per year in Europe, at an annual cost to society of more than €58 billion (European Federation for Transport and Environment 2018).

Air pollution from ships continues to increase as the sector grows. According to IMO

greenhouse gas study, the nitrogen dioxide increased by 3.5% between 2013 and 2015, while the sulfur dioxide increased by 1%. The shipping emission can be driven by a number of factors such as the growing number of the international fleet, the more powerful engines, and the speeding up of ships. To reduce the emission from ship, one way is to improve the energy efficiency and develop alternative energy sources.

Sound pollution from ships can also harm the marine species who may rely on sound for reproduction. Growing levels of noise pollution in the ocean could drive fish away from their habitat into their death. For example, baby reef fish use sounds made by fish, shrimps, and sea urchins as a cue to find coral reefs (Simpson and Jeffs 2011). With human noise pollution from ships, this crucial behavior is coming under threat.

Social Impact of Maritime Industry

Social impacts are effects on individuals, communities, and society. There are many potential social impacts of maritime sectors. Employment can be considered as one of the most important means. Maritime sectors help to generate significant quantities of jobs. According to the ICS and BIMCO report, the global supply of seafarers in 2015 was estimated at 1,647,500, of which about 774,000 are officers and 873,500 are ratings. It can generate significant social benefit.

Another social impact when it comes to maritime industry is the public safety. The safety of vessels is critical to the global economy. According to the 2017 review conducted by Allianz Global Corporate & Specialty (AGCS), the number of shipping losses during 2016 is up to 85. Cargo vessels (30) account for more than a third of 2016's losses. The loss of a large container ship or cruise ship in environmentally sensitive waters could cost billions of dollars. The potential cost could be wreck removal, cargo liabilities, oil pollution, etc.

Piracy threat and the cyber threat should not be underestimated. Pirates attacked 43 ships and captured 58 crews during the first quarter of 2017, slightly up year-on-year. The introduction

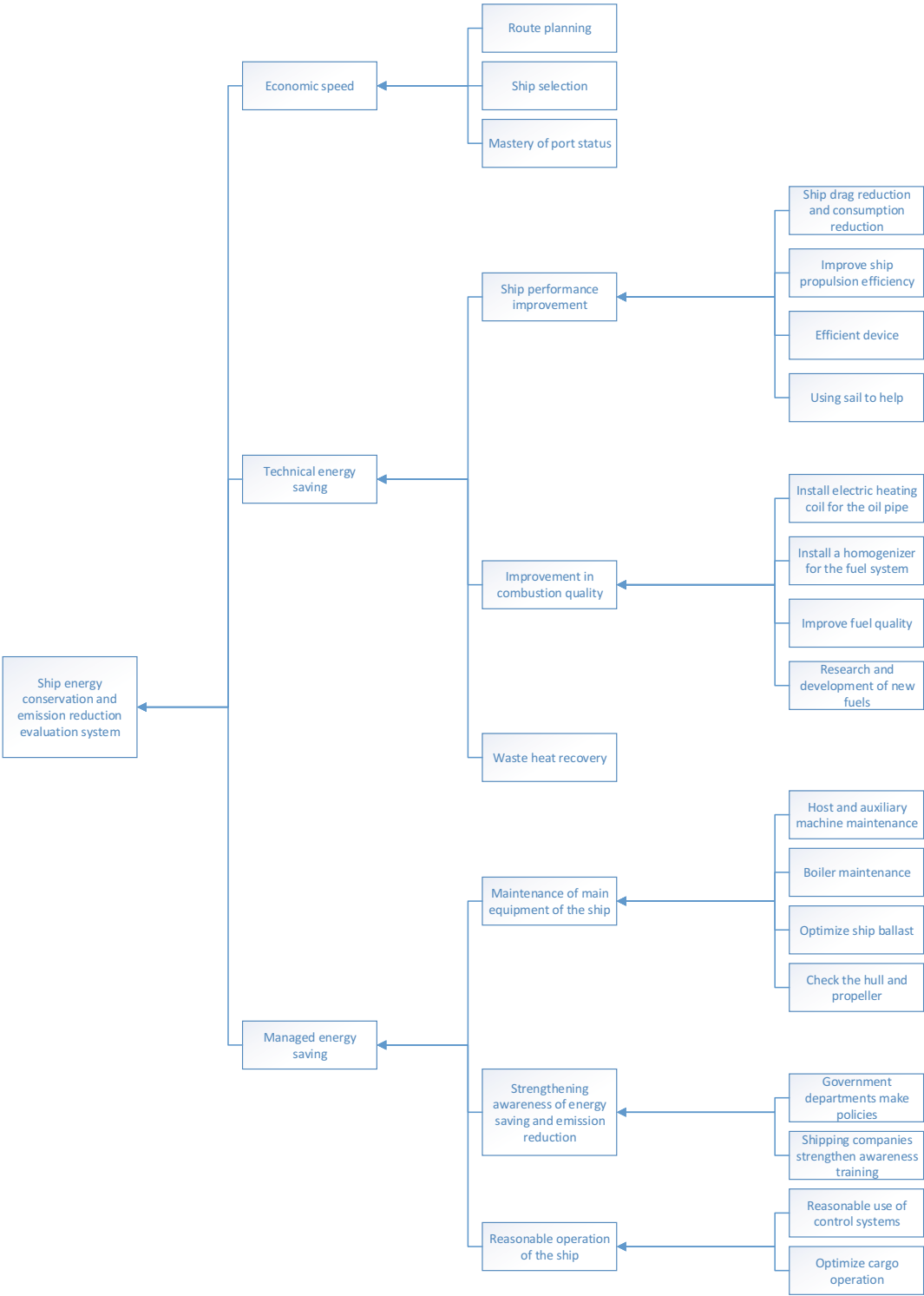
of armed guards onboard vessels and the presence of a multinational naval task force successfully reduce the threat of pirates. With the development of autonomous ships, ships are increasingly dependent on IT services, such as Automatic Identification System (AIS), Global Positioning System (GPS), or other onboard control systems which are controlled through wireless network. Cyber threat may result in the disruption from technical failure or serious maritime event such as collision or other damage. According to the Draft IMO Guidelines Cyber Risk Management, to manage cyber risk, five steps are recommended to follow: identify, protect, detect, respond, and recover (IMO 2017).

Energy Efficiency Measures to Reduce Emissions from Shipping

The energy consumption of maritime transport mainly comes from the combustion of fuel oil from ships. The emission comes from the exhaust gas generated after combustion. A ship's fuel consumption rate is an indicator that comprehensively reflects the potential of energy saving and emission reduction technologies that can be applied on the ship. First, a ship's own performance can determine her energy-saving and emission reduction strategies. For example, the status of main and auxiliary engines of different ships, paint primer and sewage conditions, operating years, models, and hull immersion volume define the ship's energy conservation and emission reduction. The main auxiliary machine of the ship is an important energy-consuming equipment and safety equipment in the ship transportation process. The higher the operating efficiency of the main and auxiliary machines, the higher the combustion efficiency of the ship. This factor could be used to develop technologies to increase the ship's engine operation efficiency, leading to reduction of the ship's fuel consumption and emissions. Furthermore, the degree of wear of a ship's main engines varies along the ship's service period. For an aged ship, her main engine wears a lot, and its unit fuel consumption also increases. It will affect energy utilization efficiency and

pollutant discharge of the main engine. The power provided by a ship's main engine will be used to overtake the ship's resistance to move forward. A ship's resistance in her actual operational environment also has significant impact to a ship's energy performance. Technologies used to lower a ship's resistance through better design, construct, and operation are also extremely important to reduce negative environmental impacts from shipping industry. For example, a ship's encountered sea weather conditions significantly affect a ship's energy consumption and air emissions. When operated in harsh sea environment, the ship's running resistance will increase dramatically, resulting in large increase of the ship's main engine load. It means more fuel consumption and exhaust gas emissions. In addition, when sailing at higher altitudes, due to lower air pressure and lower oxygen content in the air, the combustion of the ship's fuel cannot be fully realized, resulting in a significant reduction in energy conversion efficiency. In order to ensure the normal operation, the main engine has to burn more fuel consumption to keep the ship's sailing safety. The geographical environment will also affect energy consumption and exhaust emissions, such as channel crossing, channel bending angle, channel depth, and channel width. In the case of the same other conditions, having a relatively small channel curvature will reduce the proportion of fuel consumption.

Furthermore, capital investment and economic benefits can also affect a ship's energy efficiency. For the newbuilding ships, the effect of implementing energy consumption and emission reduction technologies can be reflected by the cost of ship capital investment and the level of benefits. When shipbuilding enterprises make high-efficiency and low-cost inputs, they can promote the implementation of low-cost and high-efficiency of ships, thus achieving the realization of energy-saving and emission reduction effects. When shipyards make higher cost inputs, they will lead to lower economic benefits and may also make the yards generate negative economic benefits. It in turn reduces the enthusiasm of shipbuilding enterprises for energy conservation and emission reduction. The economic benefit of



Impact of Maritime Transport, Fig. 2 Overview of strategies to improve a ship’s energy efficiency from life cycle perspectives

a shipyard refers to the ratio of product input in the operation of the ship. The level of its benefits affects the entire shipbuilding industry to a certain extent. The higher the economic benefit, the faster the development of energy conservation and emission reduction of ships. The same situation is also applied to ship owners/shipping companies, if the benefits of energy efficiency are to be claimed by them. Therefore, economic benefits are always one of the important factors affecting a ship's energy efficiency and emission reduction during a ship's whole service life, i.e., design, construction, operation, and wrecked stages.

In addition, shipping management is another important factor that can be used to maximize a ship (fleet) energy efficiency. The management factors mainly include the shipping companies' optimal planning of the shipping routes, the choice of ships to call the ports, training of crews, maintenance of ships, arrangement of ship cargoes, etc. Reasonable route planning can reduce unnecessary sailing time, avoid harsh sea conditions, reduce ship consumption, and achieve energy saving and emission reduction. Better planning strategy of ship calling port and optimal arrangement of calling port order can minimize ship in port and the docking time. By reducing unnecessary waiting time in port, anchoring conditions, a ship can be either maximumly utilized in cargo transport or sailing with low speeds, leading to a high energy efficiency in terms of unit cargo transported. Moreover, from a ship's long-term management perspective, a ship's energy efficiency will normally decrease in terms of her sailing time at sea. Good maintenance strategy can also help to reduce a ship's total resistance and increase her engine propulsions during her entire service time. It should be noted that any kind of benefits claimed by various energy efficiency measures can be only reached by the good implementation/operations from crew members. Thus, the training of crew members and human factors involved in a ship's life cycle operation process will finally determine the actual energy efficiency of a ship. Figure 2 categorizes a brief summary of all possible energy efficiency measures that can be used to increase a ship's energy efficiency.

Cross-References

- [Alternative Fuel for Ship Propulsion](#)
- [Energy Efficiency Regulations from IMO](#)
- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

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In Situ Permeability Measured Using Piezo-penetrators

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Definition

Piezo-penetrator

The penetrometer with pore pressure transducers attached to the probe. It is used to obtain both the penetration resistances and pore pressure.

Introduction

Consolidation of soil is concerned frequently in geotechnical practices. The consolidation depends on soil permeability which may be measured through one-dimensional compression test in laboratory and/or in situ dissipation tests of piezo-penetrometers. The disturbance of soil sample is avoided in the in situ tests, which is especially appreciated by the offshore geotechnical investigations. The dissipation tests of two widely used penetrometers, the piezocone and piezoball, are introduced in this entry, followed by the interpretation of the testing data.

Background

Consolidation of clayey soil and then the settlement of footings depend on a fundamental soil property, permeability. Soil permeability can be measured in laboratory, of which the reliability becomes doubtful if soil sample undergoes moderate or serious disturbances. Compared with the one-dimensional compression test in soil laboratory, the in situ tests are carried out in natural soil, without the need to obtain (nominally) undisturbed soil samples for later testing in the laboratory. This is especially important for offshore practice, since the cost of obtaining high-quality samples for laboratory testing is very high, and the low shear strengths in the upper few meters of the seabed pose particular practical difficulties. In situ tests using piezocones or piezoballs have thus been developed during the last three decades to obtain the coefficient of consolidation c_v and then the coefficient of permeability.

The Dissipation Test of Piezocone

The cone penetrometer is argued to be the most widely used offshore site investigation tool. Traditionally, the histories of the cone resistance and frictional resistance along the sleeve are measured and recorded with the downward movement of the penetrometer. If the conical probe is equipped with a pore pressure transducer located just behind the cone shoulder, the pore pressure during

the entire penetration process and the subsequent dissipation test can be measured as well. It is assumed that the penetration is nearly under undrained conditions in clays, silty clays, or clayey silts, as the standard piezocones with project areas of 1000 mm^2 or 1500 mm^2 are displaced with a velocity of 20 mm/s . The dissipation test is conducted by halting a penetrometer at the targeted depth, and the variation of the pore pressure accumulated during the penetration phase is thus recorded. The so-called operative coefficient of consolidation, c_h , is deduced by matching the dissipation curve recorded with an analytical or numerical curve, in which both curves are expressed as a relationship between normalized dissipation time and normalized excess pore pressure. A representative numerical curve was proposed by Teh and Houlsby (1991), with the normalized time T^* defined as function of dissipation time t , diameter of cone shoulder D , and rigidity index of soil I_r and c_h :

$$T^* = \frac{c_h t}{(0.5D)^2 \sqrt{I_r}} \quad (1)$$

The normalized excess pore pressure indicates the ratio between the excess pore pressure recorded during the dissipation phase and its nominal initial value, for example, suggested by Sully et al. (1999).

Interpretation of the dissipation test is complicated by a number of factors related to the elastoplastic response of soils. For example, both the initial distribution of the excess pore pressure and the subsequent dissipation are affected by the preyield and plastic stiffnesses of the soil, the degree of overconsolidation in situ, and the extent of any consolidation during penetration. In the 1980s and 1990s, a number of simplified decoupled approaches have been proposed to tackle the penetration and dissipation phases separately: (a) The continuous penetration is regarded as an ideally undrained process, then cylindrical or spherical cavity expansion or the more sophisticated strain path method can be used to obtain the excess pore pressure at the end of penetration. (b) Dissipation around the cone belongs to the traditional consolidation problems which are solved using finite element or finite difference

approaches based on Biot's theory or Terzaghi-Rendulic consolidation theory (Lunne et al. 1997). The shortcoming of the Terzaghi-Rendulic theory is that the total stresses during the dissipation phase are assumed to be constant. Recently, large deformation finite element approaches were developed to simulate the entire "penetration-dissipation" process (Mahmoodzadeh et al. 2014).

The operative coefficient of consolidation c_h was found empirically to be much larger than c_v in most cases, and their relationship has remained unclear until recently. According to parametric study using the modified small strain and large deformation finite element approaches, Mahmoodzadeh et al. (2014) suggested that

$$c_h = \frac{3(1-\mu)}{(1+\mu)} \left(\frac{\lambda}{\kappa} \right)^\alpha c_v \quad (2)$$

where μ is the Poisson's ratio of soil, λ and κ are the slopes of the normal consolidation line and swelling line, respectively, and α is a fitting coefficient. A typical value of α may be taken as 0.75. The dissipation around the cone is mainly along the horizontal; however, c_h does not represent the "horizontal" coefficient of consolidation. When the value of c_v is available, the coefficient of permeability can be deduced from classic Terzaghi's consolidation theory.

The Dissipation Test of Piezoball

Except for cone penetrometers, a full-flow penetrometer with ball-shaped probe, termed ball penetrometer, has been used increasingly during the last decade, in particular in soft seabed sediments due to its advantages of larger projected area and minimal correction for overburden pressure (Kelleher and Randolph 2005). Similar to the piezocones, the ball penetrometer may be fitted with a pore pressure transducer at equator or mid-face of the probe. The piezoball with dual pore pressure transducers is another option.

Piezoballs and piezocones share similar procedures of penetration and dissipation tests. A limited number of in situ and centrifuge model

tests have been carried out for piezoballs, assisted by the large deformation finite element simulations (Mahmoodzadeh et al. 2015). The normalized dissipation curves of piezoballs were similar in shape to that of piezocones. However, the normalized dissipation time of piezoballs, T_b , depends not only on the ball diameter D_b , but also on the diameter of the shaft connecting with the ball probe, d , which can be expressed as:

$$T_b = \frac{c_h t}{D_b d I_r^{0.25}} \quad (3)$$

The definition of c_h in Eq. 3 is identical to that through Eq. 2. The shapes of the normalized dissipation curves in terms of the equator and mid-face of the spherical probe can be found in Mahmoodzadeh et al. (2015).

Key Application

The dissipation tests of piezo-penetrometers have been used in a few offshore projects to determine the in situ permeability of cohesive soils, as reported in Lunne et al. (1997).

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Index Matching Fluid (IMF)

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- [Ultra-short Baseline Underwater Acoustic Location Technology](#)

Infrastructureless Network

- [Underwater Acoustic Sensor Network](#)

Inherent Variability

- [Spatial Variability](#)

Installation and Decommissioning

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Definition

Pipeline installation is a highly nonlinear 3D process (Trapper 2019), which involves coupled in-plane and out-of-plane phenomena, such as full pipeline-seabed interaction, lateral buckling, lateral stability under hydrodynamic loads, torsion, sideways walking, etc. If the laying of pipelines is finished prior to the arrival of the platform or encounters bad weathers, the pipelines need to be abandoned onto the seabed, and recovery and reinstallation of pipelines need to be carried out when the platform is available. In the *abandonment operation*, the A&R cable lowers a pipeline down to the seabed by a pull head and in the *recovery operation* lifts it up to the sea level (Wang et al. 2015).

Scientific Fundamentals

Installation of Risers and Pipelines

With the discovery of offshore oil fields in the shallow waters of the Gulf of Mexico during the late 1940s, offshore pipeline installation was invented. The first “offshore” pipeline in the Gulf of Mexico was constructed in 1954. Now, offshore fields are being discovered in water depths of 3,000 m, and the pipeline installation technology is keeping up. The most common methods of pipeline lay installation methods are:

- S-lay (shallow to deep)
- J-lay (intermediate to deep)
- Reel lay (intermediate to deep)

Shallow water depth ranges from shore to 150 m. Intermediate water depth is assumed to be 150–300 m. Deep water has water depths greater than 300 m. Offshore magazine produces a survey of most of the pipeline lay barges that work in the United States every year. This survey does not cover all the lay barges of all the countries that do offshore work, but it does cover the bigger international ones – Heerema, Saipem, Stolt, Technip, Allseas, McDermott, Global, and Subsea 7.

Other methods that have been used for pipeline installation are tow methods consisting of:

- Bottom tow
- Off-bottom tow
- Mid-depth tow
- Surface tow

Tow methods can be used for installing pipelines from shallow water depths to deepwater depths depending on the design requirements.

Pipeline Installation Design Codes

The most commonly used offshore pipeline installation codes are as follows:

- DnV OS F101 (Det Norske Veritas)
- API RP 1111 (American Petroleum Institute)
- ASME B31.4 – pipeline transportation systems for liquid hydrocarbons and other liquids
- ASME B31.8 – gas transmission and distribution systems

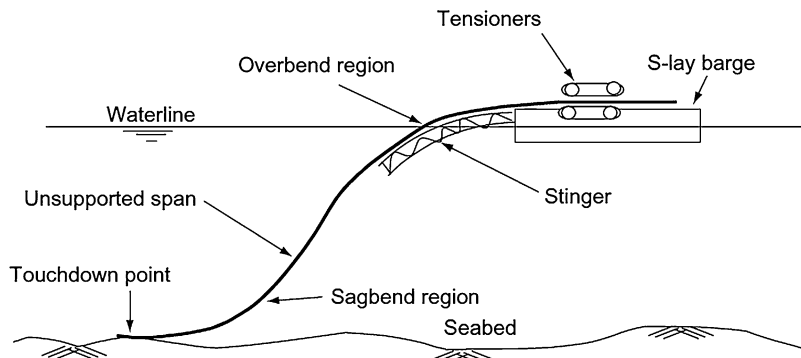
Lay Methods

S-Lay

The most common method of pipeline installation in shallow water is the S-lay method. A typical S-lay configuration is shown in Fig. 1 (Guo et al. 2014). In the S-lay method, the welded pipeline is supported on the rollers of the vessel and the stinger, forming the overbend. Then it is suspended in the water all the way to the seabed, forming the sagbend. The overbend and sagbend form the shape of an “S.”

In the S-lay method, tensioners on the vessel/barge pull on the pipeline, keeping the whole section to the seabed in tension. The reaction of this pull is taken up by anchors installed ahead of the barge or, in the case of a dynamically

Installation and Decommissioning,
Fig. 1 S-lay configuration



positioned (DP) vessel, by thrusters. These barges/vessels are fitted with tension machines, abandonment and recovery (A&R) winches, and pipe-handling cranes. The firing line for welding the pipe may be placed in the center of the barge or to one side. The firing line consists of a number of stations for welding, nondestructive evaluation (NDE), and field joint application. The field joint station is located after the NDE station and the tension machines.

J-Lay

To keep up with the discovery of deepwater oil and gas fields, the J-lay system for pipeline installation was invented. In this system, lengths of pipe are welded in a near-vertical or vertical position and lowered to the seabed. The J-lay configuration is shown in Fig. 2 (Guo et al. 2014). In this configuration, the pipeline from the surface to the seabed is one large radius bend resulting in lower stresses than an S-lay system in the same water depth. There is no overbend, and a large stinger required in S-lay to support the pipe in deep water is eliminated. The horizontal forces required to maintain this configuration are much smaller than required for an S-lay system. This lends itself for DP shipshape vessels and derrick barges to be equipped with a J-lay tower.

The J-lay method is normally used in water depths greater than 150 m. These water depths

are normally too great for moored lay vessels to operate, because the required tensions and pipe bending stresses are too large.

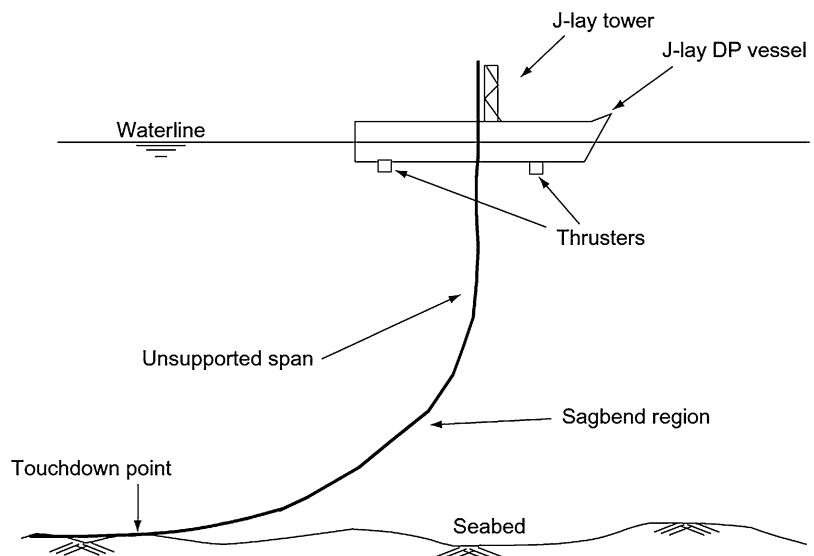
Reel Lay

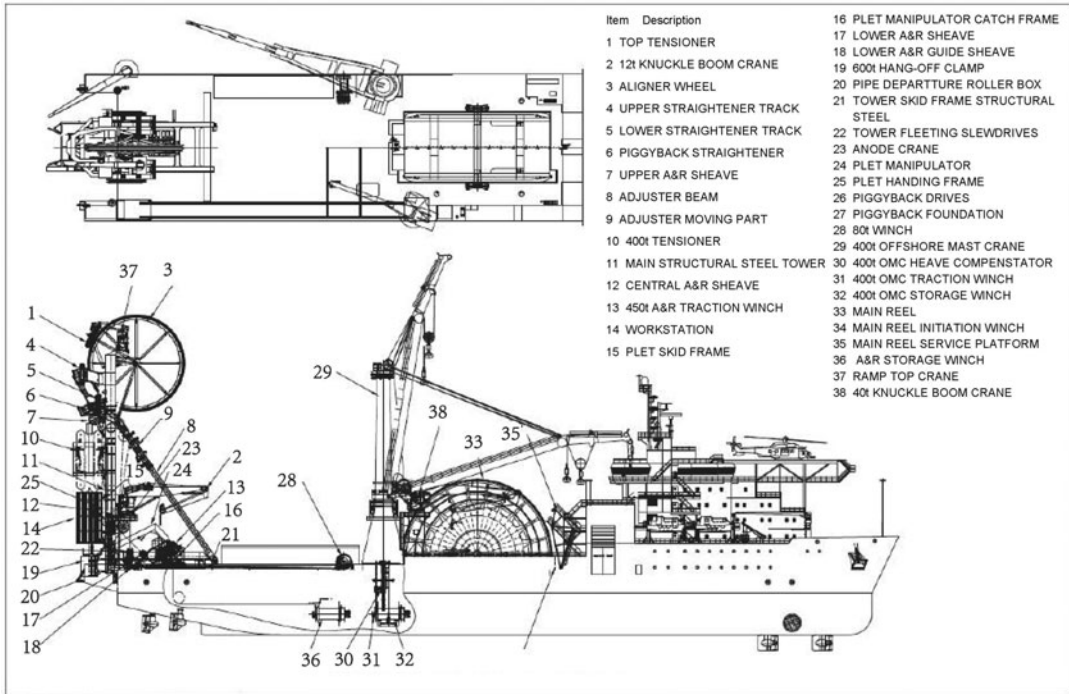
Reel pipelay is a method of installing pipelines in the ocean from a giant reel mounted on an offshore vessel, as shown in Fig. 3 (Smith and Clough 2010). Pipelines are assembled at an onshore spool-base facility and spooled onto a reel which is mounted on the deck of a pipelay barge. The first application of the reeled pipeline was on D-Day when the allies were supplied with fuel across the English Channel using a small diameter pipeline unreeled from a vessel. Commercial application of reeled pipeline technology was not available until the early 1970s when Santa Fe Corporation built the first reel vessel.

Reel technology also provides a safer and more stable work environment, thus speeding pipeline installation. Reeled pipelines can be installed up to 10 times faster than conventional pipelay. The greater speed allows pipelines to be laid during a short weather window. This can extend the normal construction season. Reel pipelay can be used on pipelines up to 457.2 mm in diameter.

The reel method reduces labor costs by permitting much of the welding, X-raying, corrosion coating, and testing to be accomplished onshore, where labor costs are generally lower than

Installation and Decommissioning,
Fig. 2 J-lay configuration





Installation and Decommissioning, Fig. 3 Seven oceans general arrangement of reel lay

comparable labor costs offshore. After the pipeline is reeled onto the drum of the pipe-laying vessel, it is taken to the offshore location for installation. The reeled pipeline can be installed in an S-lay method or J-lay method depending on the design of the reel vessel and the depth of water. Reel vessels can have vertical reels or horizontal reels.

Horizontal reel vessels lay pipelines in shallow to intermediate water depths using a stinger and S-lay method. The station keeping of vessels with horizontal reels can be done by anchors or DP.

The vertical reel vessel can normally install pipelines from the intermediate water depths to deep water, and station keeping is always DP. For deep water, the J-lay configuration is used and no stinger is required.

The pipe is unreeled, straightened, devalized, and connected to the wire rope from the seabed preinstalled holdback anchor. The sagbend stresses are controlled by the tensioning system on the reel vessel. The vessel moves ahead while it

slowly unreels the pipeline from the drum. When the end of the pipeline on the drum is unreeled, a pull head connected to a wire rope is attached. The end of the pipeline is lowered to the seabed by paying out the A&R wire rope from the reel vessel slowly in a controlled method always maintaining sufficient tension in the pipeline. A buoy is attached at the end welded pipeline on the reel drum. On returning, it pulls the end of the pipeline using the A&R cable, removes the pull head, and welds it to the pipeline on the drum. It then begins the unreeling process again.

The main disadvantages of the reeling method are the following:

- Connecting the ends of the pipeline segments.
- Amount of time to re-reel the pipeline to remove a buckle.
- Establishing a spool base close to the location where the pipeline is to be laid.
- Concrete-coated pipelines cannot be reeled.
- Only specifically designed pipe-in-pipe pipelines can be reeled.

- The pipeline is plastically deformed and then straightened. Some thinning of the wall and loss of yield strength of the material in localized areas can occur (Bauschinger effect) on the A&R cable. The reel vessel returns to the spool base to load more.

Tow Methods (Guo et al. 2014)

In the tow methods, the pipeline is normally constructed at an onshore site with access to the water. These methods can be used for installing pipelines across inland lakes, across wide rivers, and offshore. In the case of an offshore pipeline, the advantage of these methods is that the pipeline is welded onshore with an onshore pipeline spread. Once the pipeline is complete and hydrotested, the pipeline is dewatered and moved into the water, while being attached to a tow vessel (a large anchor handling vessel). It is then towed to a location offshore where each end is connected to pre-installed facilities. This could be cheaper than using a lay barge spread to install the pipeline offshore. The advantage occurs mainly if several small lines need to be laid and can be bundled inside a larger pipe. However, a case-by-case analysis is required to determine the risk versus the reward. The pipeline can be made up either perpendicular or parallel to the shoreline.

For a perpendicular launched pipeline, a land area that can accommodate the longest section of the fabricated pipeline must be leased. A launchway consisting of a line of rollers or rail system needs to be installed leading from the shore end right into the water. First, all the sections that make up the pipeline are fabricated and tested. Then, the first section of pipeline is lifted by side booms and placed on the rollers on the launchway. The cable from the tow vessel is attached, and the section is pulled into the water, leaving sufficient length onshore to make a welded tie-in to the next section. In this manner, the whole single pipeline is fabricated and pulled into the water. A holdback winch is always used during these pulls to maintain control and, if need be, to reverse the direction of pull.

In the parallel launch method, the land area acquired along the shore is normally the total

length of the pipeline to be towed. This could be longer than that acquired during perpendicular launch. No launchway is needed. After the sections of the pipeline are welded and tested, the sections are strung along the shoreline. The pipeline sections are welded together to make up the length of pipeline to be towed. The completed pipeline is moved into the water using side boom tractors and crawler cranes for the end structures. The front end is attached to the tow vessel, while the rear end is attached to a holdback anchor. The anchored tow vessel winches in the tow cable in such a manner that it gradually moves the pipeline laterally into the water, while the curvature is continuously monitored. When the whole length of pipeline and its end structures make one straight line, the tow vessel begins to tow the pipeline along the predetermined tow route. For pipelines that are to be towed into deep water, pressurized nitrogen can be introduced into the pipeline to prevent collapse or buckling under external hydrostatic pressure. A depth of 900 m can be achieved. Greater depths would require a stop for another recharge of pressurized nitrogen from the surface. This has never been done.

Bottom Tow

As the name indicates, the bottom tow method pulls the pipeline along the seabed to its final location. The length of a single section of pipeline is limited by the available bollard pull of the vessel used. The bollard pull must be greater than the total submerged weight of the pipeline, plus the partially submerged weight of the end structures, times the friction coefficient of the soil. For an estimate, the initial friction coefficient is taken as unity. Two to three vessels can be used in tandem to obtain additional bollard pull capability.

The ends of a bottom-towed pipeline are normally connected by deflect-to-connect method. In this method, the end sections of the pipeline are made to float a few feet above the seabed by providing additional buoyancy for this length and attaching anchor chains at discrete spacing along this length.

The buoyancy and chains are attached onshore with chains strapped over the buoyancy

pipe during towing and deployed at the pipeline's final location. This length can then be pulled laterally by attaching cables to the end of the pipeline from the facility. Once the pipeline end structure is secured at the facility, the connection can be made by flanges (in diving depth) or by hydraulically activated connectors (in deep water).

The disadvantages of bottom tow are the following:

- An extensive bottom survey along the tow route is required.
- The route must not cross existing pipelines. Otherwise, additional costs will be incurred for installing and removing structures to protect existing pipelines.
- Subsea transponder systems are required to locate the pipeline during tow and to place it in its final destination.
- The pipeline lying along the beach or near-shore can be subjected to large wave forces from storms, and this could compromise pipeline integrity. A pipe anchor system is required on standby for this emergency.
- In crossing shallow water areas, a chase vessel is required to keep fishing vessels from crossing the bottom-towed pipeline.

Off-Bottom Tow

In the off-bottom tow method, the submerged pipeline is buoyant and floats above the seabed at a predetermined height during the towing. This is achieved in the same manner as described in the above section for connecting the ends of bottom-towed pipeline. The buoyancy and chains are attached in discrete modules for the length of the pipeline.

The advantage of this method over bottom tow is that existing pipelines can be crossed by placing concrete mats placed over these pipelines and allowing the hanging chains to drag over the mats. No extensive protection structure is required. However, buoyancy and chains are required for the entire length of the pipeline. If several pipelines are needed for field development, the buoyancy and chains can be recovered and used again.

Mid-depth Tow

In the mid-depth tow method, the entire length of pipeline is kept at a considerable height above the seabed during towing. To achieve this, discrete buoyancy, chains, and a large tension applied to the pipeline are required.

The tension is applied by two tow vessels pulling in opposite directions at each end of the pipeline. Once the pipeline reaches its desired height, the front tow vessel applies more thrust while the back tow vessel cuts back on its reverse thrust. A third vessel monitors the height of the pipeline in the middle by using a subsea transponder system. This vessel sends its signal to the two tow vessels, which see the height in real time and adjust their thrusts appropriately to keep the pipeline within the desired height range. This method is not suited for very long pipelines (greater than 4.8 km).

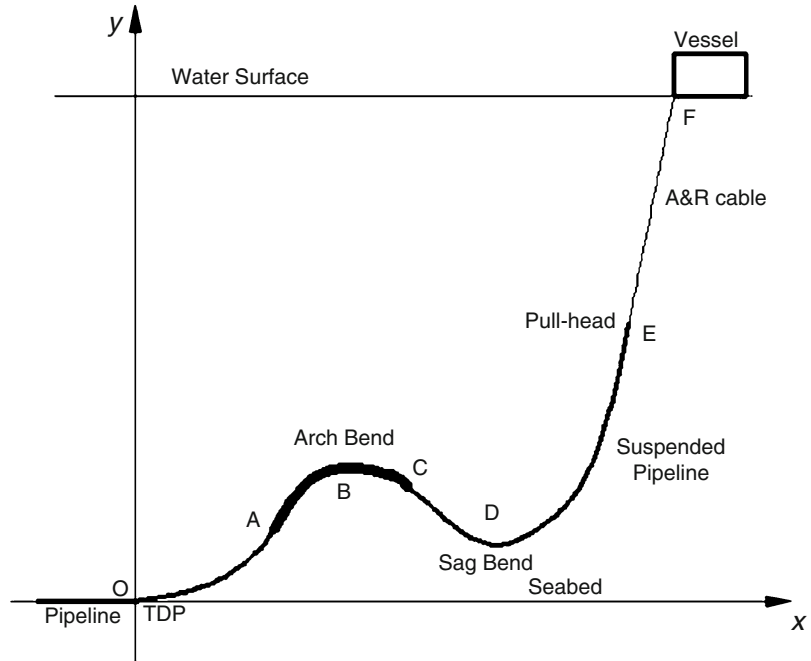
Surface Tow

Surface tow of pipelines is similar to mid-depth tow except that the pipeline will not require any chains. The two vessels, one at each end, keep the pipeline in tension while it is towed on the surface. Only a survey of the final pipeline route is required. This method can be used for shallow water. For deep water, a sophisticated controlled flooding and/or buoyancy removal system is required. Not many pipelines are installed by this method.

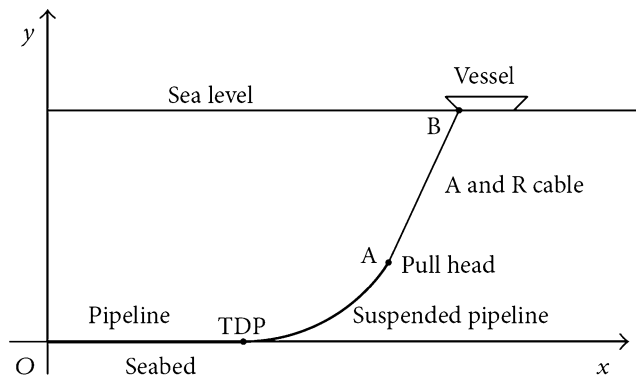
Abandonment and Recovery of Risers and Pipelines

The sketch of the SLWR abandonment and recovery operation is shown in Fig. 4 (Wang et al. 2015). In the process, the SLWR is put down to the seabed from the sea surface or lifted up from the seabed to the sea surface by pull head E which is hanged by the A&R cable of the installation vessel. In the abandonment and recovery system, there are the pipelines (it includes the lower section O-A, buoyancy section A-B-C, upper section C-D-E), the A&R cable which supplies the tension to the riser, and the installation vessel in

Installation and Decommissioning,
Fig. 4 SLWR abandonment and recovery operation



Installation and Decommissioning,
Fig. 5 Pipeline abandonment and recovery operation



which there is the A&R winch. The engineers control the tension and length of the A&R cable and the location of the installation vessel to conduct the abandonment and recovery of SLWR. Because in the abandonment and recovery processes, the riser is hung off by cable at the hang-off point, the two physical processes are also called one point lowering and lifting. The processes are mutually inversed and can be described by the same mathematical model. The mechanical behavior of the riser and the A&R cable should be analyzed for the installation safety.

A Simplified Mathematical Model of Pipeline Abandonment and Recovery in Deep Water

The sketch of the simplified pipeline abandonment and recovery operation is shown in Fig. 5 (Zeng et al. 2014). In the processes a pipeline is lifted up from the seabed to the sea surface or put down to the seabed from the sea surface by joint A. The two physical processes are generally called one point lifting and lowering.

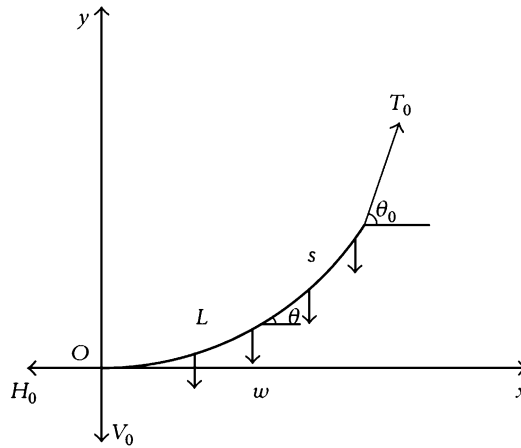
On the problem the following simplifications are made based on offshore engineering

experience: the marine environment is stable, the seabed can be regarded as rigid plane, the lifting and lowering processes are slow, and the material of pipelines is isotropic and always in the elastic state. As shown in Fig. 6, the touchdown point (TDP, the point where the suspended pipeline contacts with the seabed) is located at the origin O of the Cartesian coordinate system, where T_0 is the resultant force at joint A, H_0 and V_0 are the horizontal force and the vertical force at the origin, w is the pipeline submerged weight per unit length, θ is the angle between the pipeline axis and the horizon, and θ_0 is the angle between the direction of T_0 and the horizon. The natural coordinate system is established along the pipeline. It is clear that the physical quantities of the pipeline are the functions of the arc length s . Taking a

short segment of the pipeline, as shown in Fig. 7, it is easy to deduce the basic governing differential equations.

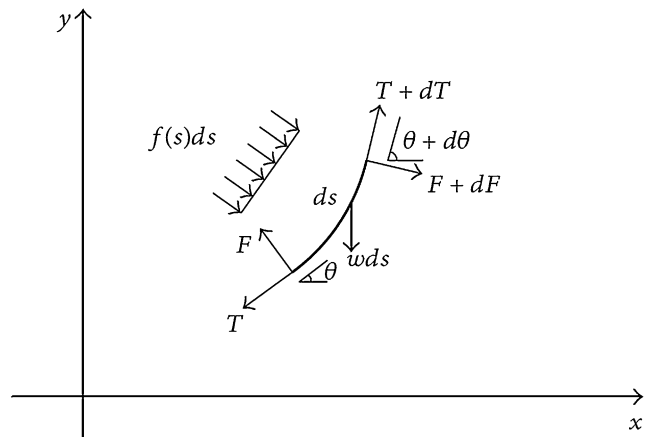
The whole mathematical model (Zeng et al. 2014) for the pipeline abandonment and recovery is the following boundary value problem:

$$\begin{aligned} EI \frac{d^3 \theta}{ds^3} - T \frac{d\theta}{ds} + w \cos \theta &= 0, \\ \frac{dT}{ds} - w \sin \theta &= 0, \\ \theta(0) &= 0, \\ \frac{d\theta}{ds}(0) &= 0, \\ T(0) &= H_0 = T_0 \cos \theta_0, \\ \frac{d\theta}{ds}(L) &= 0. \end{aligned}$$



Installation and Decommissioning, Fig. 6 Mechanical parameters of the pipeline single point lifting and lowering model

Installation and Decommissioning, Fig. 7 Force analysis of a short pipeline segment



Cross-Reference

► [Steel Pipelines and Risers](#)

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Installation of Offshore Pipelines

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Synonyms

[American Petroleum Institute \(API\)](#); [Det Norske Veritas \(DnV\)](#); [Touchdown zone \(TDZ\)](#)

Definition

A submarine pipeline (also known as marine, subsea, or offshore pipeline) is a pipeline that is laid on the seabed or below it inside a trench. In some cases, the pipeline is mostly on-land but in places it crosses water expanses, such as

small seas, straits, and rivers. Submarine pipelines are used primarily to carry oil or gas, but transportation of water is also important. A distinction is sometimes made between a flowline and a pipeline. The former is an intrafield pipeline, in the sense that it is used to connect subsea wellheads, manifolds, and the platform within a particular development field. The latter, sometimes referred to as an export pipeline, is used to bring the resource to shore (https://en.wikipedia.org/wiki/Submarine_pipeline).

Introduction

Offshore pipelines transport hydrocarbon product and other fluids between subsea wells and in-field or onshore processing facilities. Offshore fields are being built in water depth exceeding 10,000 m and technology required for pipeline installation is demanded. Installation of pipelines constitutes some of the most challenging offshore problems.

Offshore pipelines, with an outer diameter ranging from 0.1 to 1.5 m, are commonly installed by laying from a vessel, using an S-lay, J-lay, or reel lay configuration. A less common installation of pipeline is the towing method where the pipeline is assembled on shore.

One major geotechnical issue for offshore pipeline is the assessment of the as-laid pipeline embedment, which is typically found greater than the penetration under static self-weight. Pipeline as-laid embedment has a strong influence on pipeline designs, such as on-bottom stability, lateral buckling, axial friction, heat transfer, and exposure to submarine slides.

The most commonly used installation methods are firstly outlined and then the evaluation of the pipeline as-laid embedment is introduced in this entry.

The most commonly used offshore pipeline installation codes are:

1. DnV OS F101 (Det Norske Veritas).
2. API RP 1111 (American Petroleum Institute).

Pipeline Installation Method

S-Lay

The most common pipeline installation method in shallow waters is the S-lay method. S-lay refers to an installation configuration in which the pipeline starts in a horizontal position on the vessel stinger and forms a characteristic S-shape on the way to the seabed, as shown in Fig. 1(a).

The suspended length of the pipeline is held by tensioners which are usually located on the ramp.

The stinger is a curved structure at the stern of the vessel and provides a controlled-shape transition from horizontal to inclined suspended section. The stinger shape is prescribed by setting the pipeline at designed angles and its length vary with water depth.

The pipeline leaves the stinger at a predefined angle, and the upper curved part of the suspended pipeline is known as overbend. Further down, the pipeline bends in the opposite direction and often the maximum curvature occurs closer to the seabed in the sagbend region.

The curvature in the sagbend is controlled mainly by tension, but excessive tension can be detrimental to the pipeline in the section over the stinger. An overly short stinger could result in excessive bending at the end of the stinger, which may result in buckling of the pipeline. In general, plastic deformation on either the stinger or the sagbend should be avoided in the design.

J-Lay

J-lay is an alternative installation method in which the pipeline leaves the vessel from a nearly vertical position (the actual J-lay tower angles vary between 0° and 15° from the vertical), as shown schematically in Fig. 1(b). A J-shape pipeline configuration is formed on the way down to the seabed. The horizontal tension required to maintain the pipeline configuration are much smaller than required for an S-lay system, and there is no overbend and a large stinger required in S-lay to support the pipeline in deep water.

A short support structure (stinger) below the holding point guides the direction of the pipeline close to the water surface. The tension is to

support the shorter suspended length and to control the pipeline curvature in the sagbend.

The J-lay method is usually adopted in the water depth greater than 500 m. Although J-lay is normally slower in laying speed than the S-lay, it is suitable for deep water region and has been projected to be capable of installing pipelines down to 3350 m of water (Faldini 1999). The positioning of the pipeline can be more precise by J-lay for the touchdown point is not that far behind the vessel comparing with S-lay. An additional benefit is that the lower tension in the pipeline on the seabed translates into shorter free spans.

Pipeline Static Penetration Resistance

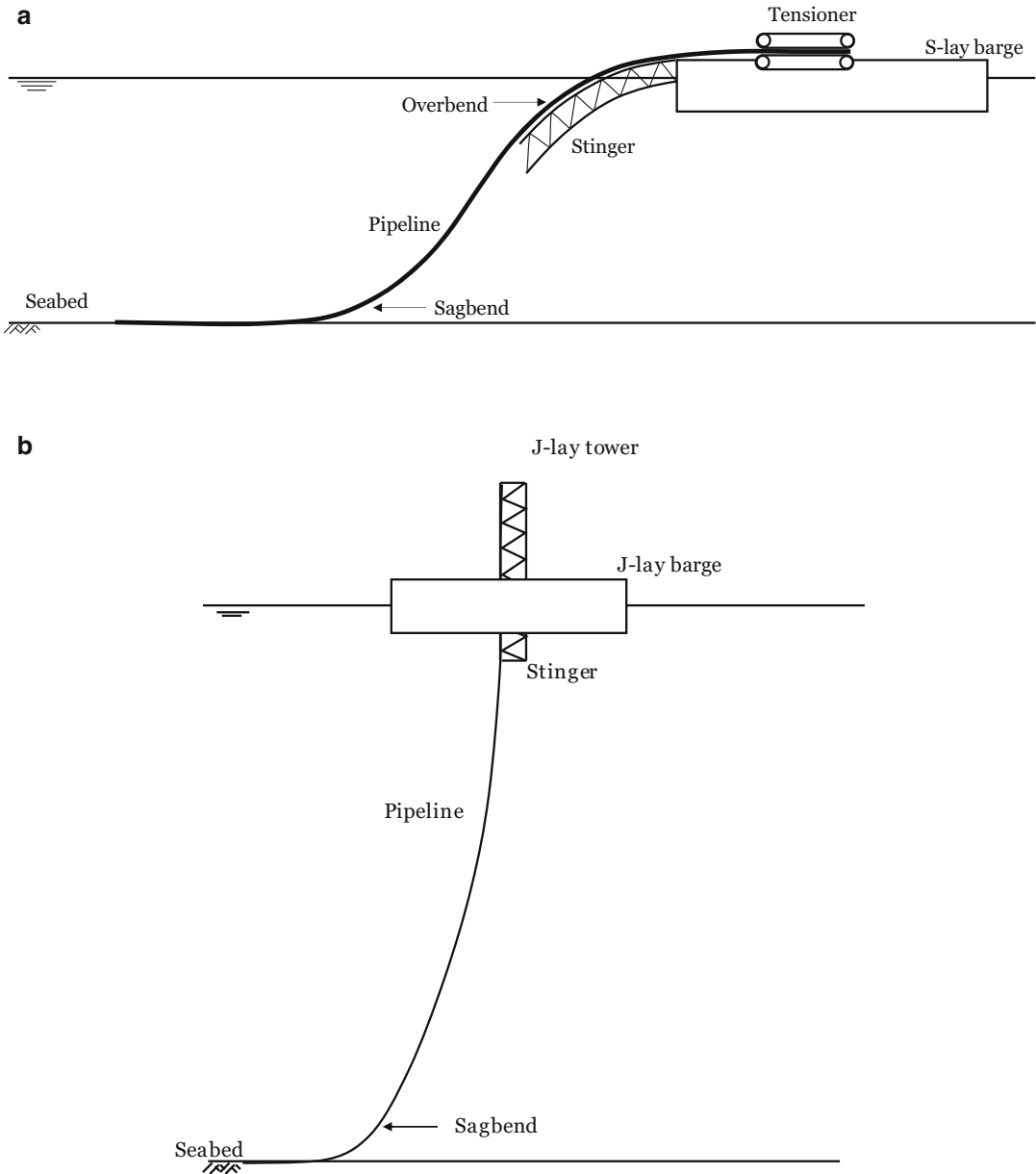
The pipeline as-laid embedment has a significant influence on the lateral stability and also the axial resistance. Therefore, it is important to estimate it as accurate as possible, but there are many factors that complicate this.

It is known that the field observations of the pipeline as-laid embedment are much greater than the estimates in the consideration of the static pipeline self-weight in the calculation. The additional embedment during the laying process results mainly from two mechanisms: (1) the stress concentration in the pipeline touchdown zone (TDZ); (2) soil remolding and strength degradation due to cyclic movements of the pipeline during laying, which are discussed in the section "Lay Effects to Pipeline Penetration." However, the static pipeline penetration resistance should form the basis of the evaluation before more complicated factors are involved in.

Existing methods to evaluate the pipeline embedment can be grouped into empirical correlations and theoretical solutions. The more rigorous theoretical solutions are preferred in the calculations.

Sandy Soil

When the pipeline is laid on the sandy seabed that permeable relative to the pipeline laying rate, generally the as-laid embedment is low and the penetration process is mainly drained.



Installation of Offshore Pipelines, Fig. 1 Schematic diagrams of (a) S-lay and (b) J-lay

The pipe vertical penetration resistance, V_{ult} (force per meter), can be expressed in the following simple relationship:

$$\frac{V_{ult}}{D} = \frac{w}{D} \cdot k_{vp} \quad (1)$$

where k_{vp} is a plastic (secant) stiffness.

The pipeline dynamic motions during the laying process will lead to a lower value of k_{vp} , especially the lateral motions of the pipeline.

Clayey Soil

On the seabed comprising soft clayey soils with low permeability, the pipeline static penetration

resistance can be assessed using an undrained analysis. The vertical bearing capacity of shallowly embedded pipelines on undrained soil with uniform or heterogeneous strength profile can be predicted using plasticity solutions. The pipeline penetration resistance, V_{ult} , can be expressed in the power law relationships, shown as Eq. (2):

$$\frac{V_{ult}}{Ds_u} = a \left(\frac{w}{D} \right)^b \quad (2)$$

where s_u is the shear strength at the pipeline invert and the parameters a and b depend on the pipe-soil roughness and the shear strength gradient with depth. $a = 6.0$ and $b = 0.25$ have been suggested for design calculations (Randolph and White 2008). w is the pipeline invert penetration, and D is the diameter of the pipeline.

Due to the higher density of the seabed sediments relative to water, the contribution of the soil buoyancy to pipeline penetration resistance is significant, especially in the shallow depth. The overall resistance is, therefore, expressed as:

$$\begin{aligned} \frac{V_{ult}}{Ds_u} &= a \left(\frac{w}{D} \right)^b + f_b A_s \left(\frac{w}{D} \right) \frac{\gamma'}{Ds_u} \\ &= N_c + N_b \frac{\gamma'}{Ds_u} \end{aligned} \quad (3)$$

where $N_c = a\hat{w}^b$, $N_b = f_b A_s(\hat{w})$, and $\hat{w} = \frac{w}{D}$. A_s is the area of the pipe lying below mudline level, shown in Fig. 2. γ' is the effective unit weight of the soil. Factor, $f_b = 1.5$ for $\hat{w} = 1.5$, is introduced to account for local heave (White et al. 2010), which will increase the buoyancy force above that due to the pipeline embedment.

The pipe embedded area, A_s , is calculated by:

$$A_s(\hat{w}) = \begin{cases} \frac{D^2}{4} \left[\sin^{-1} \left(\sqrt{4\hat{w}(1-\hat{w})} \right) - 2(1-2\hat{w})\sqrt{\hat{w}(1-\hat{w})} \right] & 0 \leq \hat{w} < 0.5 \\ \pi \frac{D^2}{4} - \frac{D^2}{4} \left[\sin^{-1} \left(\sqrt{4\hat{w}(1-\hat{w})} \right) - 2(2\hat{w}-1)\sqrt{\hat{w}(1-\hat{w})} \right] & 0.5 \leq \hat{w} < 1 \\ \pi \frac{D^2}{4} & 1 \leq \hat{w} \end{cases} \quad (4)$$

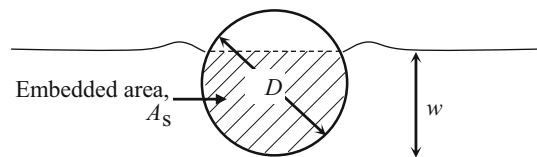
Lay Effects to Pipeline Penetration

Pipeline as-laid embedment is essentially a function of the self-weight of the pipeline and the strength of the seabed, but is much complicated by the laying process. Two effects must be considered in designs:

1. Stress concentration in the TDZ. The pipe-soil contact stress in the vicinity of the touchdown point will exceed the submerged self-weight of the pipe and any contents.
2. Dynamic movements of the pipe during the laying process. Driven by the vessel motions and hydrodynamic loadings of the hanging pipe, the pipeline actually moves laterally and vertically in the laying process, which leads to the local softening of the seabed sediments and push the soil away to the side of the pipe

alignment. The net effects can lead to pipeline penetration that is an order of magnitude greater than estimated from static loading.

It is common to assess the separate effects of the touchdown stress concentration and the dynamic touchdown motion by applying multiplicative adjustments to the static embedment approach described previously. The as-laid pipe weight, p , is multiplied by a concentration factor



Installation of Offshore Pipelines, Fig. 2 Partially embedded pipeline

due to the catenary shape, f_{lay} , to yield the maximum anticipated vertical force between the pipe and soil, $f_{lay}p$. This maximum touchdown load is then used in a conventional bearing capacity type of calculation to assess the pipe embedment, w_{static} , based on the anticipated maximum static touchdown load. Dynamic lay effects are then accounted for by multiplying w_{static} by a dynamic touchdown factor, f_{dyn} , to finally reach the predicted as-laid embedment for lay effects.

Force Concentration in TDZ

During the pipe laying, whether by J-lay or S-lay, the contact stresses between the pipe and the soil in the TDZ is much higher than the submerged pipe self-weight. As schematically shown in Fig. 3, the degree of “over-stress” is affected by the water depth, z_w , the stiffness of the seabed, k , the bending rigidity of the pipeline, EI , and the effective tension in the pipeline in the touchdown zone, T_0 .

From a simple catenary solution, the horizontal tension may be expressed in terms of the water depth, z_w , hang-off angle, φ , and the pipe submerged weight per unit length, W' , as,

$$\frac{T_0}{z_w W'} = \frac{\cos \varphi}{1 - \cos \varphi} \quad (5)$$

Following Pesce et al. (1998), nondimensional controlling parameter, the characteristic length, λ , is introduced. This is expressed as:

$$\lambda = \sqrt{\frac{EI}{T_0}} \quad (6)$$

The seabed stiffness can be expressed in non-dimensional form as:

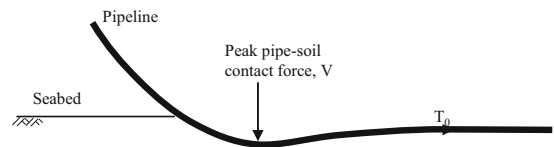
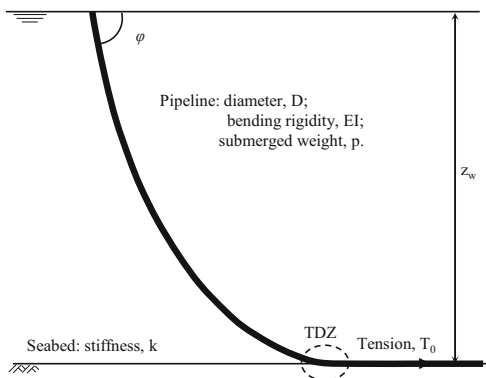
$$K = \frac{\lambda^2}{T_0} k = \frac{EI}{T_0^2} k \quad (7)$$

For horizontal tension of $T_0 > 3\lambda W'$, which is for most pipelines, parametric solutions for the static lay conditions were presented by Randolph and White (2008) from analytical solutions (Lenci and Callegari 2005) and numerical analysis using OrcaFlex all converge to unique design lines. The value of flay can be expressed approximately as:

$$\begin{aligned} \frac{V_{max}}{W'} &\approx 0.6 + 0.4K^{0.25} \\ &= 0.6 + 0.4 \left(\frac{\lambda^2 k}{T_0} \right)^{0.25} \\ &= 0.6 + 0.4 \left(\frac{EI}{T_0^2} k \right)^{0.25} \end{aligned} \quad (8)$$

Lateral Motions

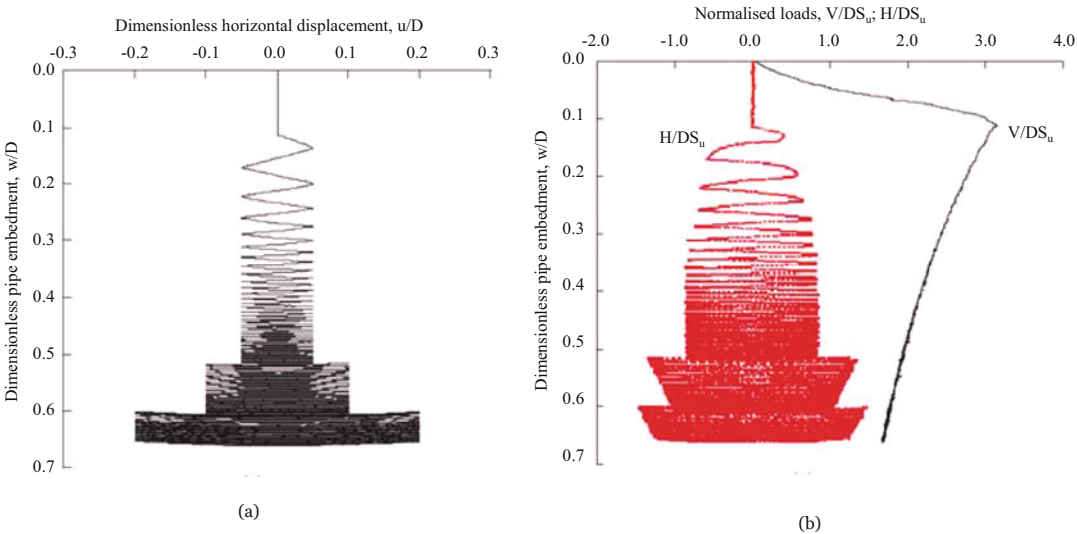
Any lateral movement of the pipeline will generate lateral resistance, and hence a decrease in the available penetration resistance due to the interaction between vertical and horizontal capacity.



Installation of Offshore Pipelines, Fig. 3 Pipeline configuration during laying: (a) complete pipeline and (b) touchdown zone

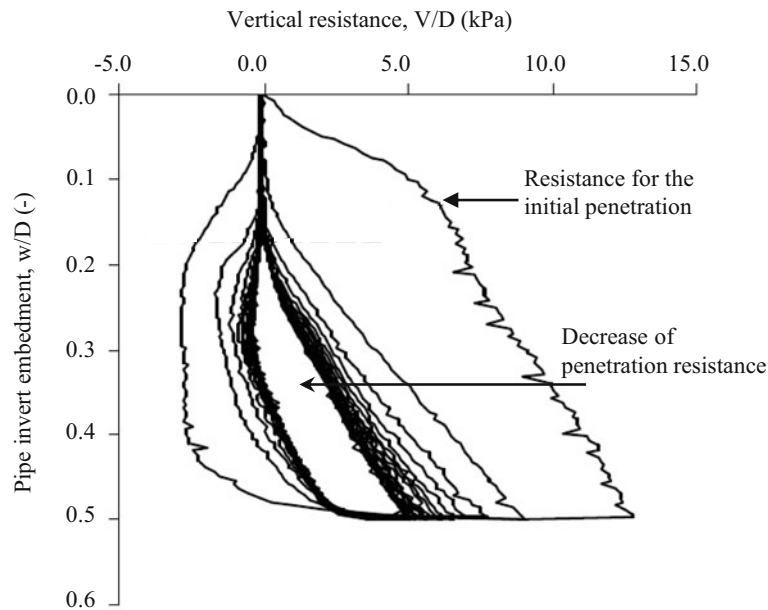
An example response from centrifuge model tests to explore self-embedment under the action of small amplitude cyclic horizontal displacements under undrained conditions is shown in Fig. 4 (Cheuk and White 2011). The test shown used kaolin clay, with a sensitivity, S_p , of 2–2.5, significant embedment

occurs for cyclic horizontal displacements as low as $\pm 5\%$ of the pipe diameter. The vertical load was kept constant, but as the pipe embeds, the normalized load V/DS_u decreases. Eventually, the embedment stabilizes at a value that is more than six times that for static vertical loading.



Installation of Offshore Pipelines, Fig. 4 Pipeline embedment under cyclic horizontal motion: (a) cyclic horizontal displacements and (b) resulting cyclic horizontal loads. (Cheuk and White 2011)

Installation of Offshore Pipelines, Fig. 5 Pipe vertical cyclic response at shallow embedment. (Hodder et al. 2008)



Cyclic Vertical Motions

The component of vertical penetration resistance due to buoyancy become significant if the soil becomes heavily remolded, S_u decreases in Eq. 3. Buoyancy always acts upwards on the pipe, whereas the soil strength contribution opposes the pipe motion, there is always a compressive reaction between the pipe and the soil during shallow vertical cyclic motion as illustrated in Fig. 5.

At deeper embedment, when the soil is weakened by remolding, any soil that is displaced away from the pipe centerline by the dynamic motion will fall back, and the “trenching” mechanism is less effective. Instead, the dynamic motion will remold and weaken the soil, leading to increased settlement under the pipe self-weight.

Based on the back analysis of as-laid surveys, recent design guidance suggests that dynamic effects can be accounted for by multiplying the static embedment by an empirical dynamic embedment factor, f_{dyn} , where this factor is typically in the range of 2–10.

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Installation of Spudcans

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Introduction

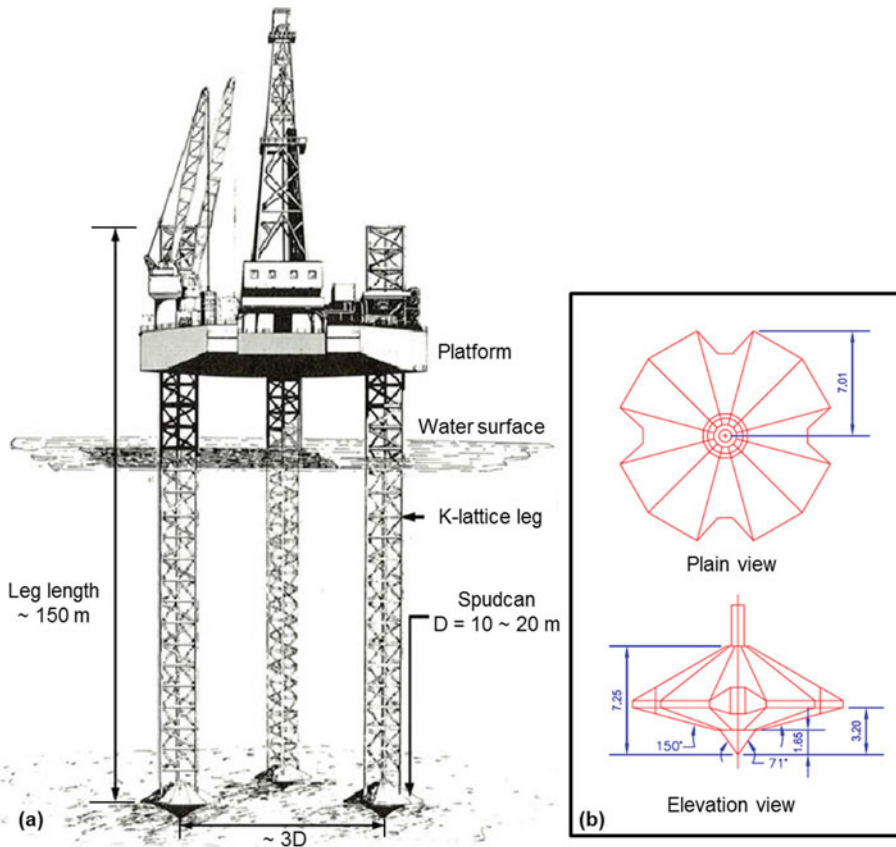
Jack-Up Rig and Spudcan Foundations

Most offshore drilling in shallow to moderate water depths (up to around 150 m) is performed from self-elevating jack-up rigs due to their proven flexibility, mobility, and cost-effectiveness (CLAROM 1993). Today's jack-ups typically consist of a buoyant triangular platform supported by three independent vertically retractable *K*-lattice legs, each resting on a spudcan (Fig. 1a). Spudcans are typically 10–20 m in (equivalent area) diameter and generally saucer-shaped polygonal or quasi-circular foundations (see Fig. 1b).

Installation and Preloading

Prior to commencing jack-up operations, spudcans are routinely proof loaded by static vertical preloading to increase the size of the yield envelope in vertical, moment, horizontal load space and thus ensure they have sufficient reserve capacity in any extreme storm design event (ISO 2015). Typically preloading is accomplished by pumping seawater into holding tanks within the hull. This causes the spudcan foundations to penetrate into the seabed until the load on the spudcan is equilibrated by the resistance of the underlying soil.

Preloading is performed in three ways: (a) simultaneous (all-round) preloading where all legs are incrementally loaded together while the hull is held with minimal draught or air gap;



Installation of Spudcans, Fig. 1 Typical jack-up rig: (a) whole jack-up rig. (After Le Tirant 1979); (b) spudcan geometry. (After Menzies and Lopez 2011)

(b) leg-by-leg preloading where each leg is incrementally loaded separately by sequential filling selected ballast tanks; and (c) combination of both where leg-by-leg preloading is carried out up to 70~80% of the full preload and then all legs are loaded simultaneously to the full preload. The preload is generally maintained for 2~4 h.

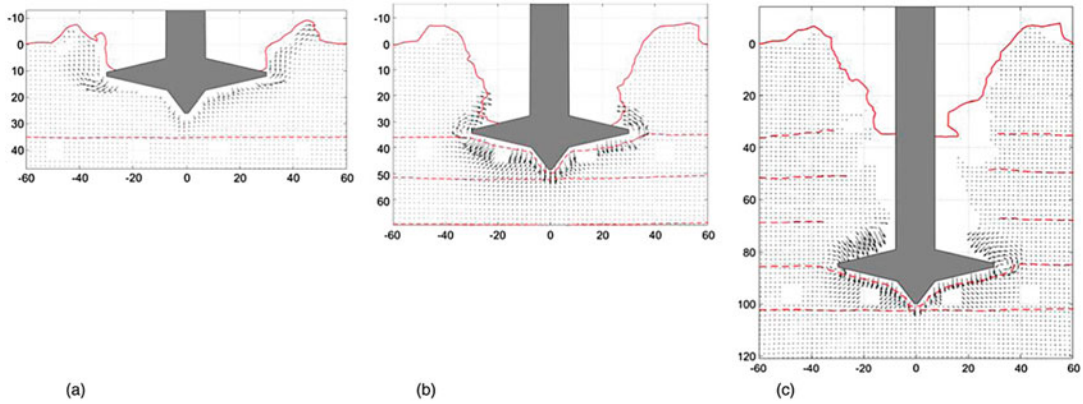
For the preloading and installation stage, critical aspects include accurate prediction of the spudcan penetration resistance profiles including the final resting depth under the full preload and the extent to which a cavity will remain above the spudcan as it penetrates or the depth at which soil will flow back over the spudcan. The aim of this report is to discuss spudcan installation in single-layer clay and sand and layered soils, corresponding design approaches for assessing

spudcan penetration, and potential risks and mitigation measures. For each design approach, all the required soil strength parameters must be obtained from proper site-specific soil investigation data.

Spudcan Installation in Clay

Cavity Depth

Hossain and Randolph (2009a) identified three distinct mechanisms of soil flow around the advancing spudcan: (a) shallow failure with outward and upward flow leading to surface heave and formation of a cavity above the spudcan (Fig. 2a), (b) gradual backflow into the cavity



Installation of Spudcans, Fig. 2 Soil failure mechanisms. (After Hossain et al. 2014a): (a) shallow failure; (b) gradual backflow; (c) deep failure

(Fig. 2b), and (c) deep failure with fully localized flow around the embedded spudcan and unchanged cavity (Fig. 2c). At a certain stage of penetration, soil backflow is initiated, and the continual backflow gradually provides a seal above the spudcan and limits the cavity depth. The subsequent backflow continues below the limiting cavity depth and above the advancing spudcan (i.e., the initial cavity is not filled up by the backflow process; see Fig. 2c). For assessing spudcan penetration response, the depth of soil backflow or limiting cavity depth above the spudcan must first be evaluated.

Hossain and Randolph (2009a) proposed a design chart, along with algebraic expressions, to estimate the limiting cavity depth, H_{cav} , above the penetrating spudcan. For single-layer clay with undrained shear strength increasing with depth, H_{cav} may be estimated according to

$$\frac{H_{cav}}{D} = S^{0.55} - \frac{S}{4} \quad \text{where} \quad (1)$$

$$S = \left(\frac{s_{um}}{\gamma' D} \right)^{1-\frac{k}{\gamma'}}$$

where k/γ' is the ratio of shear strength gradient to soil effective unit weight. This equation is now implemented in the current offshore design guidelines ISO (2015). A deep localized failure mechanism occurs at a penetration of $0.7D \sim 1D$ beyond the onset of backflow.

Design Methods

Several design methods are used in the jack-up industry for predicting spudcan penetration resistance. In this report, three methods will be discussed including (a) the Skempton ISO method (Skempton 1951), (b) the Houlsby-Martin ISO method (Houlsby and Martin 2003), and (c) the mechanism-based Hossain-Randolph approach (Hossain and Randolph 2009a, b; Hossain et al. 2014a).

ISO methods: The vertical bearing capacity, Q_v , of a foundation in clay of uniform shear strength (undrained failure in clay) at a specific depth can be expressed as

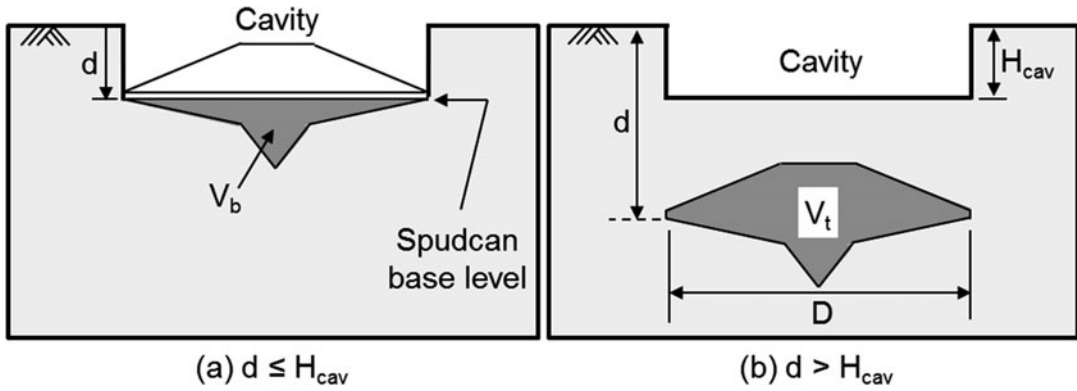
$$Q_v = s_{ui} N_c A + \gamma' (A d + V_b) \quad \text{for } d \leq H_{cav} \quad (2)$$

$$Q_v = s_{ui} N_c A + \gamma' (A H_{cav} + V_t) \quad \text{for } d > H_{cav} \quad (3)$$

where V_t and V_b are as depicted in Fig. 3. For the Skempton ISO method,

$$N_c = 6 \left(1 + 0.2 \frac{d}{D} \right) \leq N_{cd} \quad N_{cd} = 9 \quad (4)$$

which is based on a rough spudcan base ($\alpha = 1$) penetrating clay of uniform undrained shear strength. For typical Gulf of Mexico shear strength gradients and spudcan dimensions, it



Installation of Spudcans, Fig. 3 Schematic presentation of general situation during spudcan penetration: (a) $d \leq H_{cav}$; (b) $d > H_{cav}$

was suggested that the relevant value of s_{ui} should be taken as the average shear strength over a depth of $D/2$ below the widest cross section (referred to herein as the spudcan base level; see Fig. 3).

For the Houlsby-Martin ISO method, N_c factors are those reported by Houlsby and Martin (2003), as tabulated in Annex E of ISO (2015). In addition, the local shear strength at the spudcan base level, s_{u0i} , should be used.

Hossain-Randolph design approach: Hossain and Randolph (2009a) and Hossain et al. (2014a) proposed a “mechanism-based” design approach, based on the soil flow mechanisms observed in model tests and LDFE analysis. For shallow penetrations, less than the limiting cavity depth ($d \leq H_{cav}$), Q_v can be calculated based on the shallow failure mechanism (see Fig. 2a) as

$$Q_v = s_{u0i}N_cA + \gamma'(Ad + V_b) \text{ for } d \leq H_{cav} \quad (5)$$

N_c factors for nonhomogeneous clay are expressed as

$$N_c = N_{c0} \left[1 + 0.161 \left(\frac{kD}{s_{u0i}} \right)^{0.8} / \left(1 + \frac{d}{D} \right)^2 \right] \text{ for smooth spudcan base } (\alpha = 0) \quad (6)$$

$$N_c = N_{c0} \left[1 + 0.191 \left(\frac{kD}{s_{u0i}} \right)^{0.8} / \left(1 + \frac{d}{D} \right)^{1.5} \right] \text{ for rough spudcan base } (\alpha = 1) \quad (7)$$

where N_{c0} is the bearing capacity factor for uniform clay, given by

$$N_{c0} = 5.69 + \frac{d}{0.2D} \left(1 - \frac{d}{2.85D} \right) \leq 9.2 \text{ for smooth spudcan base } (\alpha = 0) \quad (8)$$

$$N_{c0} = 6.05 + \frac{d}{0.22D} \left(1 - \frac{d}{3.65D} \right) \leq 10.2 \text{ for rough spudcan base } (\alpha = 1) \quad (9)$$

For deep penetration in excess of H_{cav} , back-flow occurs above the advancing spudcan, but leaving the original cavity largely unchanged (see Fig. 2b), and a deep localized flow-round mechanism is finally developed (see Fig. 2c). For $d > H_{cav}$, the vertical bearing capacity, Q_v , can be calculated according to

$$Q_v = s_{u0i}N_{cd}A + \gamma'V_t \text{ for } d > H_{cav} \quad (10)$$

The N_{cd} values were found to merge into a narrow band, regardless of strength non-

homogeneity (kD/s_{umi}) and spudcan base roughness (α), finally reaching a limiting ultimate value once the deep failure mechanism was established. A simple expression for N_{cd} was proposed, given by

$$N_{cd} = 10 \left(1 + 0.065 \frac{d}{D} \right) \leq 11.3 \quad (11)$$

This expression takes into account both the transitional stage between the onset of backflow and attainment of deep failure and the final stage with deep localized failure where the bearing capacity factor becomes independent of penetration depth.

These expressions were developed based on the large deformation finite element (LDFE) analyses results considering a simple elastic-perfectly plastic soil model. However, in practice, natural soils exhibit strain-rate dependency and also soften as they are sheared and remolded. Therefore, Hossain and Randolph (2009b) proposed an adjustment factor to modify the just discussed

design approach developed on the basis of ideal elastic-perfectly plastic soil behavior as

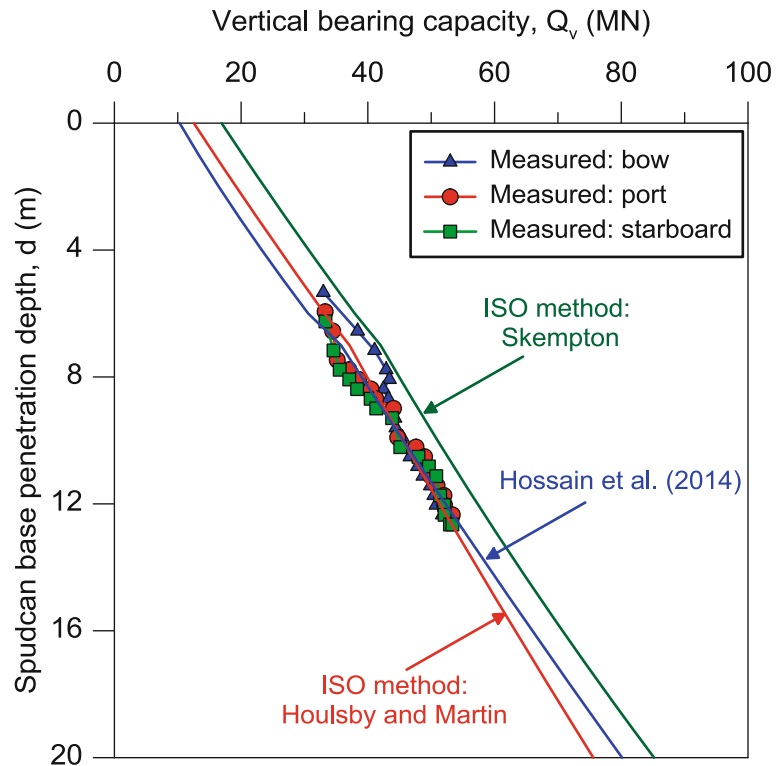
$$\frac{N_{c,adjust}}{N_c} \text{ or } \frac{N_{cd,adjust}}{N_{cd}} = \frac{(1 + R_b \mu)}{S_t} \left(1 + (S_t - 1) e^{-7.2/\xi_{95}} \right) \quad (12)$$

where N_c and N_{cd} are the capacity factors from Eqs. 6~9 and 11. The rate coefficient R_b ranges from 1.47 (rapid penetration during initial stages of preloading) to 0.77 (slower penetration during final stages) and essentially drops to zero (negligible rate effect) during additional settlement under the final preload.

If the values for pertinent parameters are available, bearing capacity factors, N_c and N_{cd} , may be adjusted following Eq. 12. Alternatively, in the absence of site-specific data, a 20% reduction is suggested to be applied to the vertical penetration resistance computed using Eqs. 6~9 and 11 (Menzies and Roper 2008; Hossain and Randolph 2009b). Example prediction profiles for a typical site in the Gulf of Mexico are shown in Fig. 4.

Installation of Spudcans,

Fig. 4 Example of prediction profiles from Gulf of Mexico site. (After Hossain et al. 2014a)



Spudcan Installation in (silica) Sand

Cavity Depth

In sand, soil backflow (flow from beneath the spudcan, around the spudcan edge, and finally above the spudcan) and inflow (caving in) into the cavity above the spudcan start to occur immediately after the widest cross section of the spudcan is below the ground surface. The backfill/infill sand would usually be assumed to rest at the angle of repose, which may be taken as approximately the critical state friction angle ϕ_{cv} (see Fig. 5). The cavity side is not vertical (or quasi-vertical) either. Instead, the slope starts from a distance of $0.3\sim 0.4D$ away from the vertical line. The volume of the backfill/infill sand should therefore be calculated accordingly.

Design Methods

Several design methods are used in the jack-up industry for predicting spudcan penetration resistance in sand. Three methods will be discussed including (a) the Martin ISO method (Martin 2003), (b) the Cassidy-Houlsby ISO method (Cassidy and Houlsby 2002), and (c) the White et al. approach (White et al. 2008).

ISO methods: Evaluating the penetration of a spudcan in silica sand in ISO (2015) is based on a drained bearing capacity calculation (no excess pore pressure is generated) and expressed as

$$Q_v = \gamma'(0.5DN_\gamma A + V_b) \quad \text{for } d \leq 0 \quad (13)$$

$$Q_v = \gamma'(0.5DN_\gamma A + dN_q d_q A + V_t - V_{soil}) \quad \text{for } d > 0 \quad (14)$$

where V_{soil} is the volume of the soil above the spudcan. For the Martin ISO method, N_γ and N_q values from Martin's ABC software should be used. These bearing capacity factors are for flat-based, rough circular footing. Cassidy and Houlsby (2002) tabulated N_γ factors as a function of cone angle, cone base roughness, and peak friction angle. For the Cassidy-Houlsby ISO method, N_q values should be picked from Martin's ABC. Depth factor of $d_q = 1 + 2 \tan \phi'(1 - \sin \phi')^2(d/D)$, as proposed by Vesic (1975), can be used.

White et al. approach: For the White et al. approach, the scale effect (i.e., essentially the stress-level effect) and progressive failure should be taken into account in order to select an operative friction angle for medium dense to dense sand. An iterative process is needed which involves the following equations together with Eq. 13 or 14

$$I_R = D_r(Q_{crushing} - \ln p') - 1 \quad (15)$$

$$\phi' = \phi'_{cv} + m I_R \quad (16)$$

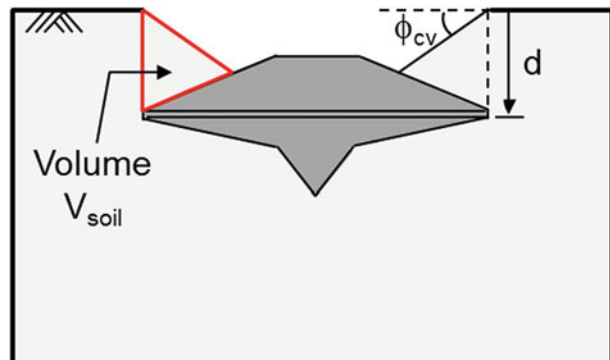
$$p' = \lambda q_{nom} \quad (17)$$

$$p' = \left(\frac{1}{1 + \tan^2 \phi'} \right) \left(\frac{1}{1 + \sin \phi'} \right) \times \frac{q_{nom} + 3\gamma'd}{4} \quad (18)$$

where I_R is the relative dilatancy index, D_r is the relative density of sand, $Q_{crushing}$ is the particle crushing strength on a natural log scale, p' is the mean stress level at failure, m and λ are con-

Installation of Spudcans,

Fig. 5 Backfill/infill in sand



stants, and $q_{\text{nom}} = Q_v/(\pi D^2/4)$ is the nominal bearing pressure. The iterative approach is most simply achieved by assuming that $\phi' = \phi_{\text{cv}}$, picking N_γ and N_q values from Martin's ABC, and then applying Eqs. 13 or 14, 15, 16, and 17 in sequence, to find a new estimate of ϕ' . This process is repeated until satisfactory convergence is achieved. White et al. (2008) suggested $m = 0.85$ and 0.7 for equivalent enclosed cone angle of 150° and 130° , respectively, and $\lambda = 0.15$ based on back-analysis of the centrifuge test data.

Overestimation of penetration resistance:

For silica sands, using an appropriate mobilized friction angle (based on the relative density), overestimation of penetration resistance, in comparison with reported field data, often occurs. This is exacerbated when a relatively thin sand layer is sandwiched between two clay layers. This may be partly due to less than fully drained conditions in the sand layer, but the dominant factor is that the thickness of the layer relative to the spudcan diameter is not sufficient to mobilize the full capacity of the layer during penetration of a 10–20 m diameter spudcan. To cope with this issue, (i) SNAME (2008) suggests, based on experience with measured field data, reducing the (assumed) mobilized friction angle in the sand by $5\text{--}20^\circ$ (as low as 12° depending on the sand relative density, D_r), thus reducing the estimated penetration resistance; (ii) InSafeJIP (2011) suggests using a mobilization factor (multiplicative to the first term of Eq. 13 and first and second terms of Eq. 14; right-hand side) of 0.25–0.5. Lower values would be applicable to more compressible materials (e.g., carbonate sands) and higher values for stiffer materials, but little quantitative evidence exists.

Spudcan Installation in Layered Soils with Potential for Punch-Through: Stiff-Over-Soft Clay

Cavity Depth

Hossain and Randolph (2010a, b) identified mechanisms of soil flow around the spudcan advancing in stiff-soft clay deposits (discussed later). The cavity depth can be assessed as

$$\frac{H_{\text{cav}}}{D} \approx 1.4 \left\{ \left(\frac{s_{u1i}}{\gamma'_2 D} \right) \left(\frac{t_1}{D} \right) / \left(1 + \frac{k_2 D}{s_{u2si}} \right) \right\}^{0.5} \quad (19)$$

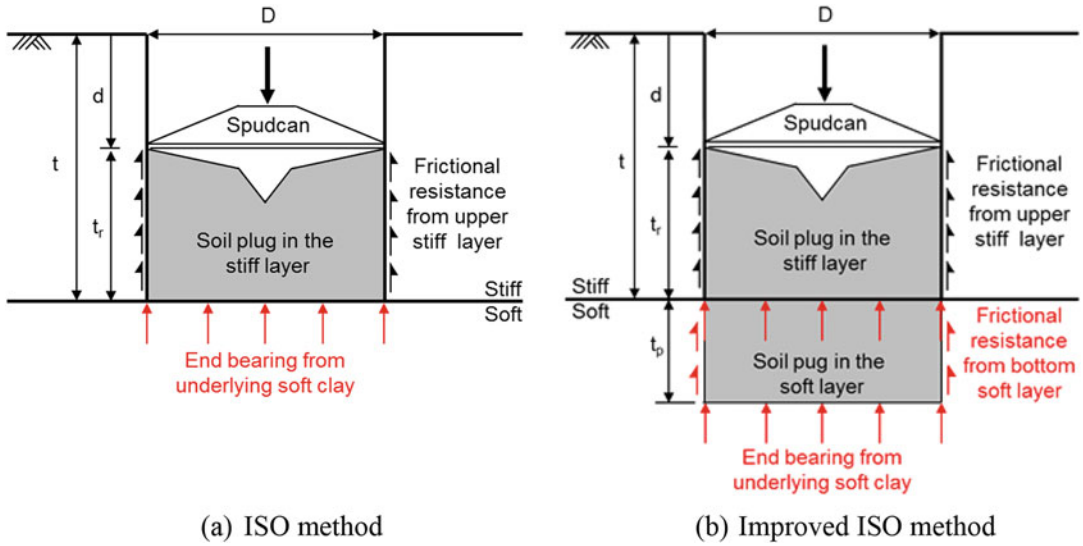
where s_{u1i} is the intact undrained shear strength of the first-layer clay, s_{u2si} is the intact undrained shear strength of the second-layer clay at the first–second layer interface, t_1 is the thickness of the first-layer clay, and γ'_2 and k_2 are the effective unit weight and strength gradient of the second-layer clay, respectively.

Design Methods

Two methods will be discussed including (a) the punching shear ISO method (Brown and Meyerhof 1969) and (b) the improved ISO method (Zheng et al. 2016). In both methods, penetration resistance is contributed by two main components: (a) end bearing resistance in the underlying soft clay layer and (b) frictional resistance in the upper stiff clay layer or in both layers.

ISO method: This method is applicable only for the stiff clay layer. For the punching shear ISO method, a cylindrical plug of diameter D is assumed between the base of the advancing spudcan and the stiff-soft layer interface. The plug base is assumed to be fixed at the interface regardless of the spudcan penetration, i.e., (i) the height of the plug reduces with the progress of spudcan penetration in the stiff layer ($t_r = t_1 - d$), and (ii) the movement of the plug in the underlying soft layer and the deformation of the layer boundary are restricted. These are the major shortcomings of this method. Figure 6a shows the analytical model. The vertical bearing capacity, Q_v , of a foundation in uniform stiff clay over uniform soft clay at a specific depth d can be expressed as

$$\begin{aligned} Q_v &= A \left\{ 4\eta \frac{t_r}{D} s_{u1i} + N_{csu2i} \right\} \\ &\quad + \gamma'_1 (Ad + V_b) \\ &\leq N_{csu1i} A + \gamma'_1 (Ad + V_b) \text{ for } d \\ &\leq H_{\text{cav}} \end{aligned} \quad (20)$$



Installation of Spudcans, Fig. 6 Conceptual models for spudcan at punch-through in stiff-over-soft clays: (a) ISO method; (b) improved ISO method

$$\begin{aligned}
 Q_v &= A \left\{ 4\eta \frac{t_r}{D} s_{u1i} + N_c s_{u2i} \right\} \\
 &\quad + \gamma'_1 (A H_{cav} + V_t) \\
 &\leq N_c s_{u1i} A + \gamma'_1 (A H_{cav} + V_t) \quad d \\
 &> H_{cav}
 \end{aligned} \quad (21)$$

The first term represents frictional resistance along the surfaces of the cylindrical plug, and the second term is due to end bearing of the underlying clay at the interface. It is assumed full shear strength was not mobilized around the periphery of the plug and hence a factor η is applied. A value of $\eta = 0.75$ (suggested by Brown and Meyerhof 1969) ~ 0.9 (proposed by Hanna and Meyerhof 1980) can be used.

For stiff-over-soft clay, $A \left\{ 4\eta \frac{t_r}{D} s_{u1i} + N_c s_{u2i} \right\} + \gamma'_1 (A d + V_b)$, N_c should be picked from Eq. 4 using $d/D = t_r/D$ (i.e., depth of the plug base) regardless of spudcan penetration depth. This is because end bearing is assumed to be mobilized at the interface. For single-layer clay, $N_c s_{u1i} A + \gamma'_1 (A d + V_b)$, N_c should be

picked from Eq. 4 using actual penetration depth of the spudcan base, i.e., $d/D = d/D$.

For bottom layer with nonuniform undrained shear strength, an average value over a depth of $0.5D$ can be used. For the bottom clay layer, the penetration resistance should be calculated using the single-layer method as discussed previously. The soil plug carried down with the spudcan from the upper layer, and corresponding contribution to the penetration resistance, is neglected.

Improved ISO method (Zheng et al. 2016): The improved ISO method was proposed by Zheng et al. (2016). This method is applicable for the penetration resistance of stiff and underlying soft clay layers and both uniform and nonuniform clay deposits. The depth of punch-through and peak resistance basically follows the original ISO method, but adding a plug (of height t_p) in the bottom layer and shifting the fictitious footing position to the base of the added plug, as illustrated in Fig. 6b. For spudcan penetration in uniform stiff-over-soft clay deposit ($s_{u2} = s_{u2i}$), it can be expressed as

$$Q_v = A \left\{ 3 \frac{t_r}{D} s_{u1i} + \min \left[6 \left(1 + 0.2 \frac{t}{D} \right), 9.0 \right] s_{u2} + \gamma'_1 (Ad + V_b) + 4.2 \frac{t_p}{D} s_{u2i} \right\} \text{ for } d \leq N_c s_{u1i} A + \gamma'_1 (Ad + V_b) \leq H_{cav} \quad (22)$$

$$Q_v = A \left\{ 3 \frac{t_r}{D} s_{u1i} + \min \left[6 \left(1 + 0.2 \frac{t}{D} \right), 9.0 \right] s_{u2} + \gamma'_1 (AH_{cav} + V_t) + 4.2 \frac{t_p}{D} s_{u2i} \right\} \text{ for } d > H_{cav} \leq N_c s_{u1i} A + \gamma'_1 (AH_{cav} + V_t) \quad (23)$$

To keep a simple form of the formulas, Eqs. 22 and 23 are also used to predict the peak resistance for spudcan penetration in uniform-over-nonuniform deposit (i.e., $k > 0$), taking s_{u2} in Eqs. 22 and 23 as the average strength over the depth of $D/2$ below the layer interface. Based on the calibrated data, the equivalent soil plug thickness t_p can be approximately given as

$$\frac{t_p}{t_r} = 0.53 \left(\frac{s_{u2i}}{s_{u1i}} \right)^{-0.5} \left(\frac{t_r}{D} \right)^{-1} \left[3 - \left(1.5 - \frac{kD}{s_{u2i}} \right)^2 \right]^{0.5t_r/D} \text{ for } kD/s_{u2i} \leq 3 \quad (24)$$

For spudcan penetration resistance in the bottom layer, a simplified single-layer approach is used (see Eq. 10), which implicitly incorporates the effect of the soil plug and corresponding additional resistance in the deep bearing capacity factor N_{cd} . The value of N_{cd} can be expressed as

$$N_{cd} = 9.8 + 1.3 \left(\frac{s_{subs}}{s_{ut}} \right)^{-1} \text{Min} \left[\left(\frac{t}{D} \right)^{0.5}, 1.0 \right] \left(1 + \frac{kD}{s_{subs}} \right)^{-1} \quad (25)$$

Spudcan Installation in Layered Soils with Potential for Punch-Through: Sand Over Clay

Cavity Depth

The volume of the soil above the spudcan, V_{soil} , can be calculated in a manner similar to single-layer sand.

Design Methods

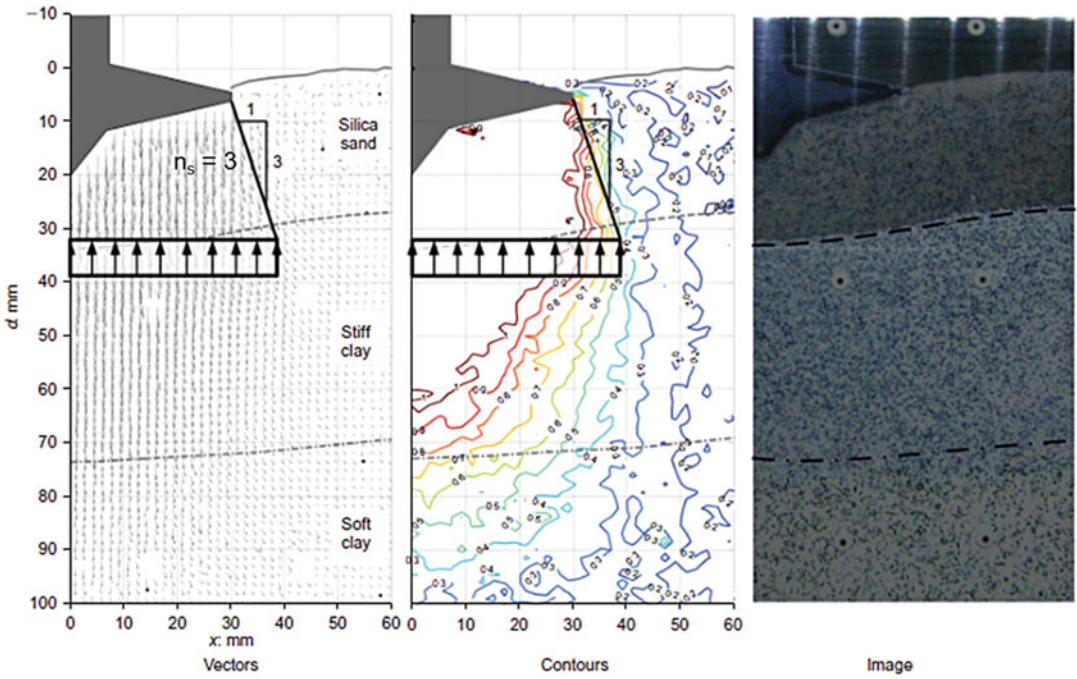
Three methods will be discussed including (a) the load spread ISO method, (b) the punching shear ISO method, and (c) the mechanism-based Lee et al. and Hu et al. approaches (Lee et al. 2013a, b; Hu et al. 2015, 2018).

ISO methods: These methods are applicable only for the sand layer. The base is assumed to be fixed at the sand-clay layer interface regardless of the spudcan penetration. In the load spread method, the bearing capacity is calculated by considering a fictitious footing at the interface between the sand and clay layers with an area $A_p = \pi D_p^2/4$, where $D_p = D + 2t_r/n_s$. The applied load is spread with a gradient of 1(horizontal): n_s (vertical), with the load spread factor $n_s = 3$ to 5 recommended by ISO (2015). Hossain (2014) also confirms from centrifuge observations that, in silica sand, n_s can be taken as 3 (see Fig. 7). The analytical model is shown in Fig. 8. The vertical bearing capacity, Q_v , of a foundation in sand over clay at a specific depth d can be expressed as

$$Q_v = A_p N_c s_{u2} + \gamma'_1 (Ad + V_b) \leq \gamma'_1 (0.5DN_{\gamma}A + V_b) \text{ for } d \leq 0 \quad (26)$$

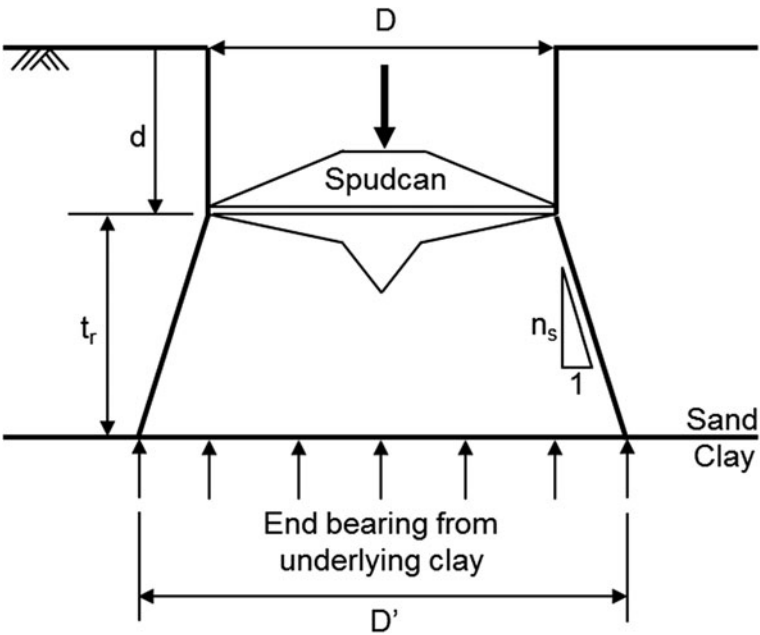
$$Q_v = A_p N_c s_{u2} + \gamma'_1 (Ad + V_t - V_{soil}) \leq \gamma'_1 (0.5DN_{\gamma}A + dN_q d_q A + V_t - V_{soil}) \text{ for } d > 0 \quad (27)$$

In this method, the contribution from the frictional resistance around the plug was neglected.



Installation of Spudcans, Fig. 7 Soil failure pattern for load spread factor. (After Hossain 2014)

Installation of Spudcans, Fig. 8 Conceptual model for spudcan at punch-through in sand-over-clay deposits (ISO method)



For the punching shear ISO method, similar to stiff-over-soft clay, a cylindrical plug of diameter D is assumed between the base of the advancing

spudcan and the underlying layer interface, with the base fixed at the sand-clay layer interface regardless of the spudcan penetration. As a

cylindrical plug is assumed, dilatancy of the sand is therefore neglected (dilation angle, $\psi = 0^\circ$). The vertical bearing capacity, Q_v , of a foundation in sand over clay at a specific depth d can be expressed as

$$Q_v = A \left(2t_r \gamma'_1 (t_r + 2d) K_s \frac{\tan \phi'}{D} + N_c s_{u2si} \right) + \gamma'_1 (Ad + V_b) \leq \gamma'_1 (0.5DN_\gamma A + V_b) \text{ for } d \leq 0 \quad (28)$$

$$Q_v = A \left(2t_r \gamma'_1 (t_r + 2d) K_s \frac{\tan \phi'}{D} + N_c s_{u2si} \right) + \gamma'_1 (Ad + V_t - V_{soil}) \leq \gamma'_1 (0.5DN_\gamma A + dN_q d_q A + V_t - V_{soil}) \text{ for } d > 0 \quad (29)$$

where K_s is the punching shear coefficient and ϕ' is the effective angle of internal friction. Theoretically K_s should lie in between at rest earth

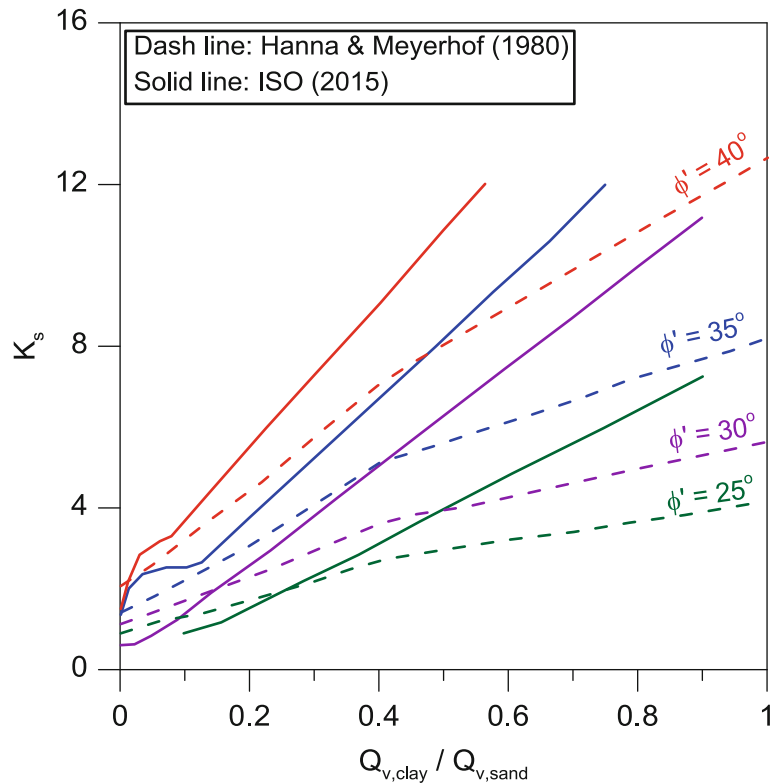
pressure coefficient (K_0) and passive earth pressure coefficient (K_p). The values of K_s suggested by ISO (2015) and Hanna and Meyerhof (1980) are given in Fig. 9, as a function of ϕ' and $Q_{v,clay}/Q_{v,sand}$ (ratio of single-layer response for a surface spudcan on the clay layer and that on the sand layer).

N_c should be picked from Eq. 4. For bottom layer with nonuniform undrained shear strength, an average value over a depth of $D/2$ can be used. For the bottom clay layer, the penetration resistance should be calculated using the single-layer approach discussed previously.

Lee et al. and Hu et al. approach: Lee et al. (2013a, b) and Hu et al. (2015, 2018) proposed a mechanism-based design approach based on the soil flow mechanisms observed in model tests and LDFE analyses. The approach simplified the spudcan penetration profile as a combination of the peak resistance and post-peak resistance in the sand layer, q_{peak} and $q_{post-peak}$, and the resistance in the clay layer, q_{clay} . q_{peak} is the sum of the frictional resistance in the sand, the bearing

Installation of Spudcans,

Fig. 9 Bearing capacity ratio versus coefficient of punching shear



capacity of the underlying clay, and the weight of the sand frustum, and it is expressed as

$$q_{\text{peak}} = (N_{c0}s_{\text{umi}} + q_0 + 0.12\gamma'_s H_s) \left(1 + \frac{1.76H_s}{D} \tan \psi\right)^{E^*} + \frac{\gamma'_s D}{2 \tan \psi (E^* + 1)} \left[1 - \left(1 - \frac{1.76H_s}{D} E^* \tan \psi\right) \left(1 + \frac{1.76H_s}{D} \tan \psi\right)^{E^*}\right] \quad (30)$$

where N_{c0} is the bearing capacity factor for clay at the base of a circular foundation, which

$$q = \frac{\gamma'_s D}{E} + \left\{ \left[\left(14.8 \frac{d - 0.1H_s}{D} + 10.6\right) s_{\text{umi}} + q_0 + \gamma'_s H_s + (\gamma'_c - \gamma'_s)(d - 0.1H_s) - \frac{\gamma'_s D}{E} \right] \exp \left[\frac{E(H_s - d)}{D} \right] \right\} \quad (31)$$

where γ'_c is the effective unit weight of clay and E is also a parameter to simplify the algebra (Hu et al. 2018).

For the resistance in the clay layer, the bearing capacity is expressed as

$$\begin{aligned} q_{\text{clay}} &= N_c s_{u0i} + H_{\text{plug}} \gamma'_c \\ &= N_c s_{u0i} \\ &\quad + 0.9H_s \gamma'_c \quad \left(0.16 \leq \frac{H_s}{D} \leq 1.00\right) \end{aligned} \quad (32)$$

where s_{u0i} is the clay shear strength at the lowest level of the spudcan's widest cross-sectional area and H_{plug} is the height of the sand plug. The bearing capacity factor N_c was summarized from a sand overlying clay centrifuge testing database and LDFE analyses of the full penetration process, incorporating various footing shapes and soil conditions.

Spudcan Installation in Layered Soils with Potential for Squeezing: Soft Clay-Over-Strong Layer

Cavity Depth

If the thickness of the soft clay layer is relatively thin and hence the (limiting cavity depth in single

is obtained from Houlsby and Martin (2003); q_0 is the effective overburden pressure; and E^* is a parameter to simplify the algebra. The depth of the peak resistance in the sand layer is taken as $0.12H_s$, where H_s is the sand layer thickness.

For assessing spudcan penetration resistance after the peak failure in the top sand layer and before penetrating into the bottom clay layer, an analytical model was proposed as

layer of that clay + limiting squeezing depth) < (thickness of the soft clay layer), soil backflow can occur earlier due to the influence of squeezing (Hossain 2014). This means H_{cav} in soft-over-strong soils may be lower compared to that on single-layer soft clay.

Design Methods

For assessing spudcan penetration in soft clay-over-strong layer, four methods will be discussed including (a) the squeezing ISO method (Meyerhof and Chaplin 1953; Brown and Meyerhof 1969), (b) the Merifield-Nguyen method (Merifield and Nguyen 2006), (c) the Meyerhof and Hanna method (Meyerhof 1974; Meyerhof and Hanna 1978), and (d) the CLAROM method (CLAROM 1993).

Squeezing ISO method: This method is applicable only for the overlying soft clay layer. A uniform strength is considered for the top layer, and the underlying layer is assumed to be rigid. The penetration resistance in the soft clay layer can be assessed according to

$$\begin{aligned} Q_v &= A \left\{ 0.33 \frac{D}{t_r} s_{u1i} + (N_c - 1) s_{u1i} \right\} \\ &\quad + \gamma'_1 (Ad + V_b) \\ &\geq N_c s_{u1i} A + \gamma'_1 (Ad + V_b) \quad \text{for } d \\ &\leq H_{\text{cav}} \end{aligned} \quad (33)$$

$$\begin{aligned}
Q_v &= A \left\{ 0.33 \frac{D}{t_r} s_{uli} + (N_c - 1) s_{uli} \right\} \\
&\quad + \gamma'_1 (A H_{cav} + V_t) \\
&\geq N_c s_{uli} A + \gamma'_1 (A H_{cav} + V_t) \text{ for } d \\
&> H_{cav}
\end{aligned} \quad (34)$$

The values of N_c should be picked from Eq. 4 using actual penetration depth of the spudcan base, i.e., $d/D = d/D$. For soft layer with a non-uniform undrained shear strength, an average value over a depth of $D/2$ can be used. However, averaging should be restricted within the soft clay layer; otherwise near the first–second layer interface, a significantly higher shear strength would be picked. Clearly, Eqs. 33 and 34 are not a function of the thickness and properties of the underlying strong layer. This is a limitation of this approach.

Merifield and Nguyen method: Recently Merifield and Nguyen (2006) and Meyer et al. (2015) carried out small strain finite element analyses on a perfectly rough-based circular surface ($d/D = 0$) footing in soft-over-stiff clay. Merifield and Nguyen (2006) presented the values of modified bearing capacity factor both numerically in tables and graphically in figures. Best fit through Merifield and Nguyen's (2006) data provides

$$\begin{aligned}
N &= 0.009 \left(\frac{D}{t_r} \right)^2 - 0.018 \frac{D}{t_r} \\
&\quad + N_{cs} \text{ for } s_{ubi}/s_{uti} \\
&= 1.25
\end{aligned} \quad (35a)$$

$$\begin{aligned}
N &= 0.033 \left(\frac{D}{t_r} \right)^2 - 0.046 \frac{D}{t_r} \\
&\quad + N_{cs} \text{ for } s_{ubi}/s_{uti} \\
&= 2.0
\end{aligned} \quad (35b)$$

$$\begin{aligned}
N &= 0.036 \left(\frac{D}{t_r} \right)^2 - 0.061 \frac{D}{t_r} \\
&\quad + N_{cs} \text{ for } s_{ubi}/s_{uti} \\
&= 2.5
\end{aligned} \quad (35c)$$

where $N_{cs} = 6.05$. For an embedded footing ($d/D > 0$), Eq. 35 can be extended using N_c instead of N_{cs} .

Meyerhof and Hanna method: The above discussion is for soft-over-stiff clay or a rigid layer. In a thin weak-over-strong layer deposit of any soils, Meyerhof (1974) and Meyerhof and Hanna (1978) suggested that the bearing capacity of a flat-based circular footing can be calculated according to

$$q_u = q_{ut} + (q_{ub} - q_{ut}) \left(1 - \frac{t_r/D}{H_f/D} \right) \quad (36)$$

where $q_{u,t}$ is the bearing capacity of the footing at the penetration depth in the top layer, $q_{u,b}$ is the bearing capacity of the bottom layer at the interface, and $H_f = 1$ for circular footing regardless of soil (i.e., either sand or clay).

CLAROM method: CLAROM (1993) somewhat blended the methods proposed by Meyerhof (1974) and Meyerhof and Hanna (1978) and Brown and Meyerhof (1969) and accounted for the effect of spudcan bottom profile as

$$q_u = q_{ut} + (q_{ub} - q_{ut}) \left(\frac{A_b}{A} \right) + \gamma'_t \frac{V}{A} \quad (37)$$

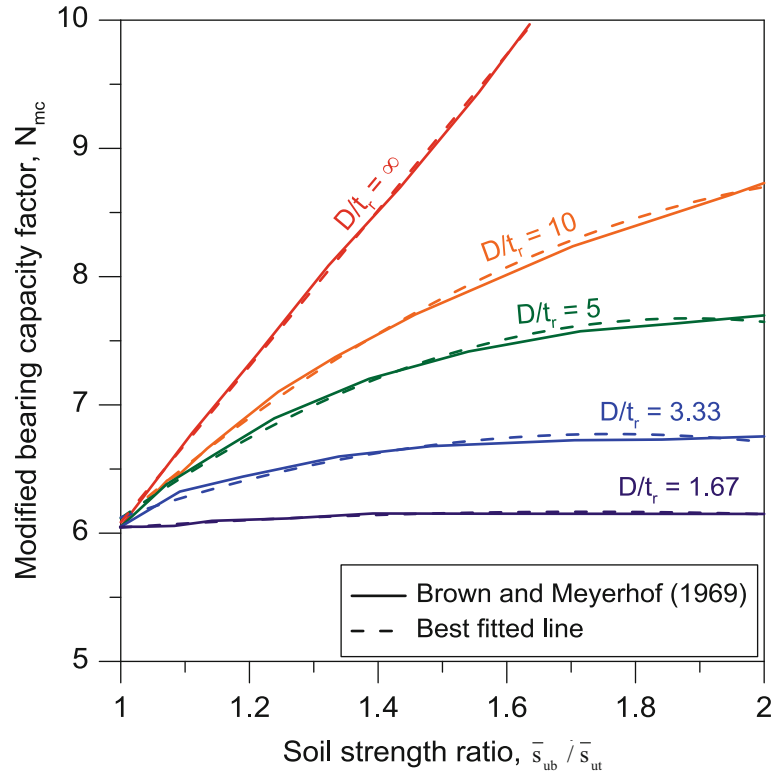
where A_b is the plan area of the spudcan bottom profile embedded in the bottom layer and V is the volume of the spudcan embedded in soil. $A_b = 0$ if the spudcan does not penetrate into the bottom layer.

For soft-over-stiff clay,

$$q_{ut} = N_{mc} \left(1 + 0.2 \frac{d}{D} \right) \bar{s}_{uti} \quad (38a)$$

$$q_{ub} = N_{cs} \left(1 + 0.2 \frac{t}{D} \right) \bar{s}_{ubi} \quad (38b)$$

where N_{mc} should be taken from Fig. 10 (Brown and Meyerhof 1969) using $\bar{s}_{ubi}/\bar{s}_{uti}$ instead of s_{ubi}/s_{uti} , $N_{cs} = 6.05$, $\bar{s}_{uti}$ is the average undrained shear strength of soft clay over thickness t_r , and $\bar{s}_{ubi}$ is the average undrained shear strength of stiff clay over thickness of $D_b/2$ (D_b is the spudcan diameter embedded in the bottom layer) under the expression of q_{ub} and over the thickness of $D/2$ under the interface for calculating $\bar{s}_{ubi}/\bar{s}_{uti}$ for picking N_{mc} from Fig. 10. This average is necessary to take into account the strength increasing with depth.

Installation of Spudcans,**Fig. 10** Modified bearing capacity factors. (After Brown and Meyerhof 1969)

For the bottom sand layer,

$$q_{ub} = 0.3\gamma'_t DN_\gamma + \sigma'_v N_q \quad (39)$$

where σ'_v is the effective overburden pressure at the interface level. For picking N_{mc} from Fig. 10, $\bar{s}_{ubi}/\bar{s}_{uti}$ should be calculated, and for which an equivalent $\bar{s}_{ubi}$ should be calculated from the expression below:

$$0.3\gamma'_t DN_\gamma = 6\bar{s}_{ubi} \quad (40)$$

In terms of limiting squeezing depth, h_{sq} (the depth range from where the spudcan starts to sense the underlying strong layer and consequently the resistance profile deviates from that without the influence of the underlying layer) solving Eq. 37, ISO suggests

$$\left(5.05 + 1.2 \frac{d}{D} + 0.33 \frac{D}{t_r}\right) s_{uti} + p'_o \geq N_c s_{uti} + p'_o \quad (41a)$$

$$\begin{aligned} & \left(5.05 + 1.2 \frac{d}{D} + 0.33 \frac{D}{t_r}\right) s_{uti} + p'_o \\ & \geq \left(6.05 + 1.2 \frac{d}{D}\right) s_{uti} + p'_o \end{aligned} \quad (41b)$$

$$\frac{t_r}{D} \leq 0.33 \quad (41c)$$

Spudcan Installation in Multilayer Soils

Cavity Depth

Investigation in multilayer soils for developing design chart for limiting cavity depth is sparse. In absence of specific expression, for multilayer clay sediments, the limiting cavity depth can be assessed using

$$\frac{H_{cav}}{D} = \left(\frac{s_{uHi}}{\gamma'D}\right)^{0.55} - \frac{1}{4} \frac{s_{uHi}}{\gamma'D} \quad (42)$$

where s_{uHi} is the intact undrained shear strength at the backflow depth, H_{cav} . Some iteration is

needed to establish H_{cav} and the resulting value of S_{uHi} .

For multilayer soil with a surface sand layer, the volume of the soil above the spudcan, V_{soil} , can be calculated in a manner similar to single-layer sands or sand over clay, as discussed previously.

Design Methods

For assessing spudcan penetration in multilayer soils, three methods will be discussed including (a) the bottom-up ISO approach and (b) the top-down approaches (Zheng et al. 2015, 2017, 2018).

Bottom-up ISO approach: The bottom-up design approach can be used combining the squeezing (for weak-over-strong layering) and punch-through (for the reverse) criteria for two-layer systems discussed previously. Firstly the bearing capacity of a spudcan at the top of the lowest two layers is computed. These two layers are then treated as one (lower) layer in a subsequent two-layer system analysis involving the immediate upper layer. The equivalent undrained shear strength at the surface of a layer is updated after the completion of assessment of penetration resistance profile of that layer (based on the calculated penetration resistance at the surface of that layer).

Top-down approaches (Zheng et al. 2015, 2017, 2018): Recently, top-down approaches have been proposed by Zheng et al. (2015, 2017, 2018) accounting for the effect of the evolution of a soil plug at the base of the spudcan, its accumulation or diminishing, and corresponding effect on the penetration resistance.

Mitigation of Punch-Through and Spudcan-Footprint Interactions

For the jack-up industry, two major concerns related to geotechnical/structural failures during installation and preloading process include (a) spudcan punch-through in layered soils (53%) and (b) spudcan lateral sliding interacting

with an existing footprint from a previous installation (15%) (Jack et al. 2013).

Spudcan Punch-Through

Punch-through incidents occur in stratified soil conditions with a surface or interbedded strong (sand or stiff clay/silt) layer overlying a weak layer. Some examples have been reported by, e.g., Young et al. (1984), Kostelnik et al. (2007), and Menzies and Lopez (2011). The consequent loss may be \$5–50 million per incident.

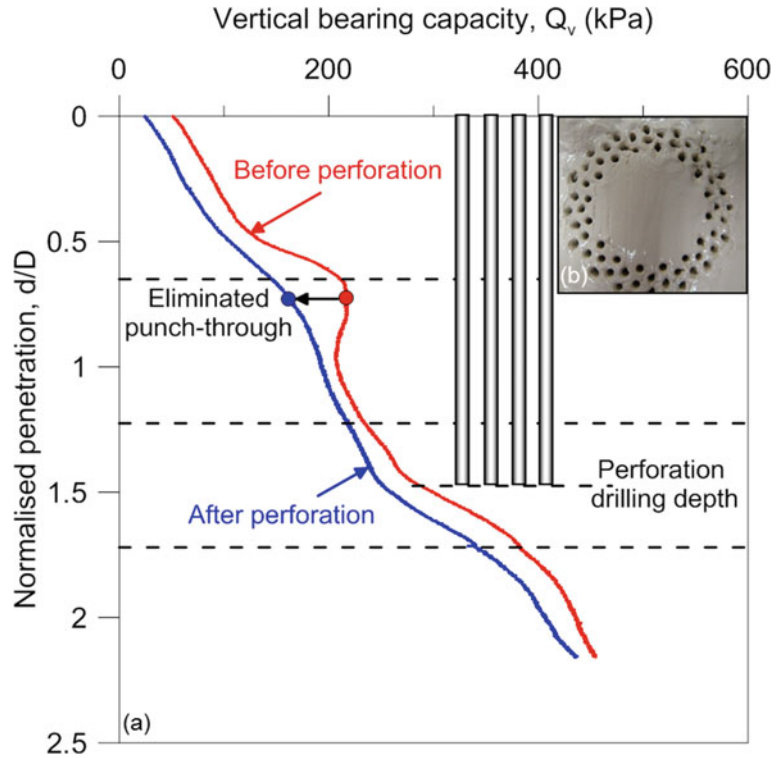
The risk of punch-through can be identified from the assessed spudcan load-penetration profiles using the design methods just discussed. Once the risk of punch-through failure is identified at a site, mitigation methods are required for allowing a safer installation of the rig. Conventionally, installation procedures (such as leg-by-leg preloading; InSafeJIP 2011) have been used for easing punch-through or controlling leg penetration. Recently, several other mitigation methods have been discussed: (a) application of cyclic loading in the strong soil layer to degrade the strength (Erbrich 2005), (b) perforation drilling through the strong soil layer prior to the installation of a spudcan (Maung and Ahmad 2000; Kostelnik et al. 2007; Hossain et al. 2011; InSafeJIP 2011), and (c) a foundation with a peripheral skirt on its bottom side (Teh et al. 2008; Gan et al. 2011; Hossain et al. 2014b; Li et al. 2018).

For stiff-over-soft clay deposits, Hossain et al. (2011) proposed an optimized perforation drilling pattern, as shown in Fig. 11. The experimental centrifuge results confirmed that perforation drilling is indeed effective at mitigating punch-through and severe leg run during jack-up installation in multilayered clays with interbedded stronger layers.

For sand-over-clay deposits, perforation drilling and cyclic degradation are not applicable. Hossain et al. (2014b), Lee et al. (2018), and Li et al. (2018) showed that a skirted spudcan with a skirt length $0.25D$ can effectively be used to eliminate or ease the punch-through risk. An example result is shown in Fig. 12.

Installation of Spudcans,

Fig. 11 Perforation drilling: (a) mitigated punch-through failure; (b) perforation drilling pattern. (After Hossain et al. 2011)



Spudcan-Footprint Interactions

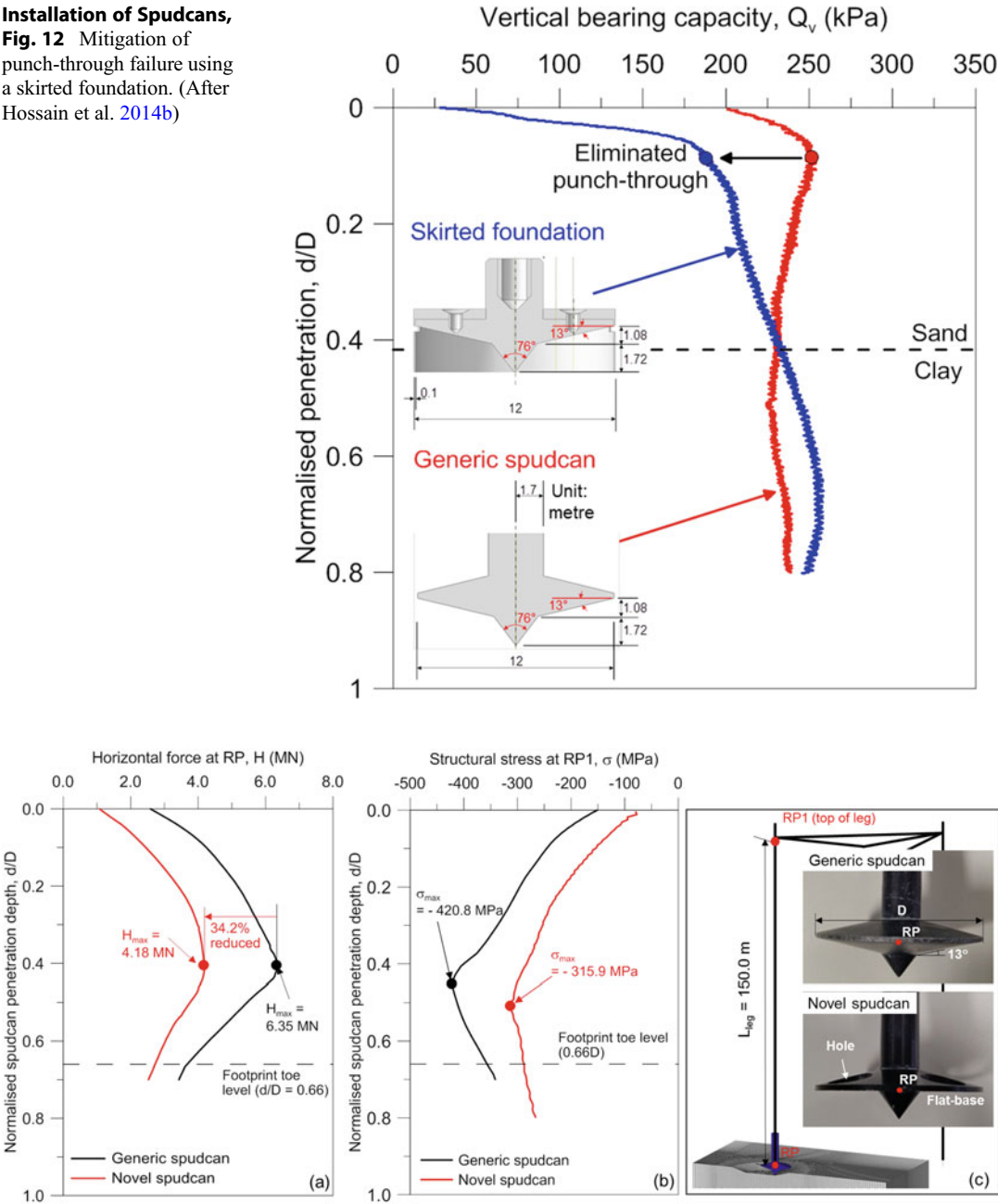
After completing the initial operation, the legs of a jack-up rig are retracted from the seabed, leaving depressions, referred to as a crater or “footprint.” Jack-ups often return to the previous operation site for more drillings, where the footprints from previous explorations are present. Where one of the jack-up legs is located near an existing footprint slope, there is a tendency for the spudcan to slide toward the center of the footprint, inducing excessive lateral displacements and bending moments or rotations of the rig. These detrimental foundation behaviors can result in an inability to install the jack-up at the required position and even worse the occurrence of leg splay and structural damage to the whole jack-up system. The frequency of offshore incidents during installation near footprints has increased by a factor of 4 from the period 1979~1988 to the period of 1996~2005 (Osborne 2005; Jack et al. 2013).

There are several mitigation methods reported, including (a) infilling crater (Jardine et al. 2002),

(b) capping the infilled crater with gravel loading platforms, (c) stomping (Jardine et al. 2002), (d) reaming (Hartono et al. 2014), (e) perforation drilling (Hossain and Stainforth 2016), (f) successive repositioning until the legs had stabilized in the desired plan position (Brenna et al. 2006), (g) use of an identical or very similar spudcan diameter and exactly on the existing footprint (Erbrich 2005), and (h) water jetting along with the spudcan preloading (Handidjaja et al. 2009). In relation to case histories in the North Sea, Jardine et al. (2002) and Grammatikopoulou et al. (2007) examined the potential of using the former two. All the other methods require additional working leading to additional cost and time to be applied in the field.

Consequently, Jun et al. (2018a, b) have focused on tweaking spudcan shapes to ease the spudcan-footprint interactions. These studies have led to establish a novel spudcan shape with a flat base and four holes, which has been shown to be effective at easing spudcan-footprint interactions (see Fig. 13).

Installation of Spudcans, Fig. 12 Mitigation of punch-through failure using a skirted foundation. (After Hossain et al. 2014b)



Installation of Spudcans, Fig. 13 Mitigation of spudcan-footprint interactions using a novel spudcan. (After Jun et al. 2018a, b): (a) horizontal resistance force; (b) structural stress on leg top; (c) reference points and model spudcans

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Insulation Layer

► Cable

Insulation Work

► Thermal Insulation

Intact Stability

► AUV/ROV/HOV Stability

Intact, Damage, and Dynamic Stability of Floating Structures

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Definition

The *stability* is the capability of the *floating structure* to maintain to its upright floating position

both in still water and in waves, whether intact or damaged, and it relates to the interaction among the *center of gravity* (COG), *center of buoyancy* (COB), and *metacenter of intact* or damaged hull under various scenarios.

Scientific Fundamentals

Introduction

The stability is one of the most essential performances for offshore structures. The stability calculation, also referred as stability check, during the design, construction, and operation stage of the offshore structure is fundamental for the safety of the offshore structure and required by the administrations (e.g. Classification Society, Maritime Office). The relation of the COG, COB, and metacenter of offshore structure can be found in Fig. 1, and the nomenclature adopted in this entry is listed in Table 1.

GM and GZ

When the offshore structure inclines from its equilibrium floating position, the center of buoyancy changes from *COB* to *COB'* as shown in Fig. 2. The righting moment is generated by gravity (*G*) and buoyancy (*F*), because the *COG* and *COB'* of the offshore structure are not vertically inline as shown in Fig. 2. The righting moment provides the stability of the offshore structure to maintain its upright floating position.

The intersection point between the straight line through the COG and the original COB and the straight line through the COG and the *COB'*

after inclining is referred as the metacenter. When the angle of inclination is small, the position of metacenter doesn't change. The distance between the COG and metacenter is named as the metacentric height (GM) under small inclination. The distance between the COB and metacenter is named as the metacentric radius (BM) which is calculated as:

$$BM = \frac{I}{\nabla} \quad (1)$$

where *I* stands for the moment of inertia of the waterline area. ∇ represents the displacement of the offshore structure. Then, GM is calculated as:

$$GM = KB + BM - KG \quad (2)$$

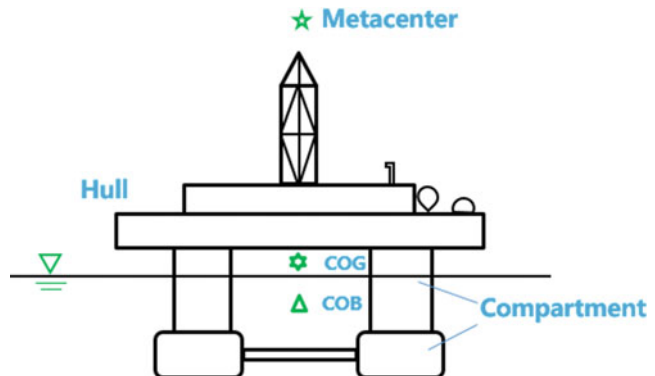
where KB and KG are the vertical heights of COB and COG from the bottom.

The lever of the righting moment, i.e., the horizontal distance between COG and COB, is referred as the righting lever (GZ) as shown in Fig. 1. Thus the restoring moment (*M*) is represented as:

$$M = F \times GZ \quad (3)$$

The change of the righting lever (GZ) against the angle of inclination forms the righting lever curve (i.e., GZ curve) as shown in Fig. 3. The righting lever curve is calculated from the hydrostatic curves. The downflooding angle stands for the angle of inclination where the water begins to flood into hull through openings. The angle of

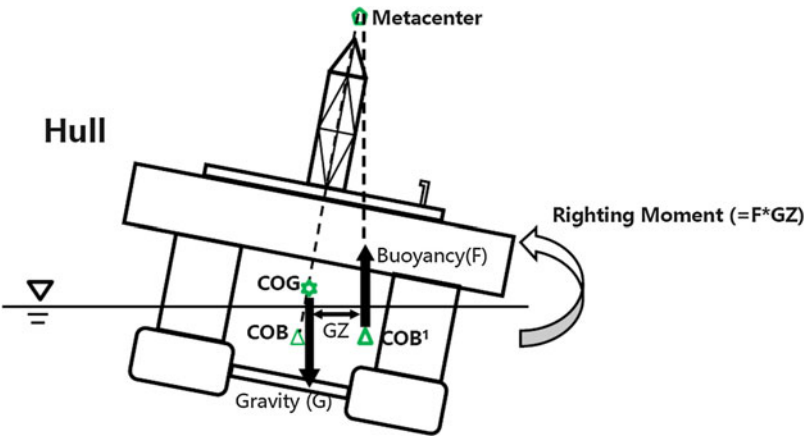
Intact, Damage, and Dynamic Stability of Floating Structures,
Fig. 1 COG, COB, and metacenter of offshore structure



Intact, Damage, and Dynamic Stability of Floating Structures, Table 1 The nomenclature

Symbol	Meaning	Symbol	Meaning
A	Attained subdivision index	KB, KG	Vertical height of COB and COG from the bottom
A _p	Projected area	I	Moment of inertia
BM	Metacentric radius	M	Restoring moment
C _s	Shape coefficient	P	Probability of flooding of the compartments
C _H	Height coefficient	R	Required subdivision index
COG	Center of gravity	S	Survival probability
COB	Center of buoyancy	V	The wind velocity
F	Buoyancy force	θ	Inclination angle
G	Gravity force	ρ	Air density
GM	Metacentric height	φ	Motion (the pitch angle for spar platform)
GZ	Righting lever	ζ	The damping coefficient
▽	Displacement	ω ₀ , ω	Natural frequency, exciting frequency of wave

Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 2 Righting moment of the offshore structure



vanishing stability is the angle where the righting lever, i.e., righting moment, is zero. The vessel capsizes after the angle of vanishing stability and returns back to the equilibrium position before the angle of vanishing stability.

At zero angle of inclination, the gradient of the GZ curve is GM. Therefore GM should be larger than zero to guarantee a positive righting lever under small inclination. The fundamental condition for the offshore structure to maintain its upright floating position is:

$$GM > 0 \tag{4}$$

Requirements for Stability Calculation

The stability calculation checks the ability of the offshore structure to hold its upright position and

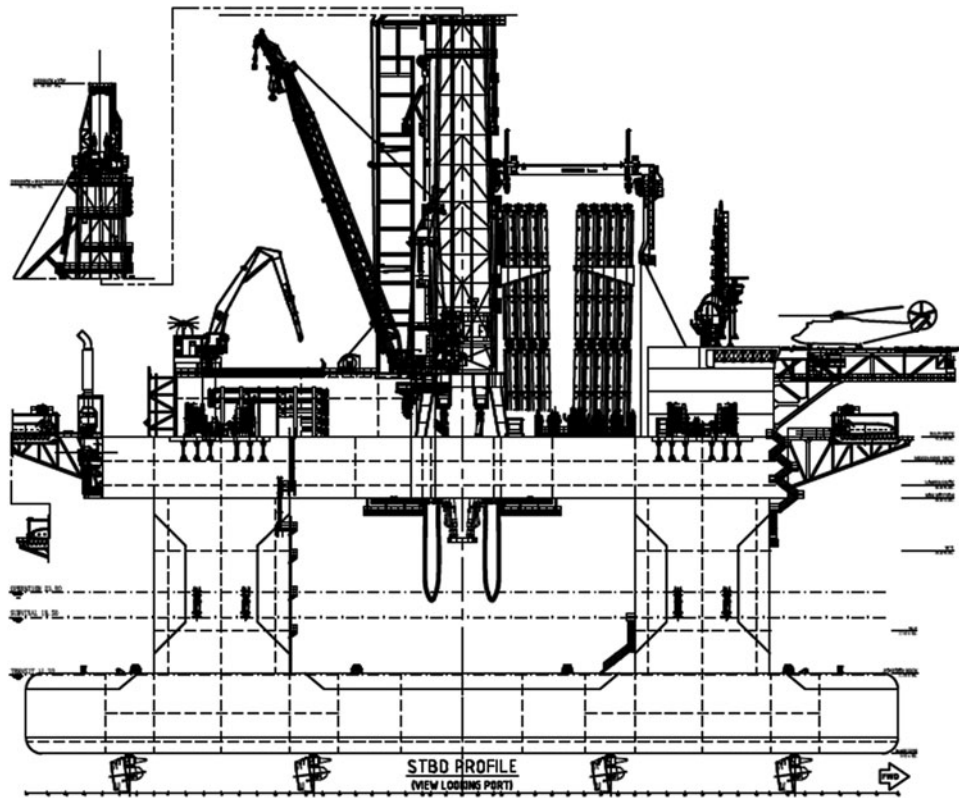
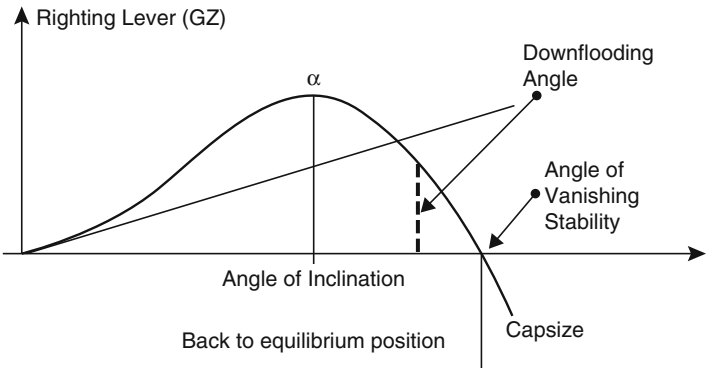
is vital for the safety of the offshore structure. In order to conduct the stability calculation, the following items are required.

General Arrangement, Compartments, and Subdivision

The general arrangement of the offshore structure is defined as the design and separation of spaces both up and under the main deck for all the required functions, machineries, and equipment as shown in Fig. 4.

In the general arrangement, the spaces are divided by transversal, longitudinal bulkheads and decks into several smaller spaces called as compartment. The major compartments include machinery and fuel compartment, cargo compartment, ballast water compartment, accommodation

Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 3 Righting lever (GZ) curve



Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 4 General arrangement of semisubmersible

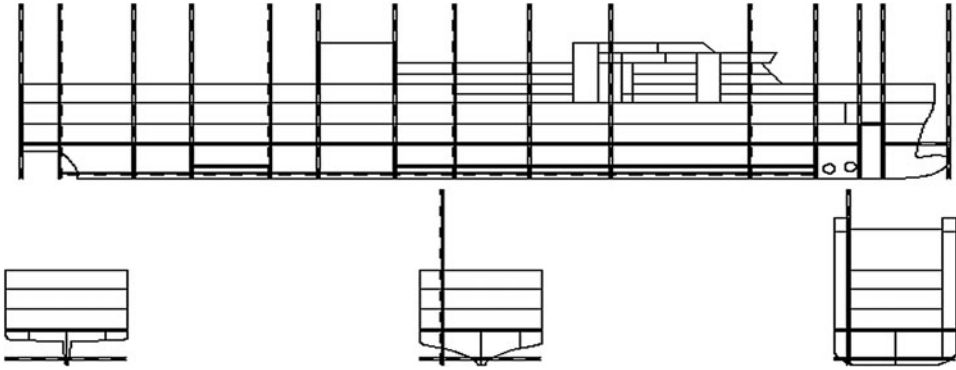
compartment, and production equipment compartment. The detail information of each compartment including volume and position are summarized in the Table of Compartments.

The watertight transversal, longitudinal bulkheads form the watertight subdivision to prevent water from propagating to other compartments

after breaching on one compartment as shown in Fig. 5.

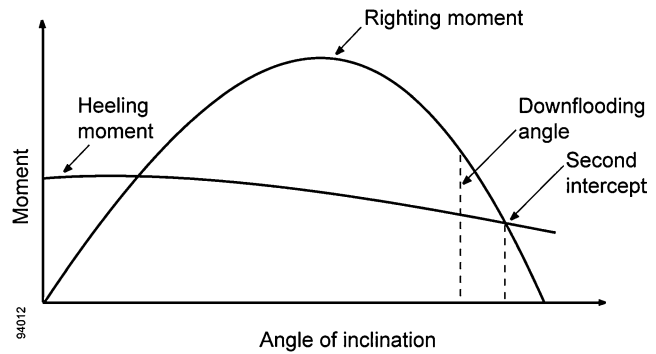
Loading Condition

Loading conditions are the conditions determined by the different loading of cargo, fuel, ballast, and provision in the compartment and on the deck



Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 5 Watertight subdivision

Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 6 Curves of the righting moment and the heeling moment (IMO 2009)



in order to meet various operational demands. Different loading conditions have different GM, GZ curves, and stability performances. Therefore, the stability calculation should be conducted on all the typical loading conditions. The loading condition table contains the load, mass, and filling rate of all the compartments. The typical loading conditions of a mobile drilling unit include towing condition, operation condition (minimum draft and maximum draft), and survival condition under storm.

Stability Curves

Stability curves are the curves required for stability calculation. The most important curves for stability calculation are the curves of the righting moment and the heeling moment as shown in Fig. 6. The right moment curve is generated from the righting lever (GZ) curve by multiplying with buoyancy force (F).

The heeling moment curve is drawn from for wind forces (F_w) calculated by the following formula (IMO 2009):

$$F_w = 0.5C_s C_H \rho V^2 A_p \quad (5)$$

where C_s is the shape coefficient depending on the shape of the structural member exposed to the wind. C_H represents the height coefficient depending on the height above sea level of the structural member exposed to wind. ρ stands for the air density. V is the wind velocity. A_p is the projected area of all exposed surfaces in either the upright or the heeled condition.

Then, the wind heeling moments are calculated by multiplying the wind force with its lever. The lever of the wind force is taken vertically from the center of pressure of all surfaces exposed to the wind to the center of lateral resistance of the underwater body of the unit.

Stability Criteria

Stability criteria are the criteria constituted by the administrations (e.g., Classification Society, Maritime Office). It sets limits for heeling angle, freeboard, GM, and stability curves to ensure the stability and safety of the offshore structure under certain wind and wave loads, whether intact or damaged. The stability criteria are designed based on the notion that the offshore structure has enough right moment to withstand the designed and extreme wind heeling moment, whether intact or damaged.

Major stability criteria for offshore structures include:

- IMO, Code for the Construction and Equipment of Mobile Offshore Drilling Units (IMO 2009)
- DNV, Offshore Standard DNV-OS-C301 Stability and Watertight Integrity (DNV 2014)
- ABS, Rules for Building and Classing Mobile Offshore Drilling Units, Part 3 Hull Construction and Equipment, 2012 (ABS 2012)
- CCS, Rules for Construction and Classification of Mobile Offshore Drilling Units, Part 3 Stability, Subdivision and Load lines, 2005 (CCS 2005).

Stability Calculation

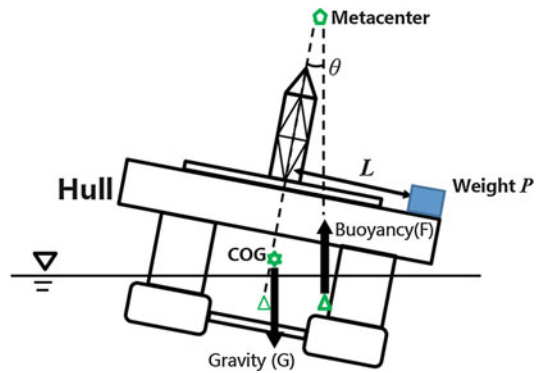
Inclining Test

An inclining test is required for the offshore structure, when it is as near to completion as possible, to measure accurately the light ship data (weight, GM, and position of COG).

In the inclining test, certain weights P (1%–2% of the light ship displacement) are displaced by the distance L from the equilibrium position to generate a moment ($M = PL$) causing an inclination of 2–4° to the offshore structure as presented in Fig. 7. The inclination angle (θ) can be measured by a pendulum or a U-tube. Therefore the metacentric height (GM) is calculated as:

$$GM = \frac{M}{\nabla g \tan \theta} \quad (6)$$

where g is the gravitational acceleration.



Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 7 Demonstration of the inclining test

Intact Stability

Intact stability is the calculation of the stability performance of the offshore structure with an intact hull. Firstly, the calculations require measuring and calculating all the centers of mass of objects on the offshore structure to identify the center of gravity (COG) and the center of buoyancy (COB) of the hull under various loading conditions. Then the intact stability calculations are divided into two categories: initial stability and stability at large angle of inclination.

Initial stability is the stability calculation at small angle of inclination (within 10–15°). It involves the calculation of COG, COB, meta-center, BM, and GM.

Stability at large angle of inclination is the stability calculation. It involves the calculation of the righting lever and moment curves.

After the intact stability calculations, the results such as weight, GM, and righting moment curve are utilized to check whether the stability criteria abovementioned are met. A typical stability criteria can be illustrated as follows (IMO 2009):

- For column-stabilized units, the area under the righting moment curve to the angle of downflooding should be not less than 30% in excess of the area under the wind heeling moment curve to the same limiting angle.
- The righting moment curve should be positive over the entire range of angles from upright to the second intercept.

Damage Stability

Damage stability is the calculation of the stability performance of the offshore structure after damage with breached compartments. It requires more complicated and heavy calculation than intact stability. Thus, special numerical software such as NAPA are typically employed to handle the heavy calculation works.

In the damage condition, one or more compartments are breached and flooded by water. Its contents are assumed to be lost and replaced by water. If contents are lighter than water (e.g., light oil), then buoyancy is lost, and the offshore structure inclines unless the compartment is on the center-line. The new floating position after flooding is achieved. Under the new floating position, the freeboard, GM, and righting moment curves are calculated (Fig. 8).

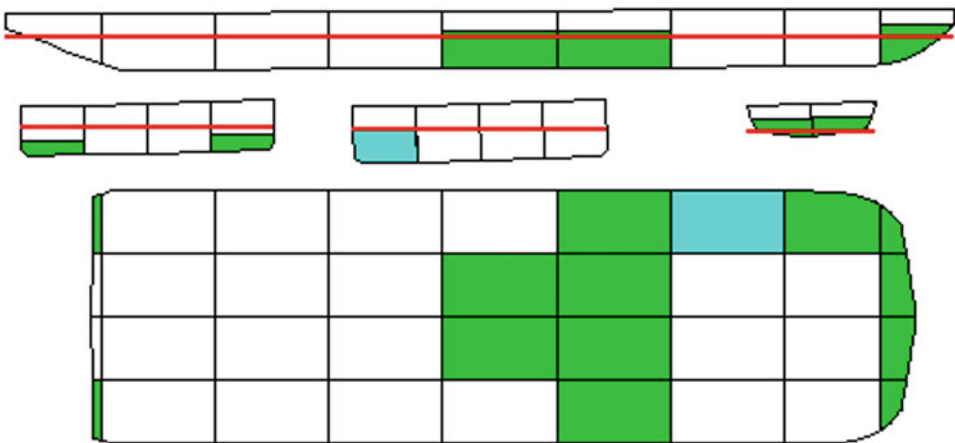
In the damage stability calculations, many damage scenarios with different compartments breached should be selected. There are two methods for damage stability calculations: deterministic damage stability and probabilistic damage stability

Deterministic damage stability is a traditional and widely used method for damage stability calculations. A deterministic extent of damage on hull is decided firstly. Then it assumed that all the compartments within this extent of damage

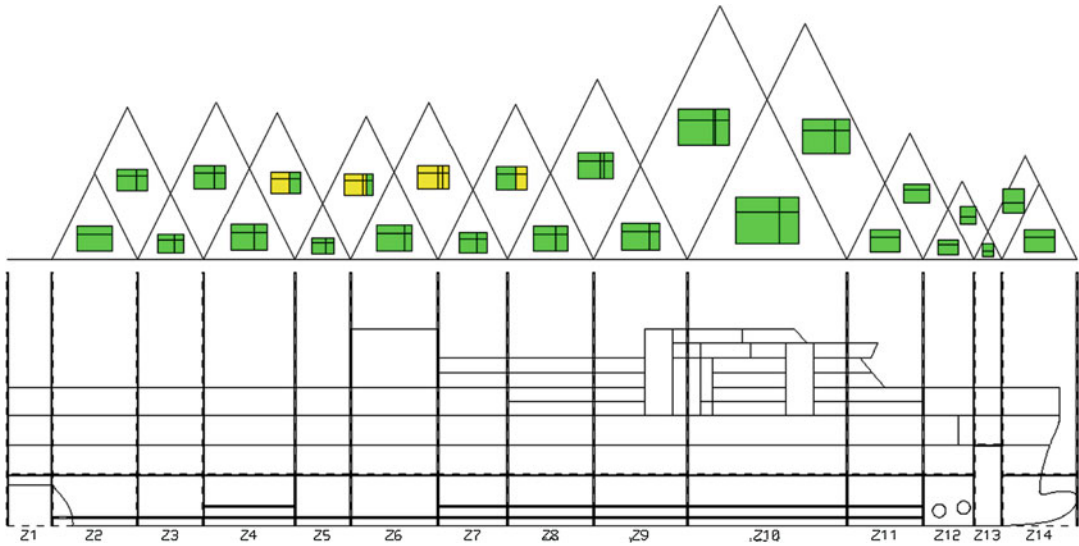
are breach deterministically without considering the breaching probability of these compartments. The results of deterministic stability calculations are the floating position, the freeboard, GM, and righting moment curves after breaching of one or more compartments. Then the results are utilized to check whether the deterministic stability criteria are met.

Probabilistic damage stability is a newly developed method especially for passenger vessels. The damage stability calculations are of a probabilistic nature, which are based on the breaching probability on one or more compartments and survival probability after breaching (Fig. 9).

The probability of flooding of the compartments in the hull position is calculated as P based on injury statistics (probability of opening). The survival probability S (which is the probability of survival without sinking and capsizing even in the waves) is calculated when the partition is damaged or flooded. If the survival probability S is greater than 0, it means that this ship does not capsize even if some compartment is flooded. If S is 0, it means sinking. The total survival probability A is calculated for the entire ship by integrating the product of P and S over the entire length of the ship. It is required that this value should be larger than the criteria's required value R . This survival probability A is called



Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 8 Floating position of a barge after breaching on a ballast tank



Intact, Damage, and Dynamic Stability of Floating Structures, Fig. 9 Diagram for probabilistic damage stability

“attained subdivision index A,” and the regulation’s required value R is called “required subdivision index R.”

After the damage stability calculations, the results such as weight, GM, and righting moment curve are utilized to check whether the stability criteria abovementioned are met. A typical deterministic damage stability criteria can be illustrated as follows (IMO 2009):

- Columns and braces should be assumed to be flooded by damage having a vertical extent of 3 m occurring at any level between 5 m above and 3 m below the draughts specified in the operating manual.
- The angle of inclination after the damage within the extent should not be greater than 17° .
- The righting moment curve after the damage should have, from the first intercept to the lesser of the extent of weather tight integrity and the second intercept, a range of at least 7° . Within this range, the righting moment curve should reach a value of at least twice the wind heeling moment curve, both being measured at the same angle.

Dynamic Stability in Waves

The abovementioned stability calculations are all of the static nature, which consider the motions of vessels, wind, and wave loads statically. However, dynamic motions and loads can also cause stability failure event such as parametric resonance and excessive acceleration.

Parametric Resonance

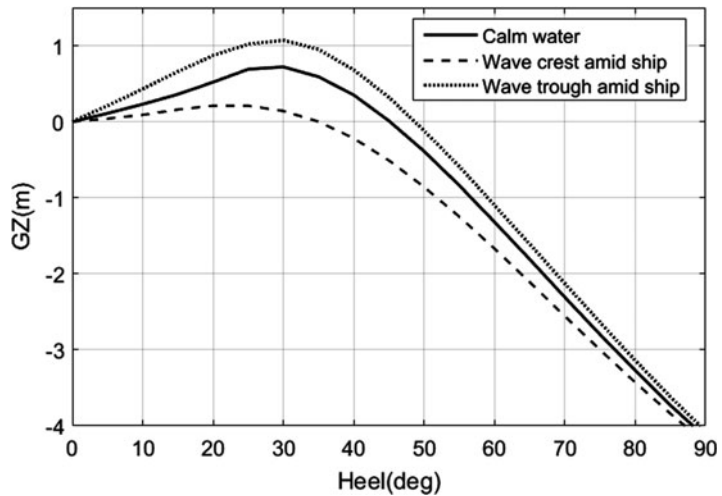
Parametric resonance, as a self-resonance phenomenon, occurs with excessive inclining motion and even leads to capsizing. It can bring severe cargo loss and damage (France et al. 2003). It is caused by the time varying GM and weakening of righting moment under certain large waves. The GM and righting moment variations can be caused by either the change of the waterline area in waves (such as container ships as shown in Fig. 10) or the change of the COB in waves (such as Spar and semisubmersible platforms).

The fundamental dynamics of parametric resonance caused by the time varying of restoring characteristics can be simplified as the damping Mathieu equation:

$$\phi'' + 2\zeta\phi' + (\omega_0^2 + \epsilon \cos \omega t)\phi = 0 \quad (7)$$

Intact, Damage, and Dynamic Stability of Floating Structures,

Fig. 10 Variation on GZ curves in waves of the KCS containership (Yu et al. 2017)



where ϕ is the motion (the pitch angle for spar platform), ζ is the damping coefficient, and ω_0 and ω stand for the motion natural frequency and the exciting frequency of wave, respectively. Through the analysis on the equation, it is found that the unstable and large motion response exists when the ω/ω_0 equals to 2. Therefore, the parametric resonance occurs when the wave exciting frequency is around twice of the motion natural frequency.

Conclusion

Stability is the capability of the offshore structure to maintain its upright floating position both in still water and in waves, whether intact or damaged. It is one of the most essential performances for offshore structures and of prime importance for safety.

Stability involves a series of calculation on the COG, COB, GM, hydrostatic curves, and stability curves based on the weight distribution, general arrangement, compartment, loading conditions, etc. Traditional stability calculations include inclining tests, intact stability, damage stability, as well as check based on stability criteria. In recent years, the dynamic stability in waves such as parametric resonance and excessive acceleration are also believed to be important parts of stability.

Cross-References

- [Fishing Ships](#)
- [High-performance Ship](#)
- [Jacket Platform](#)
- [Luxury Cruises](#)
- [Research Ship](#)
- [Semi-submersible Platform](#)
- [Service Ships](#)
- [Special Marine Vehicle](#)
- [Surface Warships](#)
- [Transport Ship](#)

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Integrated Navigation

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Synonyms

Acoustic positioning system (APS); Auxiliary navigation; Doppler velocity log (DVL); Global navigation satellite system (GNSS); Long-baseline (LBL); Magnetic compass (MCP); Multi-source information fusion navigation; Numerically controlled oscillator (NCO); Short baseline (SBL); Strapdown inertial navigation system (SINS); Time of flight (TOF); Ultra-short baseline (USBL)

Definition

Integrated navigation refers to the use of two or more different navigation systems to measure the same information source and extract the errors of each system from the comparative values of these measurements to correct each system. The navigation system using integrated navigation technology is called an integrated navigation system, and the navigation systems involved in the integration are called subsystems. As the inertial navigation system has the advantages of autonomy, concealment, information comprehensiveness, and broadband, it is generally used as the key subsystem of integrated navigation system. Besides the inertial navigation system, the integrated navigation system may be composed of different radio navigation systems, satellite navigation systems, acoustic positioning systems, geophysical navigation, and so on. There are many ways and depths of combination. Integrated navigation can take the advantages of each system, make up for its shortcomings, and improve positioning accuracy and data redundancy. So the integrated navigation system is more reliable and complex than a single navigation technique.

Scientific Fundamentals

Main Underwater Navigation Technology

Inertial Navigation

Inertial navigation is fully autonomous navigation that does not receive external signals or transmit signals to the outside. With three orthogonal gyros and three orthogonal accelerometers, the inertial navigation system can continuously measure the three orthogonal linear accelerations and the three orthogonal angular rates in an inertial reference frame. By navigation calculation, the information of position, altitude, and velocity is acquired. Two types of inertial navigation systems are platform and strapdown. Due to the influence of cost, energy, volume, and other factors, the strapdown inertial navigation system (SINS) is usually used for underwater vehicles. With the rapid development of the fiber-optic gyro, SINS will be more accurate, smaller in size, more reliable, cheaper, and lower in energy consumption. However, the system errors of SINS will accumulate over time, which is difficult to meet the long-distance and long-time navigation requirements. Therefore, the external navigation information from other navigation systems is necessary for SINS to correct diffuse navigation errors.

Global Navigation Satellite System (GNSS)

GNSS is a global, all-weather, high-precision, continuous and real-time navigation and position system based on radio. GNSS has the advantage of not accumulating errors with time and the obvious disadvantage of no signal underwater. In the long-distance navigation, the underwater vehicles can float onto the surface of water regularly to correct the accumulating velocity or position errors of the other navigation systems.

Acoustic Positioning System (APS)

Compared with the electromagnetic signal, the acoustic signal travels farther in water. Therefore, acoustic transmitter can be used as the beacon of underwater vehicles. At present, long-baseline (LBL), short baseline (SBL), and ultra-short baseline (USBL) are the three main APS methods. All three navigation methods need to know the ocean

condition in advance. Transducers or transducer array devices are deployed in the sea area beforehand. One or more acoustic sensors pre-installed on the underwater vehicles receive a pulse signal emitted by the transducer acoustic source. The relative position of the transducer acoustic source and underwater vehicles can be obtained by calculating the received pulse signal according to the pre-set mathematical model.

APS has no position error accumulation in underwater vehicle navigation. Accuracy of APS is related to the baseline length and the distance between the APS and the underwater vehicles. LBL positioning principle is triangulation with at least three acoustic transponders, which are deployed in the mission area. For SBL, transponders are installed at both ends of the hull of the mother ship, and triangulation is used, and the baseline is determined by the size of the mother ship. The USBL method can determine the relative position of underwater vehicles to the mother ship, and the relative distance and orientation are solved by the time of flight (TOF) and the phase difference of an acoustic pulse arriving at different receivers on the bottom of the mother ship.

Geophysical Navigation

Geophysical navigation needs to be able to establish relatively accurate underwater environmental survey maps of geophysical parameters (such as seabed depth, gravity, magnetic field, etc.) in advance. Because the seabed structure, topography, and depth are different in different sea areas, underwater vehicles can estimate their positions by measuring the geophysical parameters and comparing with the previous underwater environmental survey maps of geophysical parameters. The precondition of geophysical navigation is that the measured parameters vary sufficiently in spatial distribution.

Auxiliary Navigation Device

Doppler Velocity Log (DVL)

DVL is to calculate the relative velocity of underwater vehicles relative to the seabed or water layer by transmitting sound waves continuously to the bottom or water layer under the carrier and

according to the relationship between the returned beam and the emitted beam received by the receiver. DVL can measure the left, right, front, and back traveling velocities of underwater vehicles. DVL can work all-weather without being restricted by various environments.

Magnetic Compass (MCP)

MCP measures the direction of the geomagnetic field through the magnetic sensor and then gives the angle between the longitudinal axis of underwater vehicles and the magnetic meridian in the horizontal plane, that is, the so-called magnetic heading angle. MCP is more susceptible to external interference, so its accuracy is low. However, because of its low cost, simple structure, and high reliability, it can be used as auxiliary navigation equipment for underwater vehicles.

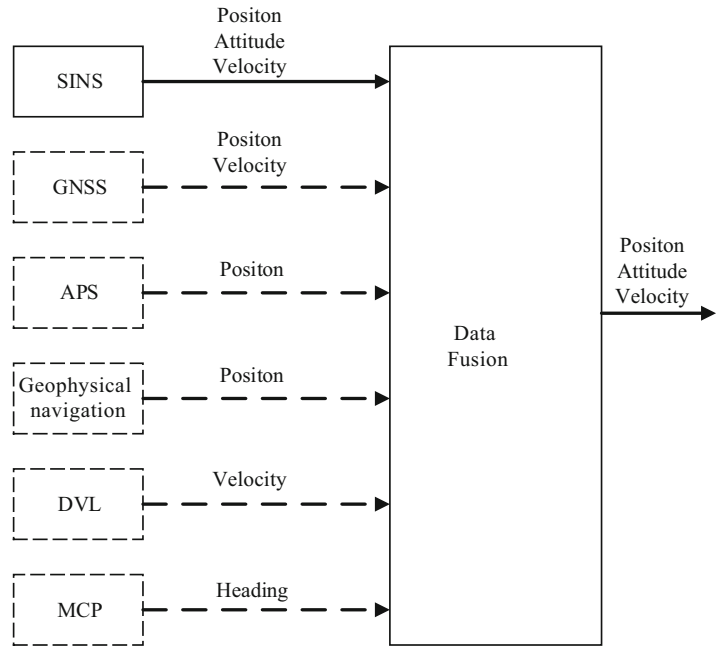
Work Principle

At present, the commonly used underwater navigation positioning methods include inertial navigation, terrain matching, dead reckoning, geophysical navigation, and acoustic navigation. Various single navigation methods have more or less deficiencies in accuracy, reliability, or other aspects, which cannot meet the needs of the development of underwater vehicles. In addition to improving the intrinsic performances of each single navigation device, the low-cost, high-performance integrated navigation has become another development direction of underwater navigation technology.

Most of the underwater integrated navigation system is mainly based on the inertial navigation system, supplemented by acoustic navigation, gravity matching, terrain, and geomagnetic matching system. The common working principle of an integrated navigation system is shown in Fig. 1. Each navigation system provides corresponding navigation information, which is fused by the filtering method to obtain more accurate and reliable navigation information. According to different application environments and requirements, one or more other navigation modules are chosen for integrated navigation. The general process of building integrated navigation system can be described as follows: by modeling and analyzing

Integrated Navigation,

Fig. 1 The principle of the integrated navigation system



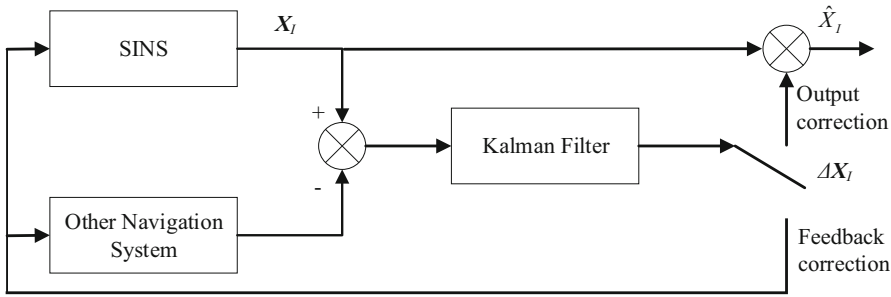
the system structure abstractly, the state equation and measurement equation of the system is established, and the state information or state error information of the system is estimated optimally by Kalman filter, and the optimal estimation is used to correct the navigation parameter information of subsystems.

Data Fusion Technology

The essence of the integrated navigation is a multi-sensor navigation information optimization processing technology, so information fusion technology is the key technology of integrated navigation. Kalman filter method is a recursive least mean square error estimation method in the time domain, which has the characteristics of engineering application and easy realization. Kalman filter is the basic technology of information fusion estimation in an integrated navigation system. Kalman filtering method includes the direct method that directly takes the navigation parameters as the estimation objects and the indirect method that takes navigation system error as the estimation objects. The direct method reflects the dynamic process of navigation parameters, while the indirect method uses error equation, which has some approximation (Yan et al. 2013).

However, the equation obtained by the direct method is non-linear, and the order of magnitude of each navigation parameter varies greatly, which easily affects the estimation accuracy. So the indirect method is generally used in engineering.

The indirect method includes output correction and feedback correction, as shown in Fig. 2. (1) Output correction uses filter estimation as the output of the integrated system or as the output of INS to improve the accuracy of output, but it does not modify the internal error state of INS. The accuracy of filter estimation determines the accuracy of correction. Its advantage is that subsystems and filters work independently and are easy to be realized in engineering. The fault of filters will not affect the operation of the system, so it has high reliability. However, system model changes will occur during long-time operation, which will lead to the reduction of filtering accuracy. (2) Feedback correction is to feedback the filter estimation to INS and other navigation systems, whose error state will be corrected. The corrected parameters of INS and other navigation systems enter the next operation. In the feedback correction mode, the corrected attitude matrix is fed back into the system to participate in the solution, so the system error is always small, and there will



Integrated Navigation, Fig. 2 Output correction and feedback correction

be no model error, so it has higher accuracy. Engineering implementation is more complex, and filter failure will directly affect the system output, thereby reducing reliability.

There are three common data fusion structures of centralized structure, distributed structure, and hybrid structure. (1) For the centralized fusion structure, there is a total data processing center to collect the data of the whole system. The data are processed and fused by the processing center according to the corresponding algorithm. Because the original data from all sensors are processed by the fusion center, there is no loss of information, and the performance is better, and the fusion result is optimal theoretically. (2) For the distributed fusion structure, each sensor has its independent data processor. First, they perform some operations and preprocess the data by themselves. Then they directly transmit the current state estimations to the main processor of the fusion center. After fusing all local state estimations, the fusion center outputs the global estimations. Therefore, the distributed fusion is also called sensor-level fusion or autonomous fusion. (3) Hybrid fusion structure is a synthesis structure of centralized structure and distributed structure. The fusion center can obtain either original information or local state estimations.

Typical Applications

SINS/GNSS Integrated Navigation

At present, the common modes of SINS/GNSS integrated navigation are loose integration, tight

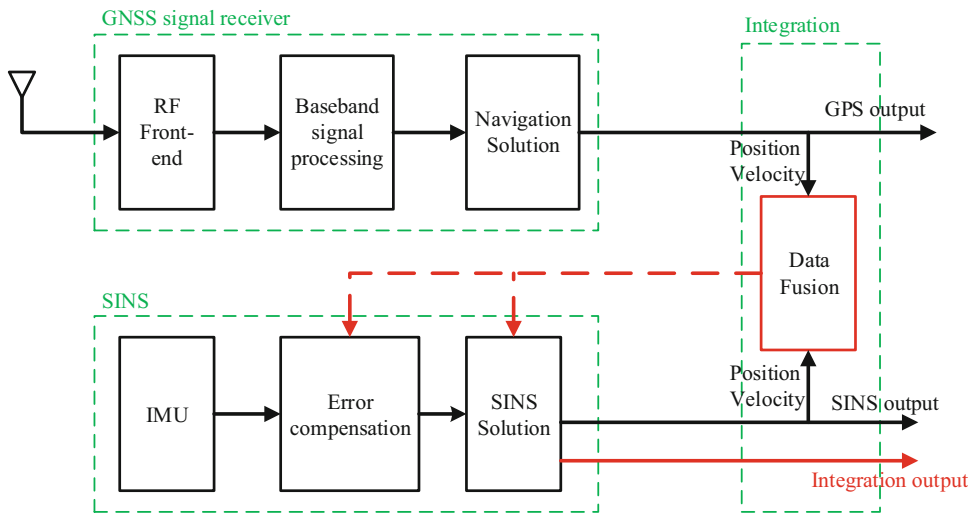
integration, and ultra-tight integration. For these three combinations, there are differences in system structure, the physical quantity used in modeling and filtering algorithm.

Loose Integration

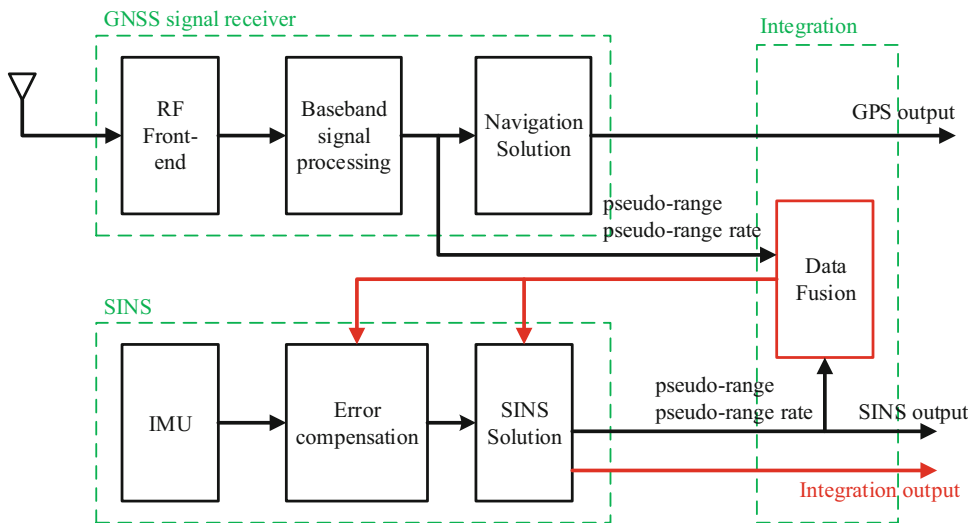
The GNSS and SINS of the loose integration system are independent of each other in their work, and the navigation data can be solved without changing the internal structure of each subsystem. The differences of position and velocity information obtained by GNSS and SINS are transmitted to Kalman filter as the observation of the integrated navigation system. The optimal estimation of error is used to correct the SINS. If any single system is not available for a short time, the loose integration system can still provide navigation output. The loose integration system has better robustness and accuracy than those of pure SINS. However, because the noise in the system is correlative, the system error may be largen. If there are less than four stars available, the long-time unlocking of the GNSS signal will cause error divergence of the whole system. The structure of the loose integration system is shown in Fig. 3.

Tight Integration

Tight integration does not directly use the position and velocity information given by all sensors but uses the intermediate quantities of the receiver: pseudo-range and pseudo-range rate. GNSS signal receiver is needed to give the original observation information. Tight integration not only has



Integrated Navigation, Fig. 3 Structure of loose integration



Integrated Navigation, Fig. 4 Structure of tight integration

all the advantages of loose integration but also can effectively improve the dynamic performance, anti-interference performance, and filtering accuracy of the system. The system structure diagram is shown in Fig. 4. Both theory and practice prove that tightly integration navigation has higher precision. Because tightly integrated navigation uses the original information of the receiver, it can still

provide high navigation accuracy even if there are less than four stars available. Because the observation information is irrelevant, the system error convergence speed is accelerated, and the GNSS cycle slip can be effectively suppressed by INS output. However, the structure of a tight integration system is more complex, which needs to change the internal structure of the receiver to

achieve, and the computation is also large, and the real-time performance is not good.

Ultra-tight Integration

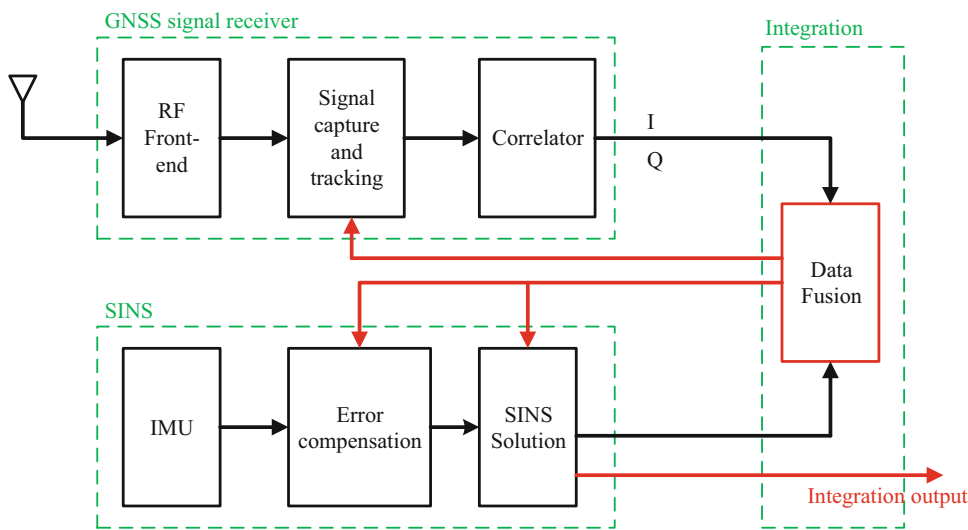
The ultra-tight integration navigation system is the most complex and the most precise integration mode. It combines INS with co-directional signal (I) and quadrature phase signal (Q) from the receiver correlator. The function of the GNSS receiver whose internal hardware structure needs to be changed is realized by software. The numerically controlled oscillator (NCO) of code ring and carrier ring are controlled by the output of navigation filter, which improves the ability of the receiver to keep signal tracking under high dynamic and interference conditions. When the receiver is out of the lock, the fast re-acquisition of the signal can also be realized by using accurate navigation parameters of the ultra-tight integration. The structure diagram is shown in Fig. 5. Because the ultra-tight integration system integrates signal tracking and data fusion, it has a complex structure, large computation, and is not easy to realize in engineering. Li et al. (2018) suggest an enhanced tightly coupled navigation approach using pseudo-range, Doppler, and carrier phase measurements, which can increase computation efficiency without iteration calculations.

INS/DVL Integrated Navigation

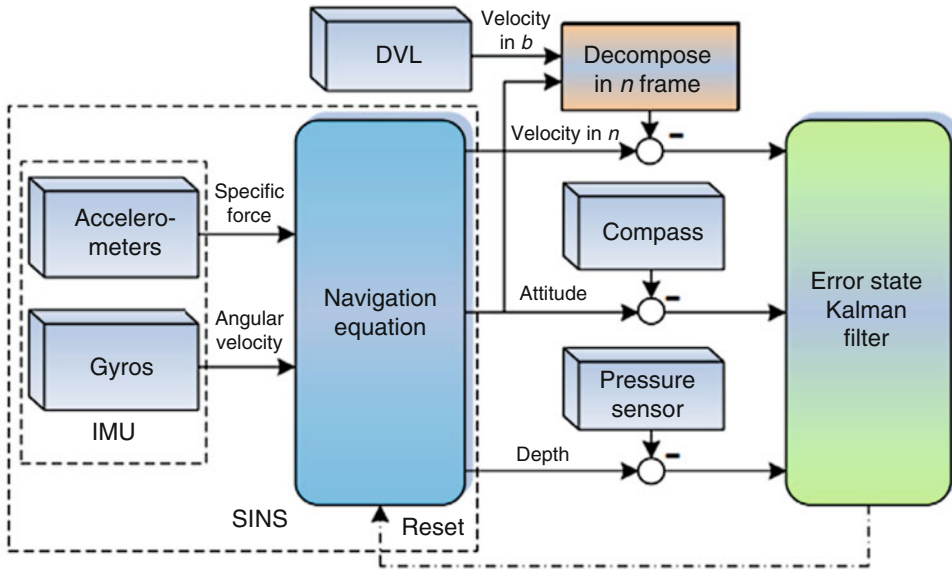
The INS/DVL integrated navigation mainly uses the velocity of DVL to restrain the INS accumulation error. In many applications, the DVL-aided inertial navigation system uses a depth sensor and compass as auxiliary sensors. Figure 6 shows a typical SINS/DVL integrated navigation structure. In Fig. 6, the error state Kalman filter is used to estimate the SINS navigation errors. The navigation solution and the navigation error estimates are independent of each other. The navigation error estimates are transmitted to the SINS to reset the navigation parameters. Subsequently, the corrected navigation parameters are used as recent initial conditions of the navigation equations. DVL measures the velocity of the carrier relative to the sea bottom.

Integrated Navigation Combining INS with USBL

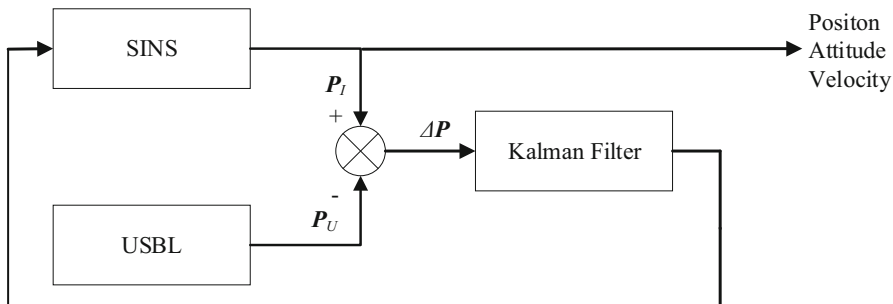
In the SINS/USBL integrated navigation system, SINS continuously outputs the real-time information of position, attitude, and velocity acquired by navigation calculation from the data measured by three orthogonal gyros and three orthogonal accelerometers. Using the method of phase difference or phase comparison to measure the phase differences of the acoustic unit, the position of the transponder/responder in the coordinate system of the acoustic array is obtained. The relative



Integrated Navigation, Fig. 5 Structure of ultra-tight integration



Integrated Navigation, Fig. 6 SINS/DVL integrated navigation structure (Bao et al. 2019)



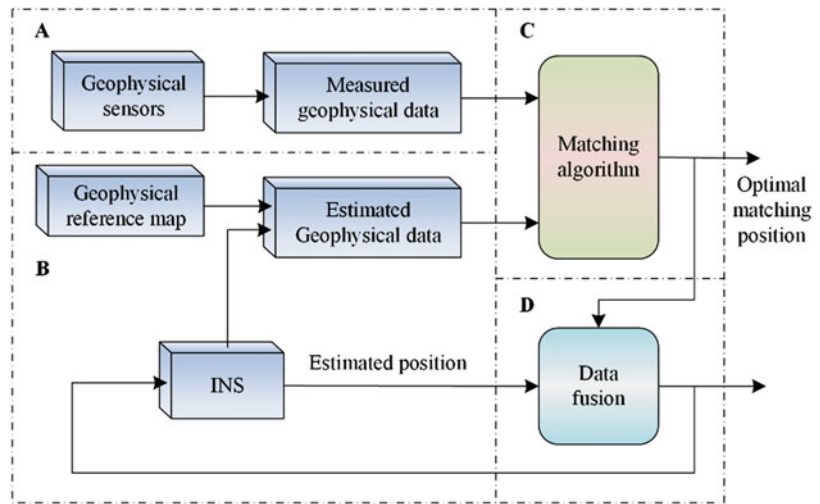
Integrated Navigation, Fig. 7 Schematic block diagram of SINS/USBL integrated navigation system

distance between the acoustic array and the underwater vehicles is calculated by the time of sound wave propagation in the water. USBL outputs the position information in the geographic coordinate system of underwater vehicles. Using the indirect filtering method, the differences between the position information of SINS and the position information of USBL are taken as the measurement quantities and output into the filter to obtain the optimal estimations of the states of the integrated navigation system. The navigation error of SINS is also corrected with the optimal estimations. The schematic block diagram of the SINS/USBL integrated navigation system is shown in Fig. 7.

Integrated Navigation Combining INS with Geophysical Navigation

An integrated navigation system composed of INS and geophysical navigation, which uses inertial navigation as its core, can achieve high-precision navigation by correcting the cumulative error of the INS in real time using the geophysical field within traveling range of the underwater vehicles. The principle diagram of the geophysical navigation assisted INS is shown in Fig. 8. In Fig. 8, the measured geophysical data are obtained using geophysical sensors mounted on the vehicles (see Fig. 8a). Meanwhile, according to the estimated position provided by the INS, the

Integrated Navigation,
Fig. 8 Principle diagram
of geophysical navigation
assisted inertial navigation
(Bao et al. 2019)



estimated geophysical data are achieved from an existing geophysical reference map (see Fig. 8b). Then the measured geophysical data and the estimated geophysical data are transmitted to the navigation computer. The optimal matching position of the vehicle is determined by the matching algorithm (see Fig. 8c). Through fusing the INS position and the optimal matching position, cumulative errors of the INS can be effectively corrected and the navigation performance of the INS is improved (see Fig. 8d). Geophysical navigation mainly consists of terrain matching navigation, geomagnetic matching navigation, and gravity matching navigation.

Li Z, Wang D, Dong Y, Wu J (2018) An enhanced tightly-coupled integrated navigation approach using phase-derived position increment (PDPI) measurement. *Optik* 156:135–147

Yan X, Ouyang Y, Sun F, Fan H (2013) Kalman filter applied in underwater integrated navigation system. *Geodesy Geodynamics* 4(1):46–50

Intelligent Algorithm Based

► Underwater Acoustic Sensor Network

Cross-References

- Auxiliary Navigation
- Doppler Velocity Log for Navigation System in Underwater Vehicle
- Long Baseline Underwater Acoustic Location Technology
- Multi-source Information Fusion Navigation

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Bao J, Li D, Qiao X, Rauschenbach T (2019) Integrated navigation for autonomous underwater vehicles in aquaculture: a review. *Inf Process Agric*. <https://doi.org/10.1016/j.inpa.2019.04.003>

Intelligent Control Algorithms in Underwater Vehicles

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Synonyms

Advanced control; Automatic control; Fuzzy control; Neural network control; Optimal control; Robust control

Definition

Intelligent Control Algorithm is a kind of control method, which could efficiently perform scheduled tasks with its own decision-making and optimization capabilities, derived from classical control theory and optimization algorithm. Underwater vehicles have become an important tool for various underwater tasks because they have endurance, stationarity, speed, and depth capability, as well as a higher factor of safety, than human divers. However, the marine environment is unstructured and hazardous, and the development of underwater vehicles will face many challenging scientific and engineering problems. In the last three decades, researchers have made great efforts to overcome these problems by using intelligent control algorithm in underwater vehicles.

Scientific Fundamentals

Underwater vehicles have been widely used in many subsea tasks, and seven critical areas in ocean system engineering are identified as follows: system for characterization of the sea bottom resources; systems for characterization of the water column resources; waste management systems; transport, power, and communication systems; reliability of ocean systems; materials in the ocean environment; and analysis and application of ocean data to develop ocean resources (Asokan and Singaperumal 2008). Very often, according to the task, the underwater vehicle is required to continuously change its operating tool and/or to pick up and release loads causing a change in behavior. That results as an inherent change in its weight, buoyancy, and hydrodynamic forces and as a consequence, a decrease in the position tracking performance. In addition, underwater vehicles have to deal with the highly dynamical underwater environment represented in the form of ocean currents and waves in shallow water (Xu et al. 2006; Xu and Xiao 2007). With this in mind, when the dynamic characteristics of the system are time dependent or the

operating conditions of the system vary, it is necessary to retune the gains or parameters to obtain the desired performance, resulting in time consumption.

Mathematical Model for Underwater Vehicle

The nonlinear model of a 6 DOF to build the mathematical model that represents the underwater vehicle dynamics two reference frames were used; one referenced to earth (called the Earth-fixed frame) and another referenced to the vehicle (called the body-fixed frame), Fig. 1 (Hernández-Alvarado et al. 2016; Fossen 2002).

Kinematic Model

The general velocity vector is represented as:

$$\mathbf{v} = [\mathbf{v}_1 \ \mathbf{v}_2]^T = [u \ v \ w \ p \ q \ r]^T \quad (1)$$

where u , v , and w are components of the linear velocity in surge, sway and heave directions, respectively, and p , q , and r are components of the angular velocity in roll, pitch, and yaw, respectively.

The position vector $\eta_1 \in \mathbb{R}^3$ and orientation vector $\eta_2 \in \mathbb{R}^3$ coordinates expressed in the Earth-fixed frame are:

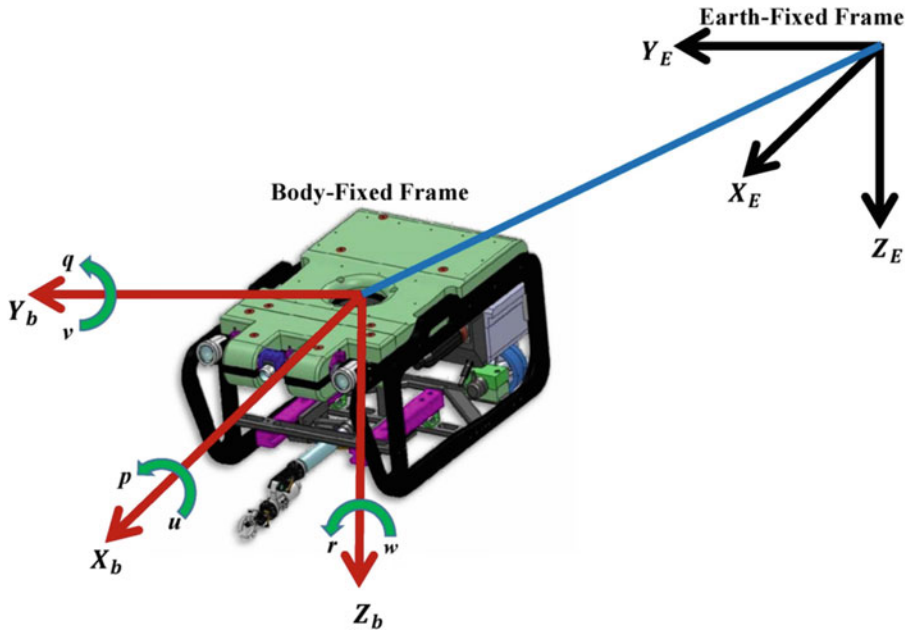
$$\eta = [\eta_1 \ \eta_2]^T = [x \ y \ z \ \phi \ \theta \ \psi]^T \quad (2)$$

where x , y , and z represent the Cartesian position in the Earth-fixed frame and ϕ represents the roll angle, θ the pitch angle, and ψ the yaw angle.

The relationship between velocities in the Earth-fixed and body-fixed frames is

$$\begin{bmatrix} \dot{\eta}_1 \\ \dot{\eta}_2 \end{bmatrix} = \begin{bmatrix} J_1(\eta_2) & O_{3 \times 3} \\ O_{3 \times 3} & J_2(\eta_2) \end{bmatrix} \begin{bmatrix} \mathbf{v}_1 \\ \mathbf{v}_2 \end{bmatrix} \quad (3)$$

where $J_1(\eta_2) \in \mathbb{R}^{3 \times 3}$ is the rotation matrix which expresses the transformation from body-fixed to Earth-fixed frame, and $J_2(\eta_2) \in \mathbb{R}^{3 \times 3}$ is another transformation matrix that relates the angular velocity $\mathbf{v}_2 \in \mathbb{R}^3$ with the time derivative of $\eta_2 \in \mathbb{R}^3$.



Intelligent Control Algorithms in Underwater Vehicles, Fig. 1 Frame coordinates of an underwater vehicle (Hernández-Alvarado et al. 2016)

Hydrodynamic Model

Equations of motion are expressed by the following equation,

$$\begin{aligned} M\dot{v} + C(v)v + D(v)v + G(\eta) &= \tau \\ \dot{\eta} &= J(\eta)v \end{aligned} \quad (4)$$

where $v \in \mathbb{R}^n$ and $\eta \in \mathbb{R}^n$ were previously defined, $M \in \mathbb{R}^n \times \mathbb{R}^n$ denotes the inertial matrix (including the added mass), $C \in \mathbb{R}^n \times \mathbb{R}^n$ is the Coriolis matrix and centripetal forces (including the effects of added mass), $D \in \mathbb{R}^n \times \mathbb{R}^n$ refers to the damping matrix, $G \in \mathbb{R}^n \times \mathbb{R}^n$ represents the vector of gravitational forces, and $\tau \in \mathbb{R}^n$ is the input control vector.

Ocean Currents

Ocean current is generated by wind, tide, variation of densities, and re-circulation of water, among others. The detailed information about ocean currents can be obtained from the model of induced ocean currents proposed by Fossen (2002). In the mentioned work, the equations of motion are

represented in terms of relative velocity of the vehicle and the currents,

$$v_r = v - v_{CI} \quad (5)$$

where $v_{CI} = [u_C \ v_C \ w_C \ 0 \ 0 \ 0]^T$ is a non-rotational vector of the current velocity according to Eq. (3). Note that the linear velocity on the fixed frame can be transformed to linear velocity in the equation by applying the elemental rotation matrices.

Suppose the current velocity in the equation as constant or at least with a minimum variation, so that:

$$\dot{v}_{CI} = 0 \rightarrow \dot{v}_r = \dot{v}$$

Then, the relative equations of motion become:

$$M\dot{v} + C(v_r)v_r + D(v_r)v_r + G(\eta) = \tau \quad (6)$$

If the reader is interested in the derivation of the above formula, you can refer to the

literature: Marine Control Systems: Guidance (Fossen 2002).

Traditional PID Control Methods and Its Limitations

For decades, PID (Proportional + Integral + Derivative) controllers have been successfully used in academia and industry for many kinds of plants. This is thanks to its simplicity and suitable performance in linear or linearized plants, and under certain conditions, in nonlinear ones. A number of PID controller gains tuning approaches have been proposed in the literature in the last decades, most of them off-line techniques. However, in those cases wherein plants are subject to continuous parametric changes or external disturbances, online gains tuning is a desirable choice. This is the case of modular underwater vehicles where parameters (weight, buoyancy, added mass, among others) change according to the tool it is fitted with. In practice, some amount of time is dedicated to tune the PID gains of an underwater vehicle. Once the best set of gains has been achieved the underwater vehicle is ready to work. However, when the underwater vehicle changes its tool or it is subject to ocean currents, its performance deteriorates since the fixed set of gains is no longer valid for the new conditions (Hernández-Alvarado et al. 2016).

Intelligent Control Algorithms

The control issue of underwater vehicles is very challenging due to the nonlinearity, time-variance, unpredictable external disturbances, such as the environmental force generated by the sea current, and the difficulty in accurately modeling the hydrodynamic effect. The well-developed linear controllers may fail in satisfying performance requirements especially when changes in the system and environment occur during the AUV operation since it is almost impossible to manually retune the control parameters in water. Therefore, it is highly desirable to have an AUV controller capable of self-adjusting control parameters when the overall performance degrades. Various advanced control schemes for underwater robots

have been proposed in the literature as some of them are summarized below.

Fuzzy logic control: The theoretical basis of fuzzy logic control is that any real continuous function over a compact set can be approximated to any degree of accuracy by the fuzzy inference system. For control engineering applications, researchers use fuzzy logic to form a smooth approximation of a nonlinear mapping from system input space to system output space. This makes it suitable for nonlinear system control. However, determining the linguistic rules and the membership functions requires experimental data and, therefore, very time-consuming, and the rule-based structure of fuzzy logic control makes it difficult to characterize the behavior of the closed-loop system in order to determine response time and stability.

Neural network (NN) control: Neural networks attracted many researchers because they can achieve nonlinear mapping. Using NN in constructing controllers has the advantage that the dynamics of the controlled system need not be completely known (Yuh 1994). This makes NN suitable for underwater vehicle control. However, NN-based controllers have the disadvantage that no formal mathematical characterization exists for the closed-loop system behavior. The validation of the final design can only be demonstrated experimentally. There are mainly two approaches to use NN for control purpose: learning with a forward model and direct learning. In the former approach, generally, the forward model is trained by the output error or state error and then used for gain derivation (Shi et al. 2006), while in the latter approach, the state or output error is used directly to map the desired control input.

Sliding mode control (SMC): SMC restricts the system states inside a certain subspace of the whole state space and makes them asymptotically converge to their equilibrium point. It requires a raw estimation of the system parameters and an estimation of the system uncertainty for the switching surface design and variable-structure control law design. Even though SMC has been well known for its robustness to parameter variations, it has the inherent problem of chattering phenomenon.

Optimal/robust control: The principles of the optimal/robust control are calculus of variations, Pontryagin maximum principle, and Bellman dynamic programming. However, due to the difficulty of deriving an accurate model of AUV system, it is difficult to apply optimal control directly. Therefore, generally optimal control combined with system identification or robust control is used in AUV control (Zhao and Yuh 2005 and Yuh 1990).

Fuzzy Control Algorithms

Classical control theory is based on mathematical models that describe the behavior of the plant or system under consideration. Fuzzy systems are known for their capabilities to approximate any nonlinear dynamic system (Wang 1992). The main idea of fuzzy control is to build a model of a human control expert who is capable of controlling the plant without thinking in mathematical model terms.

AUV's fuzzy modeling is constructed based upon the input-output data that have been characterized from the open loop system results of the AUV's mathematical model. The input data is considered as the force generated by thruster or pump which moves the AUV in a certain direction, and the output data is considered as the

resulted linear or angular velocity of the vehicle taking into account the coupling effect of the other degree of freedom in that direction.

Figure 2 shows the configuration of a fuzzy model system, which it consists of three fuzzy logic modules that represent surge, pitch, and yaw motions. A Mamdani fuzzy controller was used in this section.

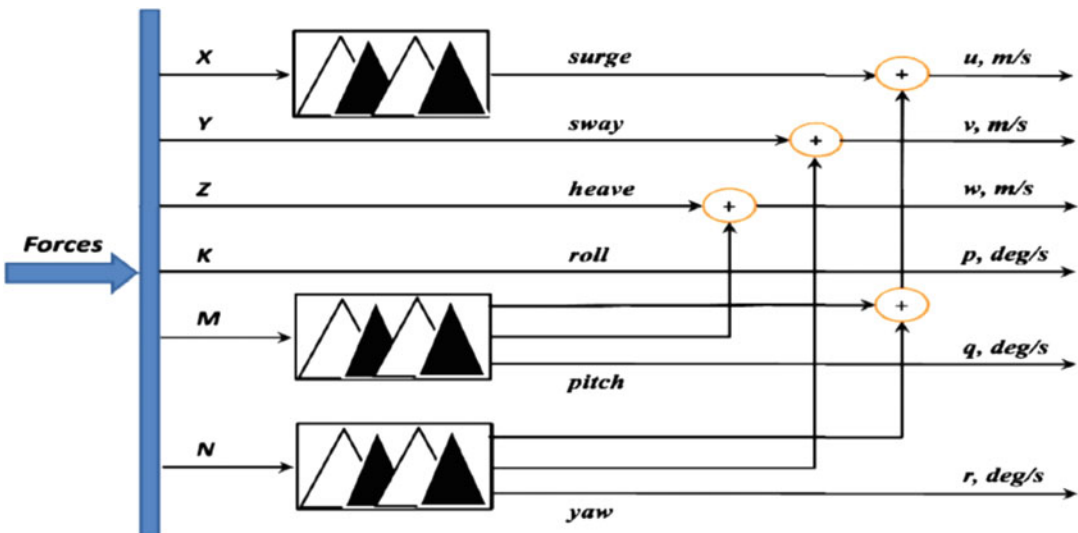
Fuzzy Modeling AUV Surge

For surge motion fuzzy model, the input for the fuzzy logic control is the force required for the thruster, X , to produce the desired motion of the AUV in forward motion corresponding to desired pose. The output of the fuzzy control is the linear velocity in x-direction, u .

Fuzzy Modeling AUV Pitch

For pitch motion fuzzy model, the input for the fuzzy logic control is the force required for the pumps (one pump for up direction and the other one for down) to produce the desired rotation of the AUV in pitch motion corresponding to desired pose. There are three outputs for this fuzzy model.

The first output of the fuzzy control is the linear velocity in x-direction, u , and this output represents the coupling effect on the forward direction, which means the effect of the pitch motion on the



Intelligent Control Algorithms in Underwater Vehicles, Fig. 2 The configuration of fuzzy modeling for the underwater vehicle (Hassanein et al. 2011)

forward motion. The second output represents the coupling effect on heave direction. The third output is the angular velocity about y-axis, q .

Fuzzy Modeling AUV Yaw

For pitch motion fuzzy mode, the input for the fuzzy logic control is the force required for the pumps (one pump for right direction and the other one for left) to produce the desired rotation of the AUV in yaw motion corresponding to desired pose. There are three outputs for this fuzzy model.

The first output of the fuzzy control is the linear velocity in x -direction, u , and this output represents the coupling effect on the forward direction, which means the effect of the yaw motion on the forward motion. The second output represents the coupling effect on sway direction. The third output is the angular velocity about z-axis, r .

Fuzzy Controller Design

The fuzzy controller is shown in Fig. 3, where the inputs to the fuzzy controller are the error and error difference. Generally the depth of AUV can be maintained by controlling the pitch and forward motion. Moving right and left of AUV can be achieved by controlling the yaw angle. The pitch and yaw angles are controlled by controlling the operating time of the pump that is responsible for moving the AUV in that direction.

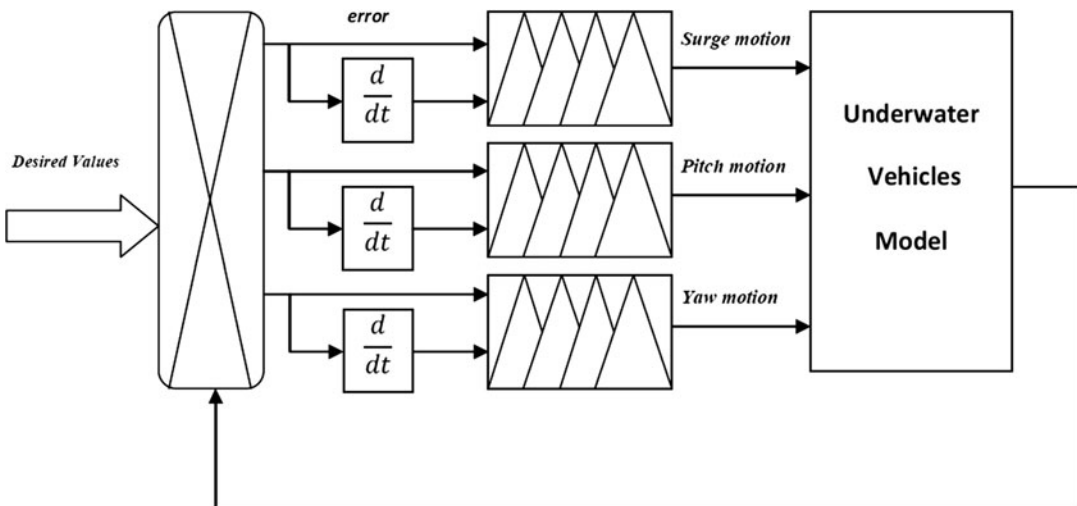
For surge motion, the inputs to the fuzzy controller are “ x ” position error and its error difference. The output of the fuzzy controller is the force required by the thruster. The membership function for this controller is shown in Fig. 4. The total number of rules used for the fuzzy controller is 9.

For pitch motion, the inputs to the fuzzy controller are pitch angle error and its error difference. The output of the fuzzy controller is the time required to operate the pump to rotate the AUV in the required direction. The membership functions are the same for surge fuzzy control with different values.

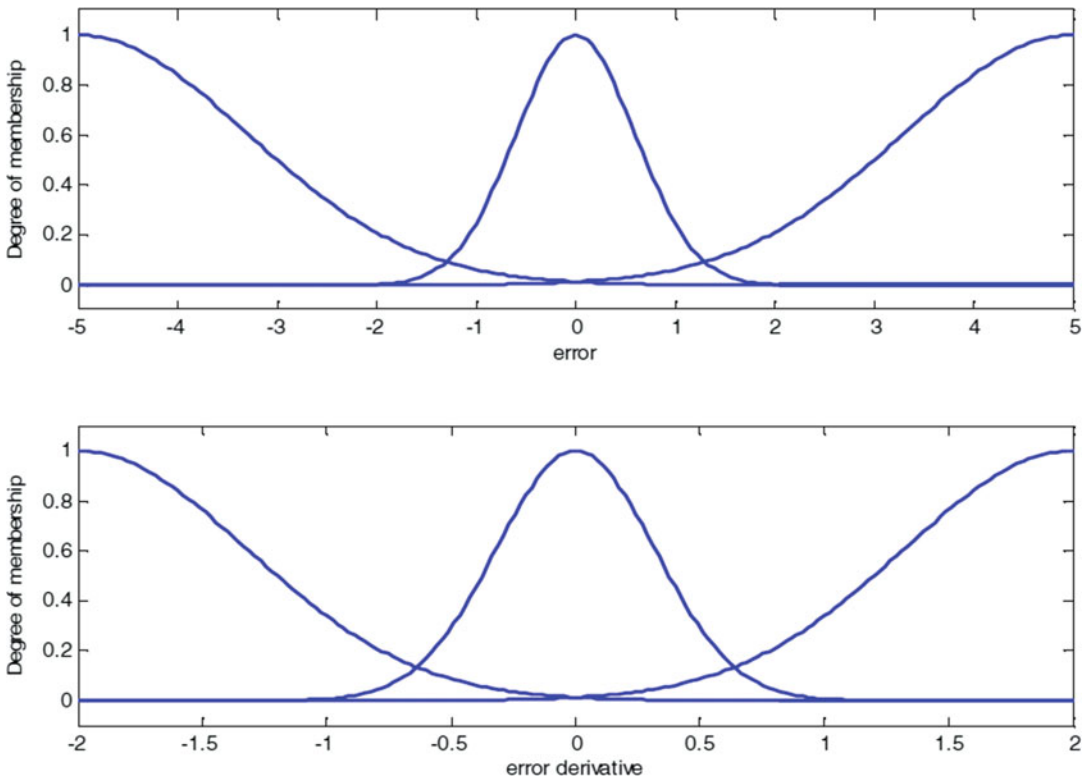
For yaw motion, the inputs to the fuzzy controller are yaw angle error and its error difference. The output of the fuzzy controller is the time required to operate the pump to rotate the AUV in the required direction. The membership functions are the same for surge fuzzy control with different values (Hassanein et al. 2011).

Neural Networks (NN) Control Algorithms

PID control is widely used underwater vehicle control systems. In the absence of accurate mathematical model for underwater vehicle systems, a PID controller may be the best controller, because it is model-free, and its parameters can be adjusted easily and separately. Tuning of PID controllers depends on adjusting its parameters



Intelligent Control Algorithms in Underwater Vehicles, Fig. 3 The structure of the fuzzy control system for underwater vehicle (Hassanein et al. 2011)



Intelligent Control Algorithms in Underwater Vehicles, Fig. 4 Membership functions for the error and the derivative of the error (Hassanein et al. 2011)

(i.e., K_p , K_i , K_d), so that the performance of the system under control becomes robust and accurate according to the established performance criteria. The proposed auto-tuning algorithm is based on NN which exhibit the following characteristics:

1. Parallelism and generalization. A NN is able to produce useful outputs for inputs not provided under the learning phase.
2. Non-linearity. A NN can be linear or not allowing it to represent systems generated by non-linear guidelines.
3. Adaptability. NN is capable of re-adjusting weights and adapting to new environmental situations. This is especially useful when the system offers non-stationary data, that is, the properties involved by the system vary over time.

4. Fault tolerance. When an operational failure occurs on a local part of the network, it lightly affects the global performance.

A block diagram of the auto-tuning control for underwater vehicle with an artificial neural network (NN) is shown in Fig. 5.

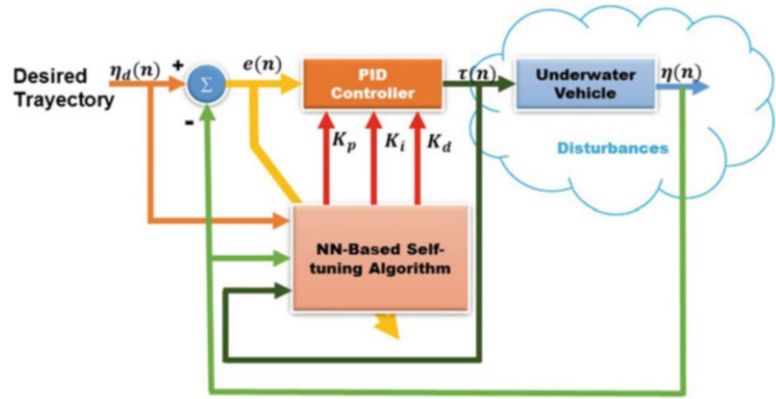
In the discrete time domain, the digital PID algorithm can be expressed as follows:

$$\begin{aligned} \tau(n) = & \tau(n-1) + K_p(e(n) - e(n-1)) \\ & + K_i e(n) \\ & + K_d(e(n) - 2e(n-1) + e(n-2)) \end{aligned}$$

where $\tau(n)$ is the original control signal, $e(n) = \eta_d - \eta$ represents the position tracking error, η_d denotes the desired trajectory, K_p is the proportional gain, K_i is the integral gain, K_d is the derivative gain, and n the sample time.

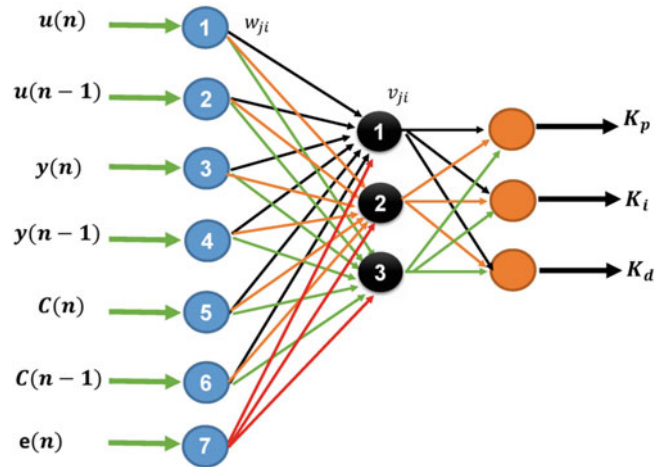
Intelligent Control Algorithms in Underwater Vehicles,

Fig. 5 Block diagram of an auto-tuned PID with artificial NN control (Hernández-Alvarado et al. 2016)



Intelligent Control Algorithms in Underwater Vehicles,

Fig. 6 Block diagram of the implemented backpropagation NN (Hernández-Alvarado et al. 2016)



The algorithm used as auto-tuning is the back-propagation method, chosen for its ability to adapt to changing environments. Operation begins applying the inputs to the network (see Fig. 6), and this is propagated from the first layer to the hidden layers, up to produce output (K_p , K_i , and K_d). The output signal is compared to the desired output and an error signal is calculated for each of the outputs; this is shown in Fig. 5. The error outputs backpropagate, starting from the output layer, to all neurons in the hidden layer that contribute directly to the output; however, the hidden layer neurons receive only a fraction of the total error signal. This process repeats iteratively, layer by layer, until all neurons in the network have received an error signal describing its relative contribution to the total error.

Figure 6 presents the topology of the NN used to auto-tune the PID controller gains implemented on the ROV.

$u(n)$ and $u(n-1)$ are reference inputs (desired trajectory), $y(n)$ and $y(n-1)$ are reference outputs (real trajectory), $C(n)$ and $C(n-1)$ correspond to the control signals, w_{ji} are the weights of the hidden layer, and v_{ji} are the weights of the output layer.

The criteria used to minimize the error:

$$E(n) = \frac{1}{2} \sum_{k=1}^t (y_r(k) - y(k))^2$$

The adjustments of weighting coefficients w_{ji} and v_{ji} can be made by means of the expressions:

$$w_{ji}(n+1) = w_{ji}(n) + \left(a \frac{\partial e_y}{\partial e_u}\right) \delta_j^2 x_j$$

$$v_{ji}(n+1) = v_{ji}(n) + \left(a \frac{\partial e_y}{\partial e_u}\right) \delta_j^1 h_j$$

where a is the learning coefficient, $w_{ji}(n+1)$ is a vector of weights for the hidden layer, $v_{ji}(n+1)$ is the vector of weights of the output layer and equivalent gain $\frac{\partial e_y}{\partial e_u}$ is unknown.

Sliding Mode Control Algorithms

Because of the nonlinearity and the unpredictable operating environment of the AUVs, many design parameters must be considered during the design of AUVs control system. Indeed, the high frequency oscillating movement can seriously affect the performance of sensors, especially optical and acoustical sensors.

In brief, the main factors that make the control of AUVs difficulty are: (1) the highly nonlinear, time-varying dynamic behavior of the AUVs; (2) uncertainties in hydrodynamic coefficients; and (3) disturbances by ocean currents. To remedy these aforementioned problems and enhance the AUVs performance along with strengthen robustness, adaptability, and autonomy, it is necessary that the motion control system has the ability of learning and self-adaptation.

The AUV dynamic in (4) can be rewritten as

$$M\ddot{q} + C(\dot{q})\dot{q} + D(\dot{q})\dot{q} + G + \tau_d = \tau \quad (7)$$

Moreover, q is the configuration and τ_d represents environmental forces and/or disturbances. To make the AUV follow a prescribed desired trajectory $q_d(t)$, we define the tracking error $e(t)$ and filtered tracking error $r(t)$ by

$$e = q_d - q, \quad r = \dot{e} - \Lambda e$$

with $\Lambda > 0$ a positive definite design parameter matrices.

The AUV dynamics are expressed in terms of the filtered error as

$$M\dot{r} = -Cr + f(x) + \tau_d - \tau$$

where the unknown nonlinear function of AUV dynamic is given by

$$f(x) = M(q)(\ddot{q}_d + \Lambda \dot{e}) + C(\dot{q})(\dot{q}_d + \Lambda e) + D(\dot{q})\dot{q} + G$$

One may define $\equiv [e^T, \dot{e}^T, \ddot{q}_d^T, \ddot{q}_d^T]^T$.

A general sliding mode controller structure is based on

$$\tau = \hat{f} + K_v r - v(t)$$

with $\hat{f}$ an estimate of $f(x)$, $K_v r = K_v \dot{e} + K_v \Lambda e$ an outer PD tracking loop, an $v(t)$ an auxiliary signal as

$$v(t) = -(F(x) + \eta) \operatorname{sgn}(r)$$

where $\operatorname{sgn}(\cdot)$ is the sign function and $F(x)$ is a known function which can be computed using the boundedness property of AUV's dynamics. This term is used to maintain the robustness in the face of external forces, disturbances, and modeling error. η is a design parameter and can be selected as a small value. Employing this controller, the closed-loop error dynamics are

$$M\dot{r} = -Cr - K_v r + \tilde{f} + \tau_d + v(t)$$

where the function approximation error is defined as

$$\tilde{f} = f - \hat{f}$$

so that $\|\tilde{f}\| \leq F(x)$ and $\|\cdot\|$ is Frobenius norm.

The stability and numerical simulation of the introduced sliding mode control system can be obtained in the work of Tabar et al. (2015).

Optimal Control Algorithms

To achieve better robustness and handling coupling in the system dynamics, MIMO-based techniques such as LQR method has been reported. The literature revealed that LQR provides better performance over classical control scheme such as

PID though with higher design and implementation complexity. On the other hand, it represents the simplest model-based MIMO controller in terms of design and implementation when compared to others such as H-infinity, μ -synthesis, and nonlinear controllers. However, LQR control is a well-known optimal control method, and the optimality of the resulting compensator is a subject of appropriate design parameters' (Q and R) selection. However, one of the common challenges associated with LQR and generally with the majority of modern control techniques is their dependency on selection of appropriate design parameters. The selection is usually based on technical guess followed by several iterative tunings which are mostly based on trial and error for a given question. Considering the complexity and dimension of the underwater vehicle, this conventional trial and error design approach would be time consuming in proportionate with experience of the designer, and often limits the achievable control performance objectives which are usual conflicting. In order to overcome this, the control problem is formulated as a multi-objectives optimization task, and a proposed multi-objectives differential Evolution algorithm is applied to achieve an optimal/suboptimal compromise between the performance objectives (Alvis et al. 2007; Castillo-Effen et al. 2007; Tijani 2012).

Optimization Problem Formulation

The optimization problem is to determine the set of decision vectors Φ , which in this context is the weighting matrices Q and R 's parameters: Φ

$$\Phi := [\Phi_1 \quad \dots \quad \Phi_{n+m}]$$

that minimizes the cost function:

$$\hat{J} = \int_0^{\infty} (\hat{x}^T Q \hat{x} + \delta^T R \delta)$$

And set of performance objectives of size Ω :

$$\min f_i(\Phi), \quad i = 1, 2, \dots, \Omega$$

subject to $Q \geq 0$ and $R \geq 0$ with $0 < \Phi_j < H_j$ where H_j is the upper bound on the decision parameter Φ_j .

The problem is casted as nonconstraint optimization problem, meanwhile, depends on design goal and application requirements; it is possible to introduce additional constraints such as frequency response parameters or actuator limitation. Figure 7 shows the proposed design method.

As shown in the Fig. 1, the performance objective function is given by the response of the full-state feedback system formed for a given decision parameter set. In this study, five control objective functions are defined for the pitch angle tracking control as follows:

$$f_i(\Phi) := \left[t_s^\theta, OS^\theta, \max t_s^{(u,w,q)}, \max \beta^{(u,w,q)}, \max \delta \right]$$

where t_s^θ and OS^θ are the settling time and percentage overshoot of the pitch angle θ , respectively, $\max \beta^{(u,w,q)}$ is the maximum off-axis (other states, $\{u, v, w\}$) settling time, $\max \beta^{(u,w,q)}$ is the maximum peak of the off-axis response, and $\max \delta$ is the maximum control signal expended.

Application to the UUV System

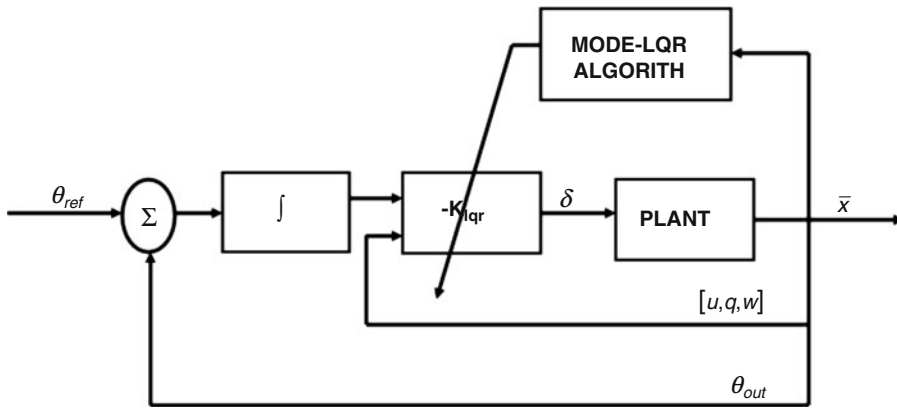
In tracking and regulation control problem as shown in the Fig. 7, the system states are first augmented with two extra integral states vector to improve the tracking performance. The new state vector $\bar{x}_i$ is given as:

$$\bar{x}_i := \left[u, q, w, \theta, \int_0^{\infty} \theta dt \right]$$

The augmented state space model is then expressed as:

$$\dot{\bar{x}}_i := A_i \bar{x}_i + B_i \delta$$

and the control law becomes:



Intelligent Control Algorithms in Underwater Vehicles, Fig. 7 Block diagram of the proposed MODE-Based LQR controller synthesis (Tijani and Budiyo 2016)

$$\bar{\delta}_i = -K\bar{x}_i$$

As indicated in Fig. 7, the design procedure involves placing the MODE in the LQR conventional design loop for optimal tuning of the Q and R matrices, and can be summarized step-wise as follows:

- Step 1:** Specify the optimization parameters: population size, NP , decision vector size, Z , DE cross-over constant, CR , and generation size, GEN .
- Step 2:** Specify the dimension of the Q and R matrices and initialize population P of size NP for the Q and R matrices' parameters.
- Step 3:** Test for the positive definiteness of the initialized matrices, and remove any unfit parameters set(s); repeat step 2 to complete the size of population.
- Step 4:** Compute the objective function and extract the objectives vector for the population.
- Step 5:** Start the optimization process: while the stopping criteria are untrue, e.g., repeat steps 6 to 9 else go to step 10 GEN k while.
- Step 6:** Generate new child vector from randomly selected three vectors from the old population using the DE operation of mutation and cross-over processes. Test the positive definiteness of the child vector and compute its objective function.

Step 7: Compare the child vector with the parent vector for Pareto dominance. That is, determined if the child vector dominates the parent vector. If not, repeat step 6 to 7, else proceed to step 8.

Step 8: Update the new population with the dominant child vector.

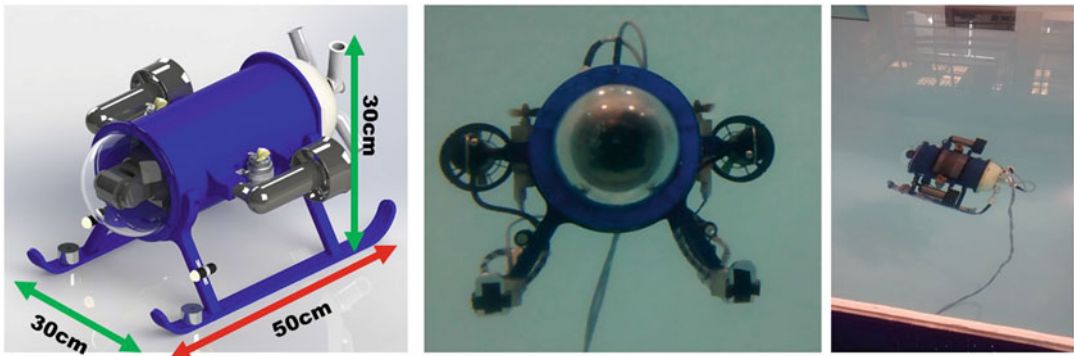
Step 9: Compute the stopping criteria and update the generation count: $k = k + 1$.

Step 10: Stop the optimization and report the Pareto-dominance candidates.

The numerical simulation of the introduced optimal control system can be obtained in the work of Tijani and Budiyo (2016).

Key Applications

In the work of Hernández-Alvarado et al. (2016), the intelligent control was implemented on an underactuated mini-ROV. This vehicle is a ROV developed in CIDESI named Nu'ukul Ja (which in the Mayan language means "water instrument"). Its dimensions are: 50 cm long, 30 cm wide, and 30 cm height; as shown in Fig. 8. It has a cylindrical pressure chamber of 15 cm diameter where the major part of the electronic architecture is placed. The total weight of the ROV is 10 kg. According to the experimental environment, it was placed in a pool of 2.5 m where both PID



Intelligent Control Algorithms in Underwater Vehicles, Fig. 8 Underactuated mini-ROV under water (Hernández-Alvarado et al. 2016)

controllers (auto-tuned and conventional) were implemented.

The gains of the conventional PID control were obtained by means of the NN. The ROV was requested to get the set point of 1 m depth by using the Auto-tuned PID controller. Once the ROV reached the stability and the PID gains, computed by the NN, became stationary, these gains were programmed into the conventional PID as K_p , K_d , and K_i . This way the conventional PID gains were tuned. It is important to remark that once the conventional PID was tuned, the gains remained constant along the experiment, even when the disturbances took place, unlike the auto-tuned PID controller wherein gains were dynamically changing to attain better performance along the experiment.

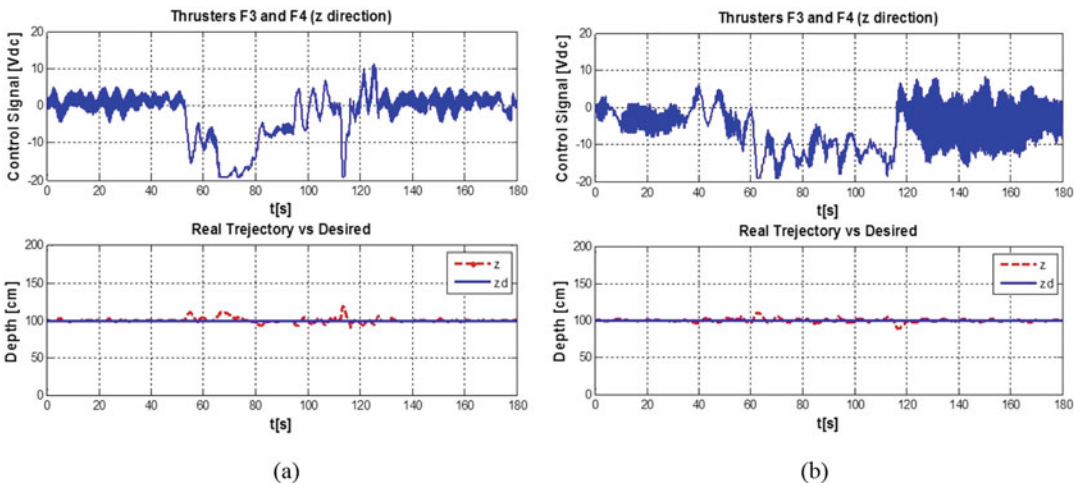
The controls were evaluated by performing a data capture of 3 m, once the ROV was placed 1 m underwater. In the first minute, nondisturbance took place. After this time, the weight was increased by 400 g as disturbance until 2 min mark. Finally, in the last minute of data capture, the weight was removed. Figure 9 shows the desired trajectory (solid line) versus the real path (dotted line) and the control signals for thrusters (Hernández-Alvarado et al. 2016).

Apparently, the control signal given by the auto-tuned PID (shown in Fig. 9), seems to be more active than the conventional PID's signal; though, the root mean square (RMS) value of each

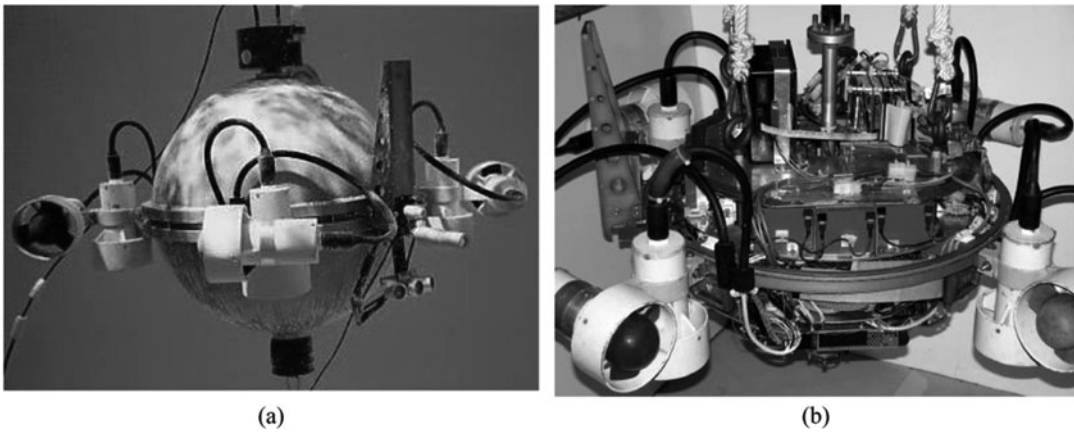
one (the complete experiment) shows that the RMS of the conventional PID is 6.8874 whereas in the auto-tuned PID is 6.6781. The auto-tuned PID has a 3.038% of energy saving against the conventional PID.

The experimental work (Zhao and Yuh 2005) of the proposed controller was carried out on ODIN III, which is shown in Fig. 10. It is a six-DOF autonomous underwater robot developed by the Autonomous Systems Laboratory of the University of Hawaii. It is a close-framed sphere-shaped vehicle that makes its dynamics in each direction nearly identical. It has eight thrusters: four horizontal and four vertical, which make ODIN III capable of six-DOF maneuvering and also have thrust redundancy for fault tolerance purpose. It also has various navigation sensors including eight sonar sensors, a pressure sensor, and an Inertial Measurement Unit (IMU). The eight sonar sensors are used to measure the displacements in the horizontal plane, the pressure sensor is used to measure the depth of the vehicle, and the IMU is used to measure the angular displacements. A Kalman filter is used to suppress the sensor noise and to estimate the translational and angular velocities. All the tests were done in the diving pool of Kahanamoku Pool at the University of Hawaii.

The effectiveness of the control system was experimentally investigated by implementing three controllers: PID, PID plus DOB, and



Intelligent Control Algorithms in Underwater Vehicles, Fig. 9 (a) Conventional PID controller, (b) auto-tuned PID (Hernández-Alvarado et al. 2016)

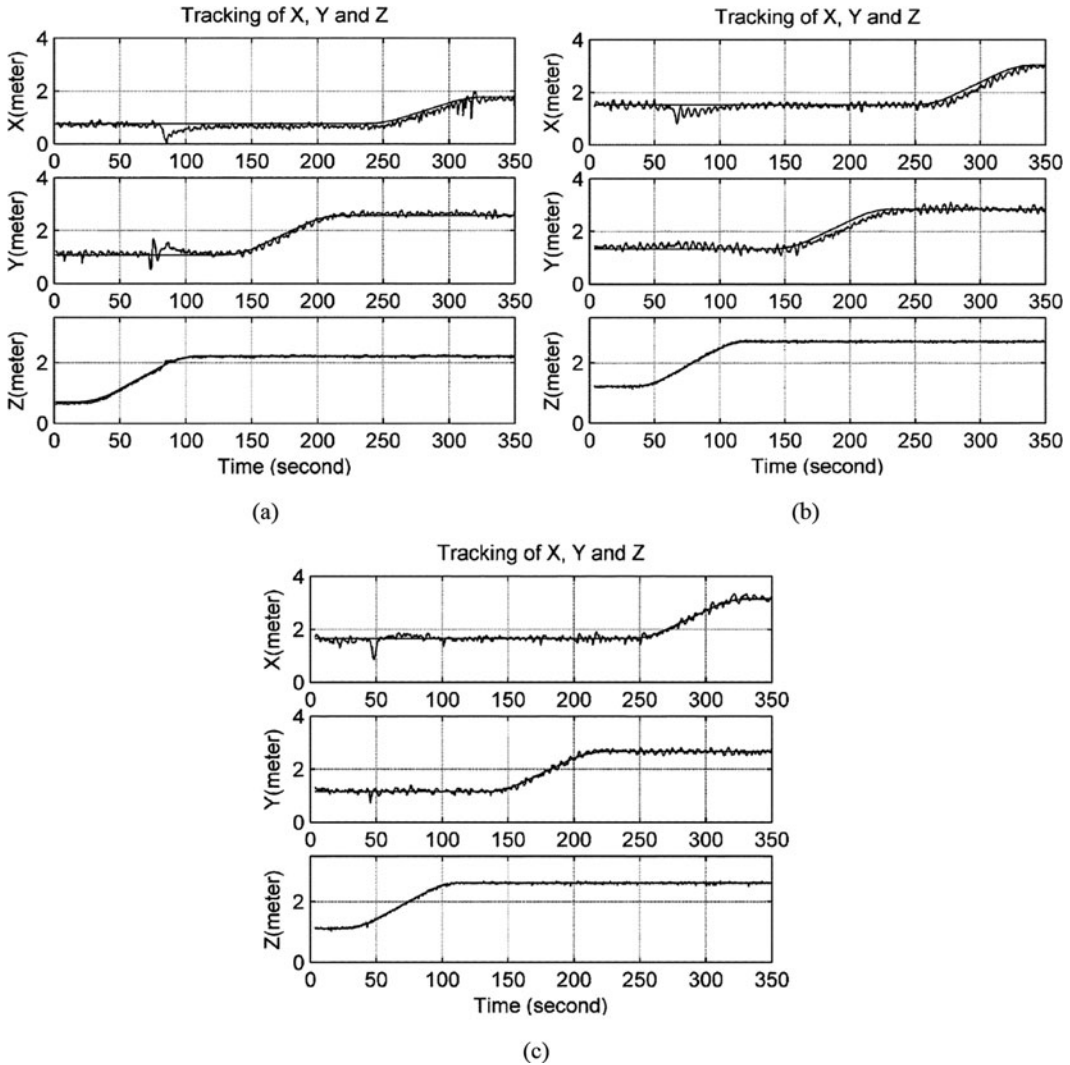


Intelligent Control Algorithms in Underwater Vehicles, Fig. 10 Omni-Directional intelligent navigator (ODIN III), (a) View in water, (b) Inside view (Zhao and Yuh 2005)

ADOB on the autonomous underwater vehicle, ODINIII. Figure 11 shows tracking performance in x, y, and z.

The ADOB consists of the regressor-free adaptive control and DOB, taking advantages of DOB robustness with respect to external disturbances and modeling errors, and the regressor-free adaptive controller’s robustness with respect to uncertainties in the system model. The ADOB controller has the capability of self-tuning control gains and adapting to changes in the system and

environment while PID requires a lengthy pre-tuning process for satisfactory performance. The PID would need retuning control gains when the performance degrades due to changes in the system and environment. However, it is almost impossible to retune control gains of underwater robots until they are brought up to the surface where hydrodynamics would change again. Therefore, as shown in the experimental result (Fig. 11), the ADOB controller is promising for underwater vehicles, especially when the



Intelligent Control Algorithms in Underwater Vehicles, Fig. 11 (a) PID control with external disturbance, (b) PID plus DOB control with external disturbance, (c) ADOB control with external disturbance (Zhao and Yuh 2005)

performance degrades or fails by PID type controllers (Zhao and Yuh 2005).

Cross-References

- [Advanced Control](#)
- [Automatic Control](#)
- [Fuzzy Control](#)
- [Optimal Control](#)
- [Robust Control](#)

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Intensified Silicon Intensifier Target (ISIT)

► [Underwater Lander](#)

Internal Rate of Return (IRR)

► [Christmas Tree](#)

Internal Tieback Connector (ITBC)

► [Steel Pipelines and Risers](#)

Internal Turret Single-Point Mooring (SPM) System

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Synonyms

[Mooring system](#); [Positioning system](#); [Single-point mooring system](#)

Definition

Internal turret single-point mooring system is a type of single-point mooring device and belongs to the turret type single-point mooring device. This type of single-point mooring is moored to the seabed through uniform or grouped anchor chains on a turret inside the hull. The FPSO hull rotates 360° around the internal turret under wind, wave, and flow environmental loads. It has a “weathervane effect,” which makes the hull adjust to the direction with smallest load. Because the SPM pontoon and turret are installed inside the hull, it is called the internal turret SPM system. The crude oil enters the oil storage tank through a subsea pipeline into a rotary sealed joint in the internal turret.

Development History

The world’s first SPM pontoon was designed by IMODCO in the United States in 1959 and put into production in the port of Detio (Yifeng 2012), Sweden. It is used as a deep water transport dock. The pontoon is connected to the sea floor by



Internal Turret Single-Point Mooring (SPM) System, Fig. 1 The world's first SPM pontoon (Yifeng 2012)

mooring lines, and the oil tanker is connected to the SPM device by steel cables (Fig. 1).

In the early 1980s, SBM Company first proposed a turret concept based on the principle of “weathervane effect” (Yifeng 2012). In 1985, an external turret was installed on the bow of a 140,000-ton floating storage tanker, which makes the concept formally applied to reality (Yifeng 2012). The advantage of the outer turret SPM is that the tanker’s weathervane effect is significant, which makes the rotation to the position with the smallest environmental load become easy, and does not occupy the hull storage space, so the oil tanker has high oil storage efficiency; the shortcoming is that the bow movement is the most responsive, especially in deep water mooring, and the riser and turret of the outer turret device are all within the range of wave slamming, which easily causes damage of the riser and the turret. Therefore, the outer turret SPM is only suitable for medium-deep waters with better marine environment.

In the early 1990s, the internal turret SPM system was proposed, in order to overcome the shortage of out turret SPM and meet the development demand of deep water oil and gas fields which are over 100m. (Zhigang and Yanping 2006), the internal turret SPM system was proposed. The early internal turret SPM system² is generally the permanent type. In order to adapt the FPSO mooring demand under the harsh sea

conditions, the development trend is the internal turret SPM system of detachable type.

Operational Principle and Component

The Internal Turret SPM System of Permanent Type

The early internal turret mooring system was not detachable. Generally the SPM is designed under the sea condition once in 100 years, to provide sufficient weathervane effect and mooring force under severe sea conditions and ensure the FPSO has mooring and transmission capabilities under most sea conditions. Some internal turret mooring systems of permanent type are equipped with locking devices that are tightened as necessary to limit the sudden movements of the hull.

The internal turret mooring system installed in the Marlim Leste field in Camps Bay (Zhigang and Yanping 2006), Brazil, in 2007 would be the world’s largest internal turret mooring system of permanent type. It can accommodate up to 75 hoses and is moored with 8 mooring legs in combination of polyester ropes and anchor chains.

The Internal Turret SPM System of Detachable Type

The detachable type is the development trend of SPM system. The detachable internal turret mooring system can be quickly disengaged and return

according to the working sea conditions, and can be quickly released under extreme sea conditions to avoid dangerous sea conditions that may cause personal injury and damage to floating structures. It is more suitable for offshore oil fields in harsh environments and complex sea conditions. The detachable internal turret mooring system is mostly in the form of internal turret pontoon. This single-point mooring system can be installed at sea at an early stage, which shortens the time for FPSO return operations. In recent years, the detachable internal turret SPM system is gradually designed under the conditions that FPSO is not released under the sea conditions of 100 years, making the mooring system to have the function that does not have to be released when faced with typhoon, further ensuring the personal and platform's safety and normal production. Common detachable internal turret mooring system includes: pontoon turret mooring system, submerged turret handling system, and submerged turret production system (Shan et al. 2008).

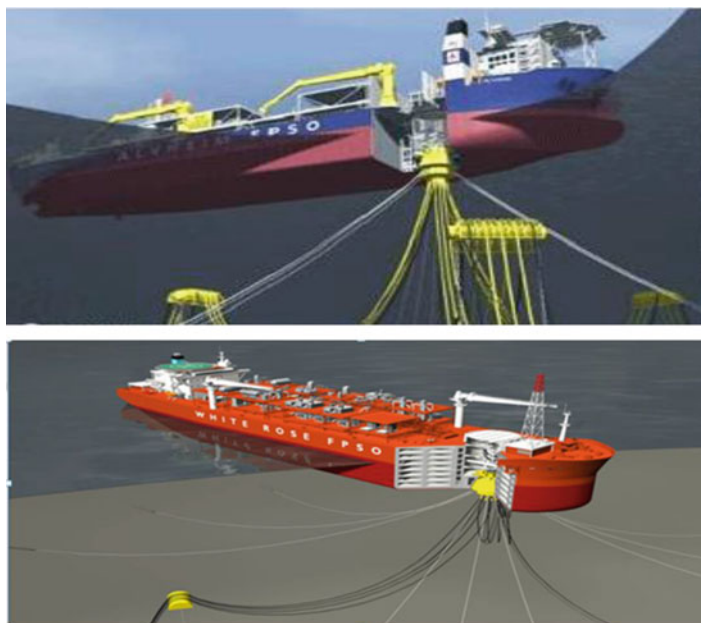
The internal turret mooring system consists of a pontoon and an internal turret which is inside the bow and connected by structural connectors. The pontoon can provide sufficient buoyancy to

withstand the weight of the mooring legs and risers. The pontoon turret mooring system is capable of repairing critical components such as manifolds under severe sea conditions. The installation of turret can be carried out in dock, reducing the amount of on-site installation work at sea. The submerged turret handling system consists of an underwater buoy and a complete turret system. The inside of the conical buoy moored to the sea floor is a turntable that connects the mooring chain and the riser system. The outer structure of the pontoon can be rotated around the turret with the hull. The submerged turret handling system can be released in any climatic conditions, and it also has a function of simple quick release and return for high safety. The submerged turret production system is a production of a submerged turret handling system and mooring technology combined with a multifunction rotary joint. It is a new generation of internal turret SPM system that integrates mooring system, turret device, and rotary joint. Figure 2 shows the detachable internal turret SPM system (<http://image.baidu.com/search/index?tn=baiduimage&ps=1&ct=201326592&lm=-1&cl=2&nc=1&ie=utf-8&word=FPSO>; Zhao 2009).

The submerged internal turret SPM mainly includes: the slip ring stack, the pontoon, the

Internal Turret Single-Point Mooring (SPM) System, Fig. 2

The internal turret SPM system of detachable type (Shan et al. 2008)



locking device, the mooring system, and so on. The details are as follows:

(1) The Slip Ring Stack

The slip ring stack, also known as the rotary joint assembly, is one of the key equipments for the mooring system. It consists of a well-flow rotary joint, an electric rotary joint (electric slip ring), a common rotary joint (common slip ring), and a supporting hydraulic control system. The rotary joint assembly is composed as shown in Figs. 3 and 4.

1. The well-flow rotary joint: The inlet end is connected to the underwater flexible hose, and the outlet end is connected to the FPSO production processing module inlet line to form a crude oil output passage. A hydraulic oil seal system is arranged on the well-flow rotary joint to ensure normal operation without leakage.
2. The electric rotary joint: Power is supplied to the platform through the electric rotary joint

and power cables for use on equipment and facilities.

3. The hydraulic rotary joint: The hydraulic rotary joint provides hydraulic power to control the emergency shutoff valve.
4. The emergency shutoff valve: It is used to shut off the oil passage in an emergency state.

(2) The Pontoon

The pontoon is conical, and its schematic diagram is shown in Fig. 5. The upper part of the pontoon is provided with a bearing assembly, and the lower part is connected to the mooring line through the lifting-eye to transmit the mooring tension. The upper part is fixed to the FPSO by a special hydraulic locking device. The FPSO rotates or drifts around the mooring center under the action of environmental loads such as wind, wave, and current through the bearing assembly on the pontoon.

(3) The Locking Device

The locking device is located in the single-point compartment, clamping and locking the pontoon, so that the FPSO is integrated with the SPM to achieve safe production. The locking device is the most critical device of a SPM system. In actual production operations, the locking device must maintain sufficient preload.

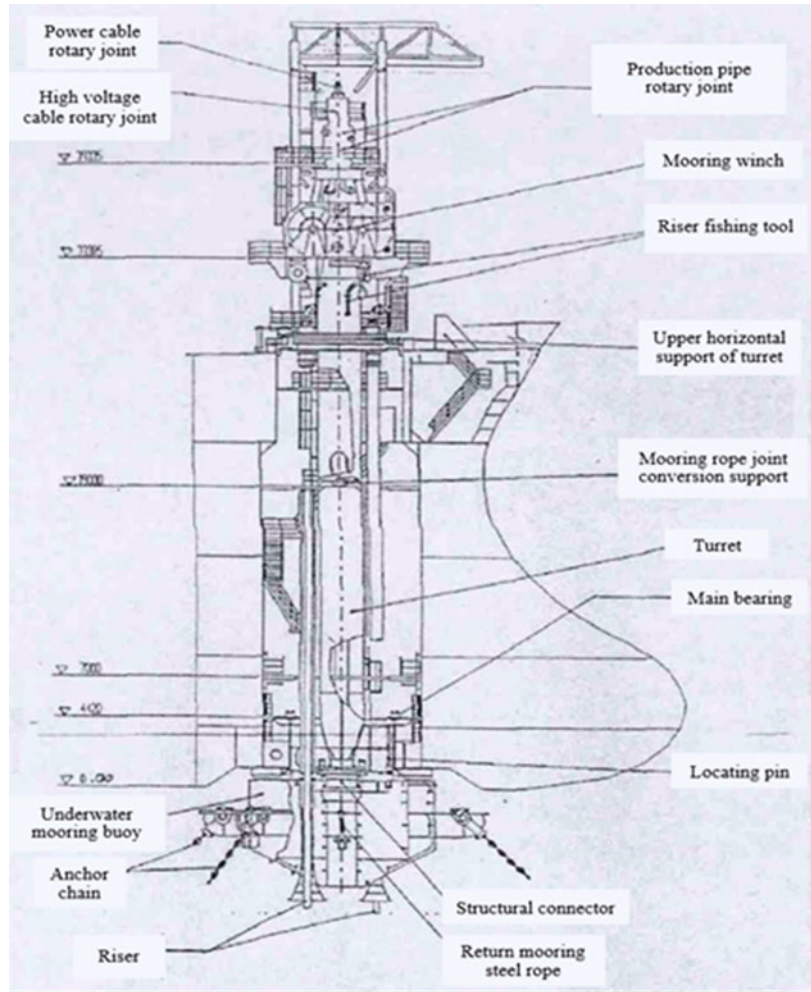
(4) The Mooring System

Most of the mooring systems adopt asymmetrical arrangement of 3×3 and 3×4 . This mooring arrangement has better adaptability to sea areas with bad conditions. In the design of the mooring system, the weighting block is generally added to the mooring line which section is laid on the seabed in order to increase the recovery stiffness and improve the resilience of the mooring system. According to this design, the length of anchor chain laid on the seabed could be reduced and the construction and installation could be more facilitated.



Internal Turret Single-Point Mooring (SPM) System,
Fig. 3 The slip ring stack

Internal Turret Single-Point Mooring (SPM) System, Fig. 4 The components of slip ring stack

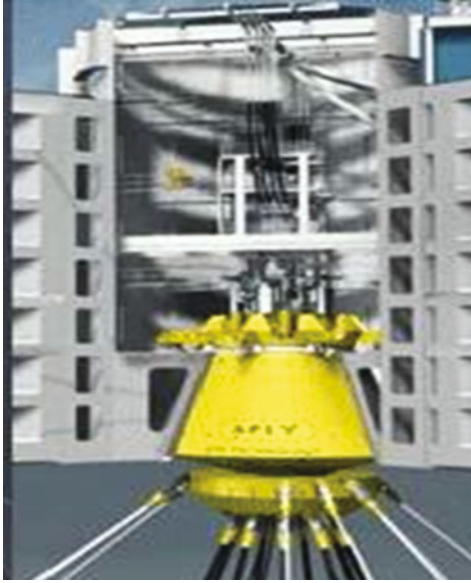


Application

The internal turret SPM is widely used, and most of the FPSOs worldwide use the internal turret SPM. The internal turret SPM device is located in the bow opening of the hull. The advantage is that the diameter of the turret can be designed to be large, which can provide sufficient space for the arrangement of equipment and manifolds. In addition, the turret is located inside the hull, which reduces the environment loads and effectively protects the SPM structure. The disadvantage is that the turret is installed inside the hull, affecting the arrangement of hull structures and the strength, as well as reducing the effective storage capacity (<http://image.baidu.com/search/index?>

[tn=baiduimage&ps=1&ct=201326592&lm=1&cl=2&nc=1&ie=utf-8&word=FPSO](http://image.baidu.com/search/index?tn=baiduimage&ps=1&ct=201326592&lm=1&cl=2&nc=1&ie=utf-8&word=FPSO)). In addition, the effect of the weathervane of mooring tanker is restricted by the turret. When the wind wave is large, the azimuth speed at which the tanker turns to the least force direction is slow. The internal turret SPM system is suitable in moderate sea conditions or in harsh conditions of deep sea operations.

At present, the world's largest internal turret single-point mooring device is built by Dubai DryDocks World (Zhao 2009; Shengfa 2012). This SPM unit is 100 m high, weighs 4300 tons, and is connected to the sea floor by 16 mooring lines. This mooring system positions the 600,000-ton Prelude FLNG in a gas field of approximately



Internal Turret Single-Point Mooring (SPM) System, Fig. 5 The pontoon schematic diagram

200 km away from the northwest coast of Australia for 25 years.

Cross-References

- ▶ External Turret Single Point Mooring System
- ▶ Mooring System
- ▶ Positioning System
- ▶ Single-Point Mooring System
- ▶ Soft YOKE Single Point Mooring System

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International Maritime Organization (IMO)

- ▶ Decommissioning of Offshore Oil and Gas Installations
- ▶ Net Structures: Biofouling and Antifouling

International Organization for Standardization

- ▶ Shallow Foundations

International Standards Organization (ISO)

- ▶ Decommissioning of Offshore Oil and Gas Installations

International Towing Tank Conference (ITTC)

- ▶ Ice Tank Test

Interpolation Technique by Small Strain (RITSS)

- ▶ Suction Piles

Intersymbol Interference (ISI)

- ▶ Underwater Acoustic Communication

Introduction to Shipbuilding (Shipyard)

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Synonyms

CAD (Computer-aided design); CAM (Computer-aided manufacturing); CIMS (Computer integrated manufacture system); CNC (Computer numerical control); NC (Numerical control cutting)

Definition

Shipbuilding technology is generally divided into hull construction process, outfitting process, and painting process.

The hull construction process is the process of manufacturing hull components, then assembling and welding them into intermediate products (blocks) and then hoisting them to the berth for assembly into the hull. Its operations generally include ship hull lofting, ship hull marking, hull processing, hull assembly, berth assembly, and ship launching.

The shipbuilding industry generally refers to the mechanical and electrical installations, operating equipment, living facilities, various attachments, and cabin decorations other than the main hull and superstructure, collectively referred to as outfitting projects. It uses not only steel, but also nonferrous metals such as aluminum and copper and their alloys, and a wide range of nonmetallic materials such as wood, engineering plastics, cement, ceramics, rubber, and glass. Outfitting operations involve as many as dozens of work types such as assemblers, welders, carpenters, copper workers, fitters, and electricians. Therefore, ship outfitting can be divided into mechanical outfitting, electrical outfitting, piping outfitting, hull iron outfitting, and woodwork

according to the professional content; if according to the stage of outfitting work, it can be divided into outfitting parts manufacturing (purchasing), outfitting palletizing, block outfitting, block outfitting, and interior outfitting, etc.; according to the regional outfitting method, it can be divided into engine room outfitting, deck outfitting, cabin outfitting, and electrical outfitting.

The painting operation is to remove rust and apply various paintings on the inner and outer surfaces of the hull and the outfitting parts according to the technical requirements to separate the metal surface from the corrosive medium and achieve the purpose of anti-corrosion treatment. It generally includes manufacturing-level production operation systems such as steel pretreatment, block painting, block painting, berth painting, and dock painting.

The modern shipbuilding method is based on the region, so that the three different types of hull construction, outfitting, and painting are coordinated and organically combined to form an integrated construction technology of hull, outfitting, and painting, that is, the regional shipbuilding method, and to establish the production process flow shipbuilding.

Introduction

The Content and Process Flow of Shipbuilding Technology

Hull Lofting

Lofting is the first process in the shipbuilding production process. Although the workload is very small compared with the entire shipbuilding, it is a very complicated and precise work, otherwise it will directly affect the quality of subsequent processes and even the final product.

Manual lofting started in the mid-1940s. The main work contents are: hull line smoothing, drawing frame line drawing, hull component expansion, drawing construction drawing and making prototypes and sample boxes for construction, and determining additional processing and assembly allowance. Full-scale lofting is performed at the lofting table at a ratio of 1:1,

while proportional lofting is performed at a certain reduced proportion on the lofting table. These two methods are traditional lofting methods which are manual operations, labor intensive, and poor accuracy. In addition, a large area of prototypes and a large amount of materials are needed to manufacture prototypes and prototype boxes. The methods are high cost and low efficiency, which is not conducive to the development of shipbuilding in the direction of mechanization and automation.

With the development of electronic computer technology and shipbuilding technology, lofting has developed from manual lofting to mathematical lofting. It is now possible to implement computer smoothing and lofting of the whole ship's line type and structure. It can also modify the line type and structural arrangement by man-machine dialogue to realize graphic display and automatic drawing and can automatically store the obtained data in the database for the next process such as numerical control drawing, numerical control marking, numerical control cutting, thus fundamentally eliminating manual lofting (Liu et al. 2011).

Hull Marking

Marking is the subsequent process of lofting. It is to copy the shape of the hull components after lofting to the steel plate or block steel according to the sketch, sample bar, template, sample box, sample diagram or projection diagram, CNC paper tape, etc., and mark the processing and assembly symbols, construction information for use in subsequent processes (Huang 2013).

At present, most shipyards have generally adopted CNC marking. In addition, the marking device attached to the CNC cutting machine has become commonplace. The most commonly used is to install a marking mechanism or a line drawing nozzle on the CNC cutting machine, which can simultaneously mark material at the same time of cutting. Moreover, nesting also belongs to the category of marking materials (Liu et al. 2011). At present, it is generally used for interactive and automatic graphics nesting with the help of large computers.

Hull Steel Processing

The hull steel processing is to process the pre-treated steel into the specifications and shapes required by the design with machinery and tools after marking. Generally speaking, the processing of ship hull steel can be divided into two categories: cold processing and hot processing according to the processing characteristics: it can be divided into edge processing (including planning, cutting, shearing, blanking, etc.) and forming processing (including rolling, pressing, bending, orthopedics, etc.) according to the shape of the component (Li et al. 2006).

Steel Pretreatment The surface of steel generally has defects such as oxide scale, rust, local unevenness, warping, or distortion, and it is necessary to perform operations such as leveling, rust removal, applying protective primer, and drying on the steel. These operations are collectively referred to as steel pretreatment.

Steel Cutting In modern shipyards, the cutting of ship steel plates has shifted from manual cutting and photoelectric tracking cutting to CNC cutting. In addition to adopting new methods, the development of cutting technology has also turned to automation and even unmanned development through numerical control, computers, and robots. In terms of cutting technology, there are mechanical cutting, gas flame cutting, plasma cutting, laser cutting, and water jet cutting.

Edge Processing of Hull Components The edge processing of hull components generally includes three processing operations: firstly, mechanical shearing or chemical or physical cutting method is used to cut the hull components from the raw materials according to the construction information; secondly, according to the technical requirements of welding and assembly welding, groove processing is carried out on the components; and thirdly, according to the design requirements, the free edges and manholes of some components are polished with grinding wheels, thus obtaining hull components meeting different technical requirements (Ji 2005).

Forming and Processing of Hull Components

After edge processing of hull components, for some components with spatial shapes such as bending, angle folding or edge folding, bending processing methods should be used to bend or bend them into the required spatial shapes. The forming process of hull components is to process the parts (flat panels, profiles, etc.) after edge processing into the actual shapes required by the components. Generally speaking, the bending operation accounts for about 10% ~ 18% of the hull steel processing work, while the bending restriction of profiles accounts for 9% ~ 16%.

① Sheet forming processing: marine sheet metal has complex line types, and its processing has always been the most difficult work in shipbuilding industry. Usually, conventional cold working equipment (plate bending machine, hydraulic press, etc.) is used for plate parts with general single curvature or small curvature, while for complex linear outer plates (hyperbolic plates), it still needs to be solved by hot working or water-fire plate bending process, etc. Mechanization and automation technology of plate processing has always been an important research content in hull construction.

② Profile forming and processing: generally, shipyards mostly use hydraulic profile bending machines to bend and form profiles. In recent years, CNC frame control cold bending technology and equipment have been continuously developed and are developing towards artificial intelligence and microcomputer control.

Hull Assembly

Hull assembly is the process of assembling qualified hull parts into sub-assembling, block, mega block and up to the entire hull. Due to the use of mathematical lofting and various numerical control processing technologies and equipment, the processing accuracy of hull components has been continuously improved, which has made the mechanization and automation of the hull assembly operation continuously improved. According to statistics, assembly and welding account for 12–18% of the total labor in ship construction. Therefore, improving the mechanization and

automation of assembly and welding operations can not only reduce labor intensity, but also effectively shorten the cycle and improve quality (Suwasono et al. 2011).

Hull assembly generally includes component assembly (such as various welded T-beams, frame frames, stern posts, and rudders), block assembly (such as flat section, curved section, semi-dimensional unit, three-dimensional unit, and mega block), and berth assembly (blocks assembly, large three-dimensional blocks, and blocks into the entire hull on the berth or in the dock).

Ship Launching

When the ship is built on the berth (or in the dock) to the predetermined amount of work, it needs to rely on special equipment and operation methods to move the ship into the water. This operation is called ship launching. There are many ways to launch a ship, which can be roughly divided into three types:

On the inclined slipway, the ship uses the component force generated by its own weight to overcome the friction between the slipway and the ship, so that the ship slides into the water by itself, which is called gravity launching.

Using the towing slide to cooperate with the launching vehicle (ship lift, etc.) to tow the ship into the water is called towed launching.

Water is drained into the dock to float the ship itself, and then the dock door is opened and the ship is towed out, which is called floating launch.

Ship Test

The ship test includes mooring test, tilt test, and sailing test, which are divided into two test stages.

The mooring test is a test that the ship is moored on the dock after the ship is basically completed and the shipyard obtains the consent of the ship owner and the survey agency. Its task is to test the ship's main engine, auxiliary machinery, and various equipment systems according to the design drawings and test procedures to check the ship's integrity and reliability. Then the ship is placed in a still water area and a tilt test is performed to determine the center of gravity of

the completed ship. These two tests constitute the first stage of the ship test.

The sailing test is a comprehensive assessment of a newly built ship, including two types of light-load test and full-load test. It belongs to the second stage of the ship test and is carried out by the shipyard together with the shipowner and the survey agency.

Before the test, the sea trial should be carried out at sea or in the river according to the type of ship. Before the ship sails, fuel, life-saving, daily necessities, test instruments, meters, and special test equipment should be prepared. During the sea trial, various technical indicators of the main engine, auxiliary engines, various equipment systems, and communication and navigation instruments should be measured, and various navigation performance indicators of the ship should be measured to check whether they meet the design requirements.

Delivery and Acceptance

After the ship test, the shipyard should immediately organize and implement various operations to eliminate the defects found in the test. At the same time, the ship and all equipment on board should be submitted to the shipowner for inspection one by one in accordance with the drawings, instructions, and technical documents, for example, the handover of cabins, the inventory handover of spare parts, the handover of main engines, auxiliary engines, various equipment systems, and communication and navigation equipment.

After the above work is completed, the delivery and acceptance documents can be signed, and the survey agency will issue a certificate of conformity, and the shipowner can arrange for the ship to participate in operation.

Shipbuilding Mode

Connotation of Shipbuilding Mode

Pattern is the standard form or standard style of things and the description of the basic characteristics of things. For complex shipbuilding projects, because different shipyards have different technical levels and production conditions, even if their basic shipbuilding characteristics are the

same, the specific shipbuilding methods adopted are also different. For example, it is also a shipyard that adopts the sectional construction method, but its berth assembly method and sectional manufacturing method will vary according to the conditions of the shipyard. Therefore, the shipbuilding mode does not reflect the specific shipbuilding method (Koenig 2001).

Shipbuilding mode is the general term of shipbuilding system and technology. It expresses the existence form and activity mode of shipbuilding industry as a whole and dynamically. Analyze the past and present of shipbuilding industry and explore its future and the shipbuilding mode have shown an orderly development.

The Orderly Development of Five Shipbuilding Modes

With the advancement of science and technology and the growth of ship demand, the shipbuilding mode is constantly developing and changing, but it is relatively stable and unchanging for a period of time.

The traditional shipbuilding industry is a labor-intensive industry. A large shipyard has tens of thousands of employees and produces tens of thousands of tons of ships per year. The world-class modern shipyard has an annual production capacity of more than one million tons of ships, and the total number of employees is only about 1000. The fundamental reason for such a disparity in production efficiency is the timely introduction of advanced manufacturing technology at that time. Before the 1950s, riveting technology was mainly used to develop the ancient wooden ship construction into modern shipbuilding with steel ship construction as the main body. In the 1960s, welding technology generally replaced riveting technology, transforming the ancient shipbuilding method oriented by "system" into "region" orientation, extending the assembly, outfitting, and painting operations originally concentrated on berths and docks to larger working surfaces of workshops and platforms. Since the 1970s, with the large scale of ships, in the construction of new shipyards and the modernization of old shipyards, the "group technology" has been introduced and studied comprehensively and deeply. Through the

similarity analysis of different types of construction processes, the classification of ship areas, operation types, and construction stages has been realized, and the shipbuilding flow water and virtual flow water production have been organized according to the concept of “intermediate products.” As a result, a large number of mechanized equipment have replaced heavy manual labor, and the original labor-intensive shipbuilding industry has undergone a qualitative change, becoming a modern “equipment-intensive” industry, and the total number of employees has been reduced by orders of magnitude. The “decentralized” specialization mechanism of social technology (organizational technology) in shipbuilding has been actually brought into play, and productivity has been significantly improved. Since the 1980s, the application of electronic computer technology in shipbuilding CAD (computer-aided design) and CAM (computer-aided manufacturing) has continued to expand and deepen, and shipbuilding precision control technology and ship engineering management technology are improving day by day, which has enabled the “integration” of shipbuilding social technology. The mechanism is fully functioning, and it is moving towards the integration of “space separation and orderly time,” namely the development of CIMS (Computer Integrated Manufacture System) of the shipbuilding industry, and then becoming an “information-intensive” industry, that is, an advanced state of modern shipbuilding mode (Podder et al. 2015).

Currently, countries with advanced shipbuilding technology in Europe, America, and Asia are developing shipbuilding technology with two different technical routes. The first is the “simulation technology” route, which uses computer technology, robotics, and artificial intelligence technology on the basis of intensive shipbuilding equipment. The second is the “innovative technology” route, through the concept of digitization, simplifying the design, manufacturing and operation of ships, characterized by precision and agility, and strive to study a new generation of shipbuilding mode.

For half a century, riveting technology, welding technology, group technology, and

information technology have promoted and dominated the development of shipbuilding mode one by one and successively formed the ship’s “integrated manufacturing mode,” “segmented manufacturing mode,” and “separated manufacturing mode,” and “integrated manufacturing mode.” This deductive process is an innovative process in which technology and economy are closely combined. The formation of each mode is due to the introduction of a new leading technology and the establishment of a new production function, which introduces a new combination of production factors and production conditions into the production system. Its development, like the whole manufacturing industry, is based on “technology as the center.” The development of shipbuilding industry is closely related to the improvement of the scientific and technological level of the whole industry. It is essentially a comprehensive effect produced by the organic combination of soft technology research and development and hard technology equipment and facilities investment. The transformation of the global shipbuilding mode fully reflects the convergence of technologies, that is, it does not tend to be the same with regional influence, showing the development law of shipbuilding technology (Tokola et al. 2016).

Development Trend of Shipbuilding

At present, the development trends of shipbuilding technology mainly include: low-consumption and high-efficiency ship production technology, green shipbuilding technology, digital shipbuilding technology, and intelligent shipbuilding technology.

High Production Efficiency of Shipbuilding

High production efficiency shipbuilding: giant block, large-scale unit, outfitting unit assembly line, laser or electric induction heating outer plate forming, curved surface block assembly line, robot group welding of hull structure, and efficient welding technology, etc (Xu 2005).

Low resource consumption shipbuilding: shipyard production processes without intermediate storage yard, shipbuilding activities moving forward and parallel, flange welding before bending pipe processing technology, etc.

Greening of Shipbuilding

Greening of shipbuilding is a shipbuilding technology with the highest utilization rate of production resources and the smallest impact on the environment. Therefore, the shipyard is responsible for environmental protection during shipbuilding; the pollution to the ocean must be reduced during ship operation; after the ship is scrapped, most of the materials can be reused. Therefore, greening of shipbuilding includes two aspects: “greening of ships” and “greening of shipyards.”

“Greening of ships” can be called 3R Ship: environmentally friendly design, harmless materials, efficient technology, and antipollution equipment are adopted in shipbuilding and using ships to reduce the consumption of materials and energy and the pollution to the land and sea environment. When repairing the ship, the spare parts can be conveniently classified and recycled. When the ship is decommissioned, most of the materials can be reused.

Research and development of greening of shipbuilding cases: the automation of hull processing and assembly and welding can be fully realized, and the utilization rate of steel can be improved; single-pass welding should be replaced by multiple-pass welding to reduce environmental pollution caused by welding smoke arc; non-emission steel pretreatment and coating technology should be adopted; shipbuilding sites should be sealed; and the air in the cabin should be filtered.

Digitization of Shipbuilding

The goal of digital shipbuilding: the entire shipbuilding process is accurately defined by the computer system and all shell outfitting operations and system test tasks are specified in the form of electronic process charts, as well as production resources and operating methods to perform the tasks. Then, the integrated process chart is released to all construction departments including subcontractors. For this reason, the database of the manufacturing part of the digital shipbuilding must be stored, and the historical data of the solution, the logical data of the relationship, and

the capability data of all parties must be updated in real time. Digital shipbuilding requires three-dimensional modeling of “ships” and “shipyards” in order to use the shipyard’s 3D environment to make process, operation plans, robot operations, NC (Numerical Control Cutting) simulations, and logistics simulations.

Before the production process, the operations of each cutting seam, assembly seam, and welding seam of the ship must be simulated in the computer to obtain relevant technical data and management data (Stott et al. 2018).

Intelligent Shipbuilding

With the development of shipbuilding technology, a large number of intelligent software and hardware suitable for it will surely appear.

Globalization of Shipbuilding

Global economic integration refers to the adjustment of industrial structure in the world and the flow of production factors in the global scope. Production resources are allocated beyond national borders, so a unified development trend, process, and result are formed. Its main feature is “parallel and seamless integration in different places.” “Parallel in different places” aims to maximize the assembly efficiency (using giant blocks and giant outfitting units) and minimize shipbuilding costs (ship component processing park and transnational ship production block. According to the comprehensive modularization and digitization of manufacturing, the “seamless integration” of the whole process of ship construction in terms of technology and time is realized.

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Inventory of Hazardous Materials

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Definition

An Inventory of Hazardous Materials (IHM) is a document in which all potentially hazardous materials onboard a vessel that can pose a risk to the health and safety of people or to the environment is located, identified, and quantified.

Scientific Fundamentals

IHM is an inventory that details the type, amount, and location of hazardous materials in the constructions, systems, and equipment of a vessel. It is increasingly recognized as a means to enhance onboard safety and environmental awareness throughout the whole life until the ship is being recycled.

An IHM on board primarily concerns a limited number of hazardous materials, such as asbestos, ODS, PCBs, PFOS, and antifouling that need to be assessed. If technically feasible, heavy metals, radioactive components, and some other substances should also be assessed.

The main objectives of IHM are:

1. To gain insight into and have at hand specific information about the presence of hazardous materials on board ships
2. To safeguard the health and safety of workers and crew throughout the ship's life cycle and beyond, thus also during dismantling
3. To prevent environmental pollution in and around shipyards, docks, shipbreaking yards, and ship recycling companies

Background

Ship scrapping industry used to be virtually unregulated and even was one of the worst safety record industries, which caused massive environmental pollution. People had little awareness or acknowledgment of the appalling working conditions and environmental pollution until environmental groups brought widespread awareness of ship scrapping practices (Naruse et al. 2010).

In order to address the issues in shipbreaking, industry working groups developed the Industry Code of Practice on Ship Recycling. This guidelines subsequently fed into discussions at the International Maritime Organization (IMO), which resulted in the IMO Guidelines on Ship Recycling, adopted by member states in December 2003. These voluntary guidelines introduced the concept of a “Green Passport,” which is the predecessor of IHM, for the first time (Lloyd's Register 2014).

In 2009, the Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships (the Hong Kong Convention, or the HKC, for short) was formally adopted at a Diplomatic Conference in Hong Kong. The HKC requires that each ship shall have on board an IHM. The Inventory is detailed in accordance with the actual conditions of each ship and shall specify at least the hazardous materials listed in

Appendices 1 and 2 of the Regulations for Safe and Environmentally Sound Recycling of Ships and the hazardous materials contained in hull structures and ship's equipment. The location and approximate number of them should be noted clearly.

The IHM consists of three parts (van Hooren 2015):

Part I: Materials contained in ship structures or equipment. Part I of the IHM shall be developed for new and existing ships.

Part II and Part III: Operationally generated wastes and stores. These parts do not need to be completed until the ship is being prepared for recycling.

Materials that shall be listed in Part I of the IHM are divided in two categories:

Table A: Comprises materials listed in Appendix 1 of the HKC. All Table A materials, if present on board, shall be included in the IHM.

Table B: Comprises materials listed in Appendix 2 of the HKC. Table B materials, if present onboard, shall be included in the IHM of new ships if above the threshold values. For existing ships Table B materials shall be listed as far as practicable.

The time window between the adoption of the HKC and its entry into force led to the appearance of a parallel convention, the EU Ship Recycling Regulation (the EU Regulation), which entered into force on 30 December 2013. The Regulation is mostly aligned with the Hong Kong Convention but differs in some aspects.

The EU Regulation requires the establishment of a list of approved ship recycling facilities (the "EU List") which meet the design, construction, and operation requirements of the EU but may be anywhere in the world (European Parliament 2013).

The requirements for an EU IHM are expected to be more onerous than for the HKC IHM, especially in terms of the accuracy and comprehensiveness. Guidance is expected to be produced by the European Commission (EC) on implementation.

It should be noted that the EU Regulation, in addition to the Hong Kong Convention requirements, does not permit the presence of the following hazardous materials: perfluorooctane sulfonic acid (PFOS) and brominated flame retardant (HBCDD).

Since the EU Regulation is at present incomplete in its requirements in so far as the permitted threshold values are concerned and is in the process of being amended, for implementation of the requirements of this Regulation, consideration should be given to the guidelines developed by the IMO to support the Hong Kong Convention.

Hazardous Materials

The Hong Kong Convention

The tables below show the category of hazardous materials and the minimum list of items for the IHM that should be listed in the Hong Kong Convention (International Maritime Organization 2009).

The EU Regulation

Hazardous materials in the EU Regulation are mostly covered by the Hong Kong Convention, but there are still some differences.

For ozone-depleting substances, except those in the Hong Kong Convention, HCFC-22 chlorodifluoromethane is prohibited in the EU Regulation. However, hydrochlorofluorocarbons (HCFCs) contained in new installations are permitted until 1 January 2020 in the Hong Kong Convention.

The EU Regulation also involves perfluorooctane sulfonic acid (PFOS) in the category of hazardous materials. Perfluorooctane sulfonic acid (PFOS) means perfluorooctane sulfonic acid and its derivatives. In accordance with Regulation (EC) No 850/2004 of the European Parliament and of the Council, new installations which contain perfluorooctane sulfonic acid (PFOS) and its derivatives shall be prohibited.

As for the list of items for the Inventory of Hazardous Materials, the EU Regulation adds brominated flame retardant (HBCDD) to the Hong Kong Convention.

Key Applications

Development of Part I of the IHM for New Ships

Part I of the IHM for new ships should be developed at the design and construction stages (Yan and Liu 2011).

Checking of Materials Listed in Table 1

During the development of the Inventory (Part I), the presence of materials listed in Table 1 should be checked and confirmed; the quantity and location of Table 1 materials should be listed in Part I of the Inventory. If such materials are used in

compliance with the Convention, they should be listed in Part I of the Inventory. Any spare parts containing materials listed in Table 1 are required to be listed in Part III of the Inventory.

Checking of Materials Listed in Table 2

If materials listed in Table 2 are present in products above the threshold values, the quantity and location of the products and the contents of the materials present in them should be listed in Part I of the Inventory. Any spare parts containing materials listed in Table 2 are required to be listed in Part III of the Inventory.

Inventory of Hazardous Materials, Table 1 Hazardous materials and control measures

Hazardous materials	Definitions	Control measures
Asbestos	Materials containing asbestos	For all ships, new installation of materials which contain asbestos shall be prohibited
Ozone-depleting substances	Ozone-depleting substances means controlled substances defined in paragraph 4 of article 1 of the Montreal Protocol on Substances that Deplete the Ozone Layer, 1987, listed in Annexes A, B, C, or E to the said Protocol in force at the time of application or interpretation of this Annex Ozone-depleting substances that may be found on board ship include but are not limited to: Halon 1211 Bromochlorodifluoromethane Halon 1301 Bromotrifluoromethane Halon 2402 1,2-Dibromo-1,1,2,2-tetrafluoroethane (also known as Halon 114B2) CFC-11 Trichlorofluoromethane CFC-12 Dichlorodifluoromethane CFC-113 1,1,2-Trichloro-1,2,2-trifluoroethane CFC-114 1,2-Dichloro-1,1,2,2-tetrafluoroethane CFC-115 Chloropentafluoroethane	New installations which contain ozone-depleting substances shall be prohibited on all ships, except that new installations containing hydrochlorofluorocarbons (HCFCs) are permitted until 1 January 2020
Polychlorinated biphenyls (PCB)	“Polychlorinated biphenyls” means aromatic compounds formed in such a manner that the hydrogen atoms on the biphenyl molecule (two benzene rings bonded together by a single carbon-carbon bond) may be replaced by up to ten chlorine atoms	For all ships, new installation of materials which contain polychlorinated biphenyls shall be prohibited
Antifouling compounds and systems	Antifouling compounds and systems regulated under Annex I to the International Convention on the Control of Harmful Antifouling Systems on Ships, 2001 (AFS Convention), in force at the time of application or interpretation of this Annex	1. No ship may apply antifouling systems containing organotin compounds as a biocide or any other antifouling system whose application or use is prohibited by the AFS Convention 2. No new ships or new installations on ships shall apply or employ antifouling compounds or systems in a manner inconsistent with the AFS Convention

Inventory of Hazardous Materials, Table 2 Minimum list of items for the IHM

Any hazardous materials listed in Table 1
Cadmium and cadmium compounds
Hexavalent chromium and hexavalent chromium compounds
Lead and lead compounds
Mercury and mercury compounds
Polybrominated biphenyl (PBBs)
Polybrominated diphenyl ethers (PBDEs)
Polychlorinated naphthalenes (more than three chlorine atoms)
Radioactive substances
Certain short-chain chlorinated paraffins (alkanes, C10–C13, chloro)

Process for Checking of Materials

The checking of materials should be based on the Material Declaration furnished by the suppliers in the shipbuilding supply chain (e.g., equipment suppliers, parts suppliers, material suppliers).

To preclude the introduction of non-compliant components into a vessel's structure or equipment after the Inventory for the vessel has been prepared, Material Declarations are also to be obtained for purchases of spare parts that would be included in the vessel's spares at delivery.

If any of the Material Declarations for spare parts contain materials listed in Tables 1 or 2 above the respective threshold values, these spare parts are to be documented in an appendix to Part I of the IHM. When these spare parts are used, Part I of the IHM is to be updated accordingly.

Development of Part I of the IHM for Existing Ships

When compiling the Part I of the IHM for existing ships, reference can be got from the IMO "Guidelines for the Development of the Inventory of Hazardous Materials" or guidelines from other classification societies.

Many of the items referred to or requested can be found in the ship's onboard documentation and plans. Machinery specifications should show details of items such as gaskets, synthetic bearings, heat insulation, oils, plastics, asbestos, and

transformer cooling media. Insulation and accommodation plans should show many of the common materials in the ship. Electrical drawings and specifications should show details of wiring and wiring coverings. The items themselves may well be labeled, for example, lighting ballasts (for PCB content) and HVAC and refrigeration systems (for refrigerant type).

The process of preparing an Inventory of Hazardous Materials is shown in Fig. 1, which is simplified from the IMO guidelines (International Maritime Organization 2011):

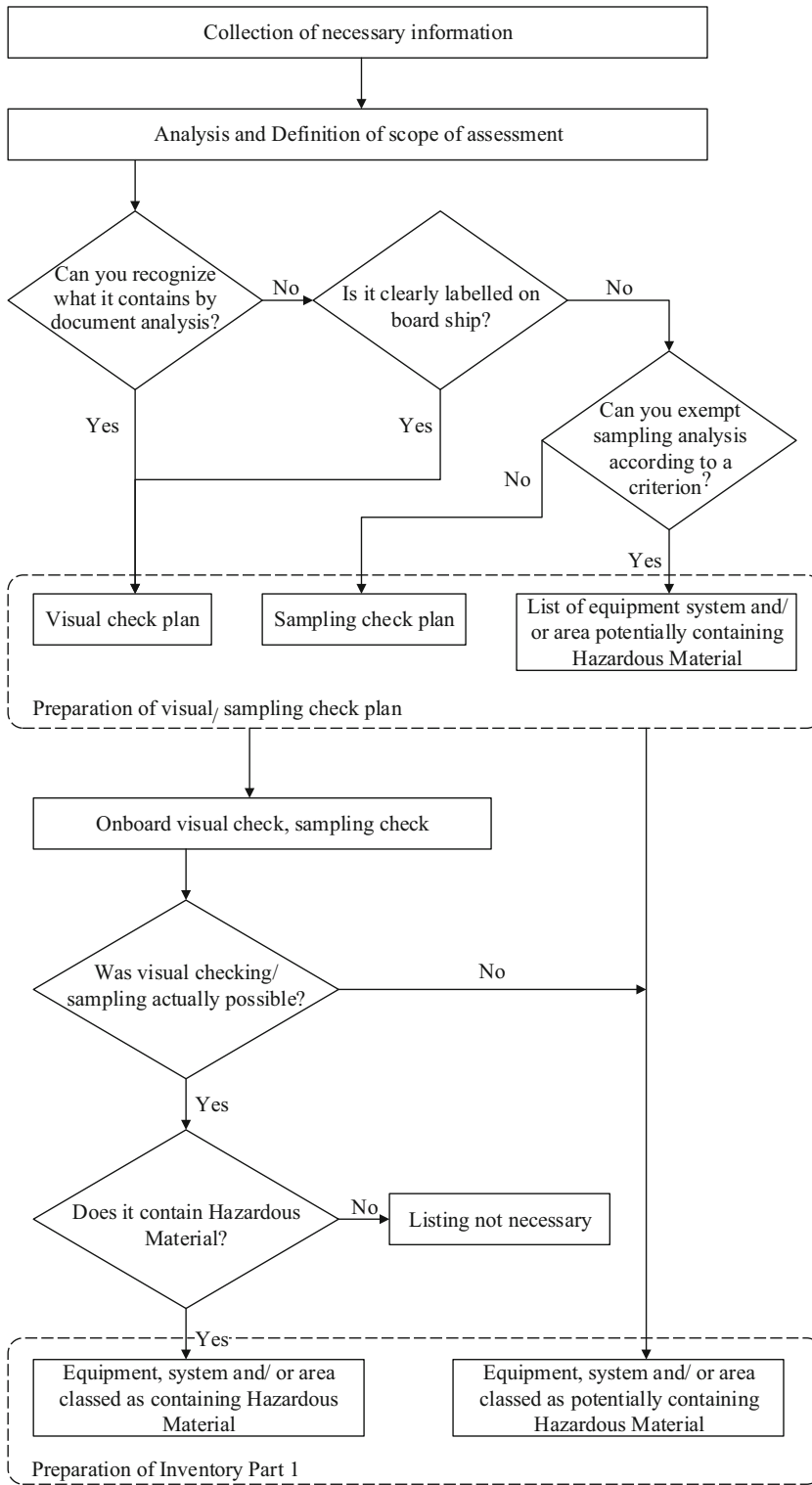
1. Collection of necessary information
2. Assessment of collected information
3. Preparation of visual/sampling check plan
4. Onboard visual check and sampling check
5. Preparation of Part I of the Inventory and related documentation

Collection of Necessary Information

The shipowner should identify research, request, and procure all reasonably available documentation regarding the ship. Information that will be useful includes maintenance, conversion, and repair documents; certificates, manuals, ship's plans, drawings, and technical specifications; product information data sheets (such as Material Declarations); and hazardous material inventories or recycling information from sister ships. Potential sources of information could include previous shipowners, the ship builder, historical societies, classification society records, and ship recycling facilities with experience working with similar ships.

Assessment of Collected Information

The information collected in Step 1 above should be assessed. If one or more materials listed in Table 1 are found to be present in concentrations above the specified threshold value, the products which contain these materials shall not be installed on a ship. However, if the materials are used in a product in accordance with an exemption specified by the Convention (e.g., new installations containing hydrochlorofluorocarbons (HCFCs) before 1 January 2020), the product should be listed in the Inventory. If one or more



Inventory of Hazardous Materials, Fig. 1 Flow diagram for developing Part I of the Inventory

materials listed in Table 2 are found to be present in concentrations above the specified threshold value, the products should be listed in the Inventory. The results of the assessment should be reflected in the visual/sampling check plan.

Preparation of Visual/Sampling Check Plan

While the original IMO Guidelines on Ship Recycling did not require sampling to be carried out, the Hong Kong Convention does require it, for existing ships. This new requirement is in Regulation 5.2, as follows:

Existing ships shall comply as far as practicable with paragraph 1. . . The Hazardous Materials listed in Appendix 1, at least, shall be identified when the Inventory is developed. For existing ships, a plan shall be prepared describing the visual/sampling check by which the Inventory of Hazardous Materials is developed, taking into account the guidelines developed by the Organization.

The requirement for sampling is further amplified in the IMO guidelines:

To specify the materials listed in Appendix 1 of these guidelines, a visual/sampling check plan should be prepared taking into account the collated information and any appropriate expertise.

The visual/sampling check plan is based on the following three lists:

1. List of equipment, system, and/or area for visual check. Any equipment, system, and/or area specified regarding the presence of the materials listed in Appendix 1 by document analysis should be entered in the list of equipment, system, and/or area for visual check.
2. List of equipment, system, and/or area for sampling check. Any equipment, system, and/or area which cannot be specified regarding the presence of the materials listed in Appendix 1 by document or visual analysis should be entered in the list of equipment, system, and/or area as requiring sampling check. A sampling check is the taking of samples to identify the presence or absence of hazardous material contained in the equipment, systems, and/or areas, by suitable and generally accepted methods such as laboratory analysis.

3. List of equipment, system, and/or area classed as “potentially containing hazardous materials.” Any equipment, system, and/or area which cannot be specified regarding the presence of the materials listed in Appendix 1 by document analysis may be entered in the list of equipment, system, and/or area classed as “potentially containing hazardous materials” without the sampling check. The prerequisite for this classification is a comprehensible justification as to the impossibility of conducting sampling without compromising the safety of the ship and its operational efficiency.

Visual/sampling checkpoints should be all points where:

1. The presence of materials to be considered for the Inventory (Part I) as listed in Appendix 1 is likely
2. The documentation is not specific
3. Materials of uncertain composition were used

Onboard Visual/Sampling Check

The onboard visual/sampling check should be carried out in accordance with the visual/sampling check plan. When a sampling check is carried out, samples should be taken, and the sample points should be clearly marked on the ship plan with the sample results referenced. Materials of the same kind may be sampled in a representative manner. Such materials are to be checked to ensure that they are of the same kind. The sampling check should be carried out drawing upon expert assistance.

Any uncertainty regarding the presence of hazardous materials should be clarified by a visual/sampling check. Checkpoints should be documented in the ship’s plan and may be supported by photographs.

If the equipment, system, and/or area of the ship is not accessible for a visual check or sampling check, they should be classified as “potentially containing hazardous materials.” The prerequisite for such classification should be the same prerequisite as in Step 3. Any equipment, system, and/or area classed as “potentially

containing hazardous materials” may be investigated or subject to a sampling check at the request of the shipowner during a later survey (e.g., during repair, refit, or conversion).

For example, if you have a room with many diverse items, all of which have a high likelihood of being asbestos, it may be economically more viable to declare them all as asbestos and treat them accordingly. Alternatively, if the ship has a space with a large amount of concrete in it which is highly homogeneous, has a low likelihood of containing asbestos, and would be very expensive to remove as an asbestos waste, then a practical set of tests to provide the required confidence that the material does not contain asbestos would be sensible.

Most importantly, the approach should be transparent and demonstrate the required duty of care. The first step to achieving this is to assume the worst possible hazard, to take all precautions, and then to design the sampling regime on this basis.

Shipowners can further enhance the Inventory and better discharge their liability by carrying out a limited amount of testing of known materials, in order to satisfy themselves that the available information is accurate and reliable.

Lastly, it is expected that an authorized recycling facility could be capable of checking the principal hazards itself or for the ship to have been pre-cleaned by a yard authorized for that hazard.

Each facility should have its own sampling procedures independent of anything which is in the IHM, since the Inventory is only based on “estimates” and the facility is expected to discharge its responsibility to its own workers in a sensible manner. This is essential when one considers the inaccuracies inherent in any Inventory and the lack of knowledge of what is behind, under, or around anything which may have been sampled.

The knowledge that a facility will “double check” and the fact that an owner can ask the facility to test materials he has been unable to test (due to time, access, cost, or other factors) can also impact on the economic balance of the sampling plan.

Only when the ship is being dismantled can actual, accurate checks of all material be made. Thus, sampling for the Inventory only adds confidence; it can never provide a factual record of all materials on board the ship, and it cannot do the job of checking during dismantling, which must be done by an authorized recycling facility.

Preparation of Part I of the Inventory and Related Documentation

If any equipment, system, and/or area is classed as either “containing hazardous materials” or “potentially containing hazardous materials,” their approximate quantity and location are to be listed in Part I of the Inventory. These two categories are to be indicated separately in the remarks column of IHM. The results of the onboard visual/sampling check are to be recorded on the checklist. Part I of the Inventory is to be developed with reference to the checklist. After the development of IHM, the shipowner is to determine the identification/verification number for Part I of IHM of the ship according to its own management system documents.

The information received for the Inventory, as contained in Tables 1 and 2, ought to be structured and utilized according to the following categorization for Part I of the Inventory: (1) paints and coating systems, (2) equipment and machinery, and (3) structure and hull. The following tables show the standard format (Tables 3, 4, and 5).

“Name of Equipment and Machinery” Column

Equipment and Machinery The name of each item of equipment or machinery should be entered in this column. If more than one hazardous material is present in the equipment or machinery, the row relating to that equipment or machinery should be appropriately divided such that all of the hazardous materials contained in the piece of equipment or machinery are entered. If more than one item of equipment or machinery is situated in one location, both name and quantity of the equipment or machinery should be entered in the column (American Bureau of Shipping 2016).

Inventory of Hazardous Materials, Table 3 Paints and coating systems containing materials

No.	Application of paint	Name of paint	Location	Materials (classification in Table 1)	Approximate quantity		Remarks
1	Anti-drumming compound	Primer, xx Co., xx primer #300	Hull part	Lead	35.00	kg	
2	Antifouling	xx Co., xx coat #100	Underwater part	TBT	120.00	kg	

Inventory of Hazardous Materials, Table 4 Equipment and machinery containing materials

No.	Name of equipment and machinery	Location	Materials (classification in Table 1)	Parts where used	Approximate quantity		Remarks
1	Switch board	Engine control room	Cadmium	Housing coating	0.02	kg	
			Mercury	Heat gauge	<0.01	kg	Less than 0.01 kg
2	Diesel engine, xx Co., xx #200	Engine room	Lead	Starter for blower	0.01	kg	Revised by XXX on 20 October 2008 (revoking No. 2)
3	Diesel generator (x 3)	Engine room	Lead	Ingredient of copper compounds	0.01	kg	
4	Radioactive level gauge	No. 1 cargo tank	Radioactive substances	Gauge	5 (1.8E+11)	Ci (Bq)	Radionuclides: ⁶⁰ Co

Inventory of Hazardous Materials, Table 5 Structure and hull containing materials

No.	Name of structure element	Location	Materials (classification in Table 1)	Parts where used	Approximate quantity		Remarks
1	Wall panel	Accommodation	Asbestos	Insulation	2500.00	kg	
2	Wall insulation	Engine control room	Lead	Perforated plate	0.01	kg	Cover for insulation material
			Asbestos	Insulation	25.00	kg	Under perforated plates

For identical or common items, such as but not limited to bolts, nuts, and valves, there is no need to list each item individually.

Pipes and Cables The names of pipes and of systems, including electric cables, which are

often situated in more than one compartment of a ship, should be described using the name of the system concerned. A reference to the compartments where these systems are located is not necessary as long as the system is clearly identified and properly named.

“Location” Column It is recommended to prepare a location list which covers all compartments of a ship based on the ship’s plans (e.g., general arrangement, engine room arrangement, accommodation, and tank plan) and on other documentation on board, including certificates or spare parts’ lists. The description of the location should be based on a location such as a deck or room to enable easy identification. The name of the location should correspond to the ship’s plans so as to ensure consistency between the Inventory and the ship’s plans. For bulk listings, the locations of the items or materials may be generalized. For example, the location may only include the primary classification such as “throughout the ship.”

The description of location of pipes and electrical systems, including electrical systems and cables situated in more than one compartment of a ship, should be described for each system concerned. If they are situated in a number of compartments, the most practical of the following two options should be used:

1. Listing of all components in the column
2. Description of the location of the system using an expression such as those under “primary classification” and “secondary classification”

“Approximate Quantity” Column The standard unit for approximate quantity of solid hazardous materials should be kg. If the hazardous materials are liquids or gases, the standard unit should be either m³ or kg. An approximate quantity should be rounded up to at least two significant figures. If the hazardous material is less than 10 g, the description of the quantity should read “<0.01 kg.”

Maintenance and Continuity of Part I of the Inventory

To maintain the IHM notation, it is the responsibility of the shipowner to keep the Inventory updated at all times. The shipowner should ensure that the Inventory is maintained with current information.

Whenever there are new installations (machinery, equipment, hull coating) added to the vessel that includes materials listed in

Table 1, as permitted by the Hong Kong Convention, the Inventory is to be updated where applicable. The shipowner has the right to request an additional survey if there are significant amount of changes to the Inventory.

Maintenance of the IHM

There is a maintenance manual which is to include the identity of the designated person; a system for maintaining and updating of the Inventory; records of new installations, repairs, maintenance, and modifications to a ship; and records of changes to the Inventory.

Part I of the Inventory should be appropriately maintained and updated, especially after any repair or conversion or sale of a ship (China Classification Society 2016). The maintenance of Part I of the Inventory is to be based on Material Declarations furnished by suppliers for equipment, parts, material, etc. Material Declarations are to be obtained for all purchases that may impact a ship’s structure and equipment (e.g., materials, machinery or equipment, spare parts, etc.). If any of the Material Declarations contain materials listed in Tables 1 or 2 above the threshold values, Part I of the Inventory is to be updated accordingly with the use of these materials, equipment, or spare parts onboard the ship.

If any machinery or equipment is added, removed, or replaced or the hull coating is renewed, Part I of the Inventory should be updated according to the requirements for new ships as stipulated in Chapter 3. Updating is not required if identical parts or coatings are installed or applied.

Shipowners should implement the following measures in order to ensure the conformity of Part I of the Inventory:

1. To designate a person as responsible for maintaining and updating the Inventory (the designated person may be employed ashore or on board).
2. The designated person should establish and supervise a system to ensure the necessary updating of the Inventory in the event of new installation.

3. To maintain the Inventory including dates of changes or new deleted entries and the signature of the designated person.
4. To provide related documents as required for the survey or sale of the ship.

The deletion of equipment and/or parts of the ship's structure previously classed as "potentially containing hazardous materials" in the Inventory was supported with reasons and evidence that the equipment, system, and/or area does not contain hazardous materials or which has subsequently been sampled and found to contain hazardous material, which has been identified in the revised Inventory or a supplement to the Inventory thereof.

Continuity of the IHM

Where there is a change of flag, owner, or operator of the vessel, the contents of the Inventory, supporting documents, and maintenance manual are to be confirmed as containing the latest information and passed on to the next shipowner.

A new designated person responsible for the Inventory and system to maintain and update the Inventory is to be established by the subsequent owner or operator of the vessel.

Survey of Part I of the Inventory

The Inventory is to be subject to surveys for the issuance and maintenance of the IHM notation.

Initial Survey

The initial survey is to be carried out with an Inventory that has been reviewed without outstanding technical comments. The survey is to verify that the Inventory, especially the location of hazardous materials, is consistent with the arrangements, structure, and equipment of the vessel through an onboard visual inspection.

Annual Survey

The Inventory will be subject to an annual survey in the course of completing other annual and periodical surveys. The annual survey is to verify the following:

1. The Inventory has been maintained and updated to reflect changes in vessel structure and equipment based on the records in the maintenance manual, to the satisfaction of the surveyor.
2. "Materials Declaration" and "Supplier's Declaration of Conformity" have been collected for purchases of materials, machinery or equipment, coating renewal, and spares from the date of the last survey verification of the Inventory or Inventory supplements thereof. Deletion of equipment and/or parts of the ship's structure previously classed as "potentially containing hazardous materials" from the Inventory complies with the requirements.
3. The Inventory, especially the location of hazardous materials, is consistent with the arrangements, structure, and equipment of the vessel through an onboard visual inspection.

Additional Survey

When a ship undergoes a replacement or repair of the structure, equipment, systems, fittings, arrangements, or material, which has a significant impact on the ship's Inventory, the shipowner is to request an additional survey of the Inventory.

The additional survey is to be carried out with an updated Inventory or an Inventory supplement that has been reviewed by engineering without outstanding technical comments. The survey is to verify that the updated Inventory or additional supplements to the Inventory, especially the location of hazardous materials, are consistent with the arrangements, structure, and equipment of the vessel through an onboard visual inspection.

Cross-References

- [Recycling Regulations](#)
- [Ship Recycling](#)

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Ishikawajima-Harima Heavy Industries (IHI)

► [Forming Technology of Steel Hull Plates](#)

J

Jacket Platform

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Synonyms

SeSA_m

Definition

Jacket refers in the oil and gas exploration and production to the steel frame supporting the deck and the topsides in a fixed offshore platform (<https://www.2b1stconsulting.com/jacket/>).

Basic Performance and Characteristics

There are multiple types of offshore platforms depending on the applications and the depth of the water. Fixed platforms are generally used for explorations at shallow water production. Jacket platform, one of the fixed platforms which are often used for production platform, has good adaptability, high safety, and high reliability. It also has the advantage of simple structure and low cost and has become the primary structure form in the exploitation of offshore oil and gas resources for more than half a century.

The steel jacket platform is fixed to the seafloor by means of piling. Since its first use in the 6-m-deep waters of the Gulf of Mexico in 1947, the steel jacket platform has been rapidly developed. By 1978, its depth had reached 312 m. So far, there are about 2,000 large- and medium-sized jacket-type offshore platforms built in the world, and the working water depth has reached nearly 500 m.

Jacket Platform Development History

The world's first fixed offshore platform was built in 1887; it was installed in one oil field in California, USA. But it is a wooden trestle structure. After World War II, many advanced scientific and technological achievements were put into use for marine and offshore development. In 1947, the world's first real steel jacket platform was successfully installed at a depth of 6 m in the Gulf of Mexico, creating a new era of offshore development. Since then, the offshore platform has been rapidly developed, and jacket platform has become the dominating structure of the offshore platforms in the middle shallow sea. At the end of the 1970s, steel jacket platforms were installed in the sea site with more than 300 m. By 1990, a huge jacket platform with a height of 486 m was successfully installed at the site with water depth of more than 400 m in the Gulf of Mexico. Jacket platform has gradually expanded to deeper water area and harsher marine environments over the past years. Jacket platforms are

mainly designed and built for exploration and exploitation of marine resources, mainly for oil and natural gas resources. So far, there are about 2,000 large- and medium-sized jacket-type offshore platforms existing in the world, and the working depth has reached to 500 m.

Features of Jacket Platform Structure

There are many types of steel jacket offshore platforms, and the cost-effect determines the choice of platform type, for example, typically in the deepwater area, using all fully functional, self-contained monolithic platforms. In shallow waters, multiple separate platforms with different functions are mostly welcomed, for example, supply platform, drilling platform, production platform, living platform, auxiliary platform, etc.

The jacket platform has three main structural parts including topside module, jacket, and pile foundation.

Topside Module

The topside module, also called deck structure, includes platform decks, bulkheads, deck pillars, and truss structures, of which primary function is to provide enough space for drilling or production at sea to deploy drilling or recovery equipment, auxiliaries, various kinds of living equipment, and helicopter lift. The topside module includes two kinds of structural forms, namely, the monolithic and block structure.

All deck structures are in principle composed of three-dimensional steel components. The main load-bearing components are divided into three forms, plate girder, box girder, and truss. Plate girders and box girders consist of a pallet. The girders that make up the truss are tubular or profiled (I-beams, channel.). In the case of monolithic structures, the equipment is installed after the construction of the structure; in the block construction, the deck foundation is constructed in prior, and then the module is transferred and fixed on the jacket structure and foundation.

Jacket Structure

Jacket is the supporting structure of the jacket platform. Jacket structure is a steel or steel welded

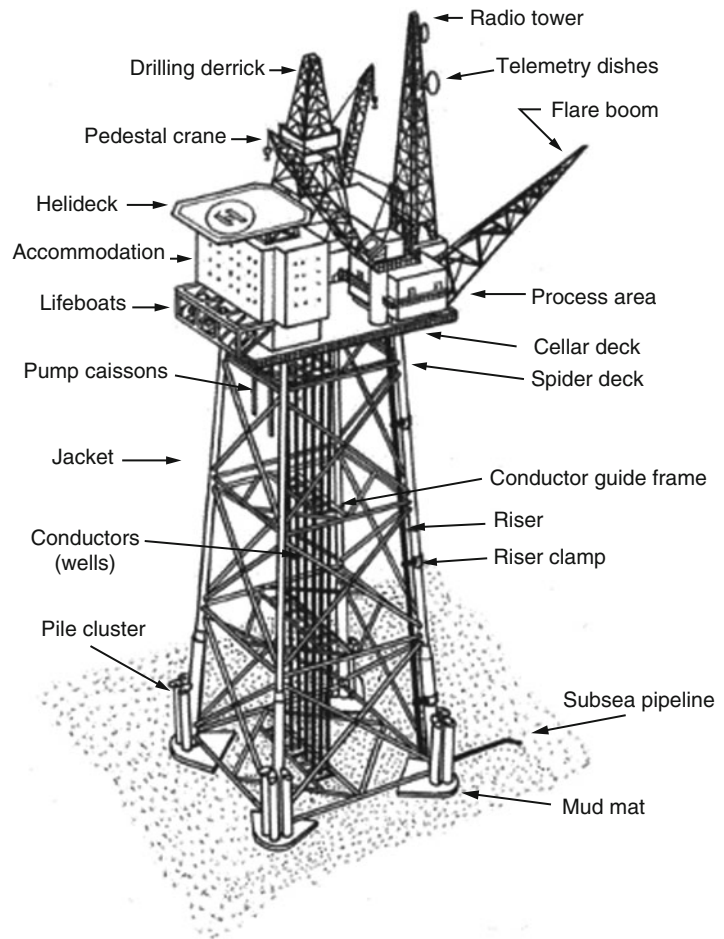
frame and is actually a three-dimensional flat panel rack or plane truss consisting of a three-dimensional space truss structure. These trusses are mainly composed of large-diameter pipe piles (chords), small-diameter pipe, small-sized profiles, and horizontal, vertical, and diagonal strut composition. Tubular node structures are formed at the intersection of the tubular members. General tube nodes are chord (namely, larger diameter tube member in the tube node) wall thickening or other measures to be strengthened. The main role of the jacket structure is to support the topside module and the platform for offshore installation and construction. The jacket's legs are as piling positioning and orientation. In addition, jacket can also be used to connect the ship, in order to facilitate the supply boat off the platform.

Pile Foundation

For most fixed marine structures, the external load and its own weight mostly rely on the pile foundation. The role of the pile foundation is to fix the platform to the seafloor and to withstand the lateral loads and vertical loads and has strong anti-seismic capability. The pile foundation penetrates into the subsoil. Commonly, pile foundation consists of single pile or group pile. The load of the topside module and jacket structure passes through the pile foundation to the seabed (Fig. 1).

Deepwater Jacket Platform

With the development of oil and gas production into deep waters, the deepwater jacket platform has come into being. The main features of deepwater jacket platforms are high, large, heavy, and complex, so weight control is one of the most critical technologies. It is a gravity-type structure which means under constant environmental load, structural self-weight, buoyancy, and ballasting can maintain its own stability. If this equilibrium condition can't be met, thus there will be excessive structural subsidence and horizontal slip or even the whole overturning of the jacket platform. Therefore, compared with shallow water jacket platform, the main challenge of deepwater jacket platform is to control the weight and center

Jacket Platform,**Fig. 1** Components of jacket platform

of gravity of the upper part of the platform. Designers are required to apply the relevant norms and standards reasonably according to the oilfield conditions and strictly control the weight of the upper facilities. As a result, jacket platform applications have water depth limitations. Averagely, deepwater jacket platform is used within the water depth no more than 300 m, but there are already some examples of deepwater jacking jackets installed with a depth more than 400 m in the world (Fig. 2).

Main Challenges in Jacket Platform Application

The main challenges in jacket platform design, installation, operation, and decommissioning

include ductile collapse, fatigue, and ultimate strength.

1. **Ductile collapse.** With the increasing of height of jacket platform, ductile collapse has becoming one of its major failure modes. Because the complexity of large jacket platform, its ductility in different directions is versatile. In the direction with high collapse redundancy, more loads will transfer through this path, and more structural components can effectively bear the load and thus improve the anti-collapse ability of the platform.
2. **Ultimate strength.** Ultimate strength is one critical issue when designing and constructing a jacket platform. Under severe sea states, wave, wind, and current will



Jacket Platform, Fig. 2 A deepwater jacket platform being towed

induce gigantic loads on the jacket platform. The ultimate load-bearing capability needs to be estimated accurately during the design stage. The calculation of ultimate strength often includes the consideration of plastic-elastic characters of the steel material of the jacket platform. In addition, another issue is also often considered, post-collapse strength, which fully considers the material plasticity to assess the strength of the jacket platform after collapse happens.

3. **Foundation failure.** Failures of pile foundation and the ground foundation have become one of the main control factors of the ultimate bearing capacity of the platform. When the platform reaches the limit state, the double plastic hinge is formed on the upper part of the pile leg, resulting in the cause of great reduction of ultimate bearing capacity of the whole platform.
4. **Wave slamming.** The design of the platform should ensure that the deck air gap has

sufficient safety margin. When the wave reaches a certain height, the jacket and the upper block connection unit have to endure the wave slamming effect and can be plastically deformed. When the plastic deformation expands to form a plastic hinge, it will cause the upper part of the platform to roll over and collapse, which is consistent with the observed platform failure mode after typhoon. Therefore, it is necessary to consider the effect of the wave load on the deck in the safety assessment of the existing offshore platform (especially in the case of serious settlement).

Jack-Up (JU)

► [Decommissioning of Offshore Oil and Gas Installations](#)

Jack-Up Platforms

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Synonyms

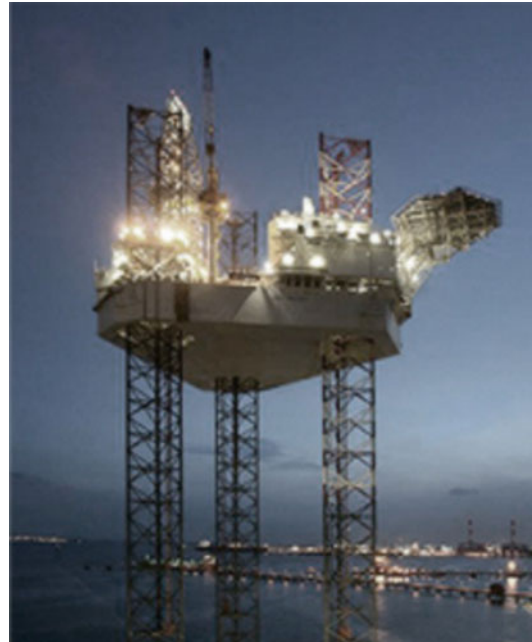
Barges; Mobile offshore drilling units (MODU);
Turbine installation vessel (TIV)

Definition

A *jack-up platform* or a self-elevating unit is a type of mobile platform that consists of a buoyant hull fitted with a number of movable legs, capable of raising its hull over the surface of the sea. The buoyant hull enables transportation of the unit and all attached machinery to a desired location. Once on location the hull is raised to the required elevation above the sea surface supported by the sea bed. The legs of such units may be designed to penetrate the sea bed, may be fitted with enlarged sections or footings, or may be attached to a bottom mat. Generally *jack-up platforms* are not self-propelled and rely on tugs or heavy lift ships for transportation (https://en.wikipedia.org/wiki/Jackup_rig).

Basic Characteristics of Jack-Up Platform

The jack-up platform is a type of removable drilling platform which stands on the seabed through its three or four legs when it is operating. The jack-up platform is a platform for offshore operations consisting of a barge-shaped hull and several legs that can be raised and lowered, as shown in Fig. 1. The legs serve to support the hull and lift or lower the hull. When the jack-up platform needs to reach the working sea area, the legs will lower the hull to the sea surface and allow the hull to float on the sea surface by the hull's buoyancy; thereby the tug can move the platform. When the jack-up platform reaches the target working area,



Jack-Up Platforms, Fig. 1 A jack-up platform

the legs raise the hull to a certain height above sea level, so that drilling, oil extraction, or other work can start.

The installation of jacket-up platform is a critical process. The different phases of jack-up installation are indicated in Fig. 2.

Jack-up platforms can work at water depth from 4 m up to 170 m. If the water depth is too large, the legs of a jack-up platform will be very long, which bring high cost and much technical difficulty.

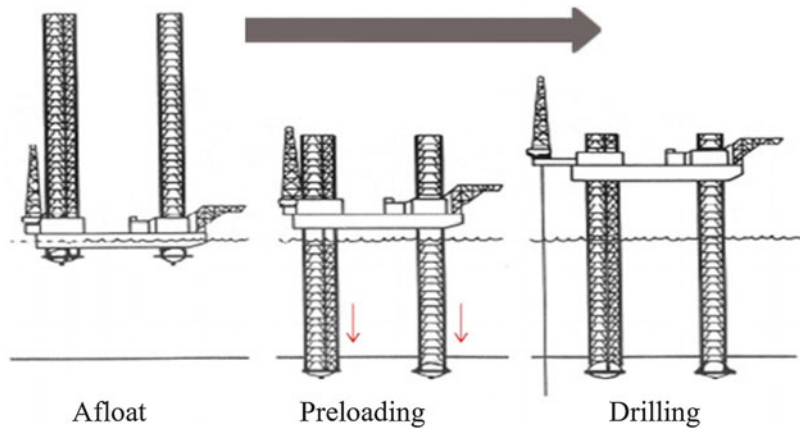
History of Jack-Up Platform

Jacket-up platform has a long history. Figure 3 shows the number of jack-up rigs built between 1965 and 2005. It is indicated in Fig. 3 that in the early of 1980s, there were many jack-up platforms built, about 100 jack-up rigs from 1980 to 1981 and about 120 jack-up rigs from 1982 to 1983, when jacket-up platform reached its top in the history.

As early as the year 1869, American Samuel Lewis first applied for a jack-up rig patent (Chen

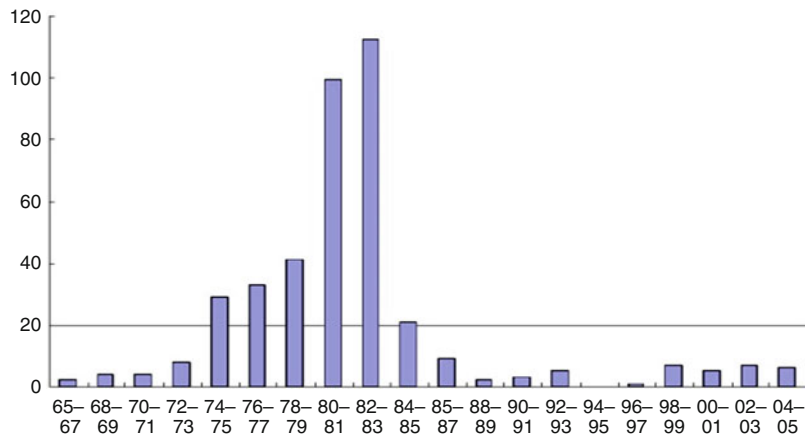
Jack-Up Platforms,

Fig. 2 The different states of a jack-up platform



Jack-Up Platforms,

Fig. 3 Number of jack-up rigs built between 1965 and 2005



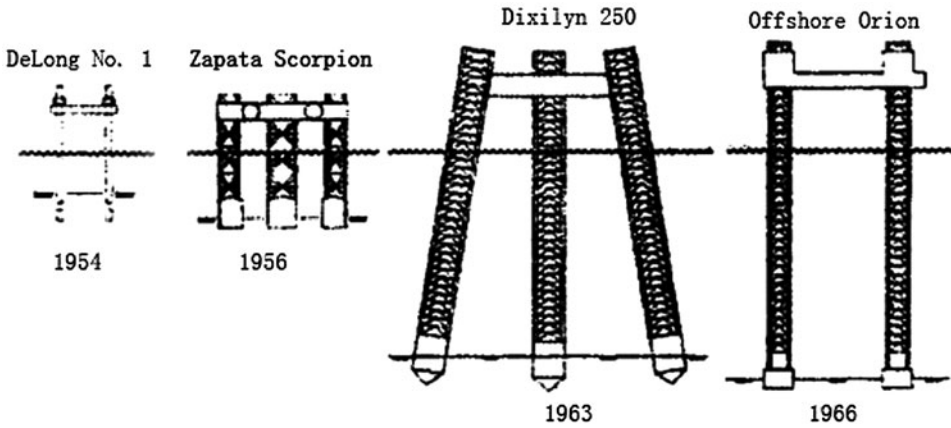
and Li 2007). It was not until 1954 that the world's first jack-up rig DeLong No. 1 (also called Offshore No. 51) (Jackup rig. 2018) appeared. DeLong No. 1 has ten legs, each leg 48.8 m long and 1.8 m in diameter, using the jacking system designed by L. B. DeLong. In 1956, the first three-legged jack-up rig Zapata Scorpion was handed over to Zapata Offshore, which was designed by LeTourneau (Jackup rig. 2018) and uses innovative lattice type legs and rack and pinion jacking system. Scorpion platform, with 56.7 m length, 45.7 m width, 42.7-m-long legs, and 4000 t weight, is the prototype of modern jack-up platform. In 1963, the first oblique-legged platform Dixilyn250 (Drilling Rigs Built in US Shipyards 2015) designed by LeTourneau was completed. In 1966, Offshore Orion (Orion), the first jack-up platform which can work permanently at the North Sea emerged (Fig. 4).

With the advent of better materials and improvements in technology, the water depth in which jack-up platforms can work continues to increase. In 2003, the jack-up platform Bob Palmer of type Super Gorilla XL, owned by LeTourneau Company, got a record depth of 168 m in the Gulf of Mexico. Its height is 273 m.

So far, jack-up platforms account for about 60% of the number of offshore removable drilling platform.

Main Characteristics of Jack-Up Platform

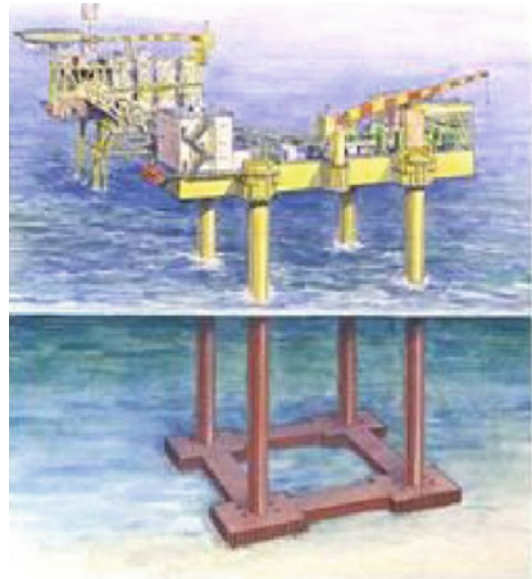
As mentioned earlier, a jack-up platform consists of a hull and several legs. As the hull and legs can be vertically moved relatively, there also exists a jacking system, including the lifting device on the hull, the lifting structure on the legs, and the



Jack-Up Platforms, Fig. 4 Major development milestones for jack-up rigs

power system. Currently, there are two main types of jacking systems: electrohydraulic type and electric rack and pinion type. In the electrohydraulic type, the movement of the piston rod in the hydraulic cylinder is used to lift or lower the legs with the help of the lock pin. This type has the advantage of not requiring complicated transmission mechanism, small size, and flexible control. The disadvantage is that the lifting is not continuous and slow. This type is commonly used in jack-up platforms in shallow waters. In the electric rack and pinion type, the gear fixed on the hull is driven by the motor through the gear mechanism to lift or lower the legs by meshing the rack fixed on the legs. When the legs stand on the seabed, the continuous movement of the hull relative to the legs can be achieved. Relatively speaking, this type has the advantage of higher speed, successive lifting and lowering, and flexible operation. However, a large and complicated transmission mechanism is required. Besides, this type is large in size and requires high manufacturing techniques for gears and racks. This type is often used in the jack-up platform working in the deep sea waters.

When the jack-up platform is in operation, the legs need to be inserted into the seabed. Therefore, the influence of the seabed soil must be considered. If the soil conditions are not good, for example, the soil is very soft, the mat structure or spudcan is often placed in the bottom of the



Jack-Up Platforms, Fig. 5 A jack-up platform with a mat structure

jack-up platform. The purpose is to make the support area larger so that the legs will not sag too deep. Figure 5 shows a schematic diagram of a jack-up platform with a mat structure.

Jack-Up Failure Accidents

Compared to other fixed platforms, jack-up platforms have a higher failure rate. Leijten and

Jack-Up Platforms, Table 1 Total loss for jack-up platforms and fixed platforms from 1970 to 1989

	1970–1979	1980–1989
Jack-ups	162	97
Fixed platform	7	1

Efthymiou (1991) investigated the accident data for both types of platform from the year 1970 to 1989 and gave the findings, see Table 1.

The failure modes of jack-up platforms include the following:

- (a) **Punch-Through:** the punch-through is an accident that the legs suddenly penetrate the soil due to the unpredictable drop in bearing capacity of the soil. This accident may cause damage to the legs or even result in the tilting of the whole platform. This is the most frequently happened accident for jack-up platforms.
- (b) **Rack Phase Difference:** “eccentric spudcan loading can result in a vertical difference in height between the chords within a leg” (Koole 2015). If the Rack Phase Difference (PRD) becomes excessive, it may result in the buckling of the bracing in the legs. PRD is found to more frequently happen in modern jack-up platforms.

Lietaert (2011) summarized a list of the risk associated with the foundation, and Table 2 presents part of them.

The Main Threat of Jack-Up Platform Transport: Stability Issue

As can be seen from Fig. 2, during towing the jack-up platform is buoyed by the hull, and the legs are raised to the very top. In this case, the windward area of the entire jack-up platform becomes larger, and the center of gravity becomes higher, which make the towing stability worse.

According to the requirements of classification society, the towing stability of a jack-up platform includes intact stability and damage stability.

Jack-Up Platforms, Table 2 Jack-up rig foundation associated risks

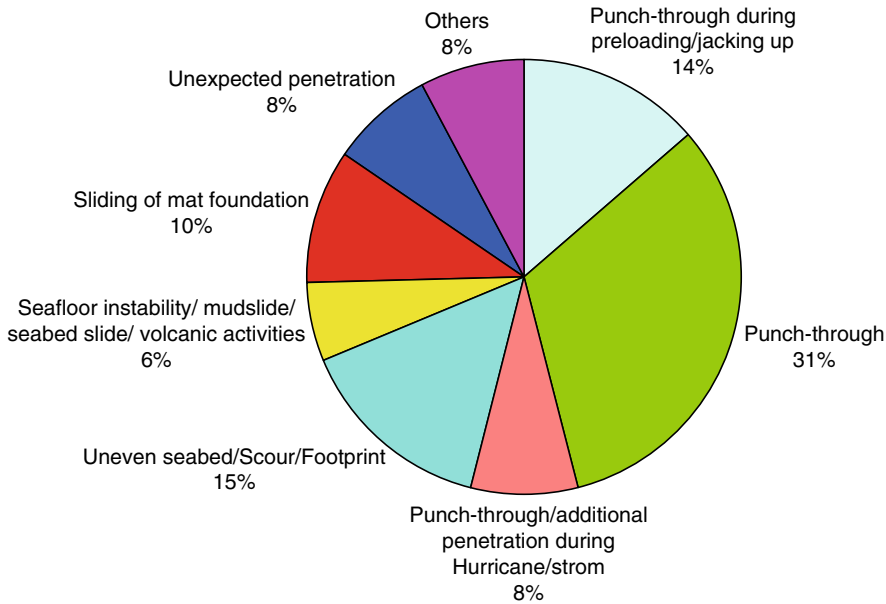
Risk	Methods for evacuation or prevention
Punch-through	(a) Seismic survey (b) Soil sampling and geotechnical testing
Settlement under storm or bearing failure	(a) Seismic survey (b) Soil sampling and geotechnical testing (c) Adequate preloading
Sliding failure	(a) Seismic survey (b) Soil sampling and geotechnical testing (c) Modify the footings
Scour	(a) Bathymetric survey (b) Surface soil sampling and analyze seabed current (c) Inspect footing foundations regularly (d) Install scour protection
Mud slides	(a) Side scan sonar and seismic survey (b) Soil sampling and geotechnical testing

Intact stability means the stability of the hull without damage. Damage stability is the stability of the vessel after the tank has been damaged and flooded. Obviously, the stability of the hull is worse if the tank is flooded. Whether the intact stability and the damage stability meet the requirements is measured by the allowable height of the center of gravity. In other words, if the actual height of the center of gravity in a draft is calculated to be less than the allowable height of the center of gravity during the towing, the stability in this operating condition is deemed to meet the requirements of the classification society.

Punch-Throughs of Jack-Up Platforms

Basic Characteristics of Punch-Through

Punching through is one of the most critical types of incidents for jacket-up platform. The accidents caused by punch-through have a big proportion of the number of jack-up platform accidents, as shown in Fig. 6, about 53%. From Fig. 6, the punch-through accidents happening during the preloading



Jack-Up Platforms, Fig. 6 Different types of accidents in jack-up rigs (Dier et al. 2004)

or jacking up account for 14%, and the accidents during hurricane or storm account for 8%.

The punching through incident happens due to the platform's losing of its capability of standing on the seabed. The seabed soil condition of the jack-up platform working area is complex. When spudcans are located on the seabed in which upper layers of soil are hard and lower layers of soil are soft, the jack-up platform may rapidly sink or tilt due to the loss of support of the soil where a specific leg rests. Punch-through causes the damage to the rig, the deformation of legs, or even platform sinking. So punch-through seriously affects the safety of the platform. Punch-through of a jack-up platform is defined as: "An unexpected jack-up footing rapid settlement resulting in consequential lost drilling time" (Osborne and Paisley 1996).

As a matter of experience, incidents caused by punch-through mainly occur during lifting, preloading, or the stormy weather.

A Punch-Through Accidents

In November 1996, a punch-through incident happened to the Maersk Victory jack-up platform (Maersk Victory); see Figs. 7 and 8.

The accidental process of Maersk Victory is described as below.

The Maersk Victory was towed onto the site by the Massive Tide, once offloaded from the Super Servant 3. Before installation, the position surveyor noticed that there was no site survey for this rig and they attempted to make a site survey on the date of 16 Nov. The first attempt at jacking up started at 6:30 am and aimed to achieve the 2 m preload air gap. It was successfully achieved. At the time of 8:25 am, the helicopter arrived, and then the preloading continued from 10:00 am. However, at 10:32 am, in one third of the way through the preloading, the rig listed to starboard suddenly, and the deck was washed by the sea water. After two or three punches, the rig stabilized. Inspections found that legs #1 and #2 were severely damaged. Then divers cut the hull free of the legs to allow the hull to be towed back to a safe anchorage near Port Adelaide. The legs of the rig were salvaged using the Dock Express 10 over December 1996 and January 1997 and then delivered together with the hull to the Far East Livingston Shipbuilding company in Singapore for repairs.

Jack-Up Platforms,

Fig. 7 The punch-through of Maersk Victory



Jack-Up Platforms,

Fig. 8 The punch-through of Maersk Victory



This was an unsuccessful preloading of the jack-up platform, which caused severe damage to the legs and resulted in lost drilling time. The South Australia Department of Mines and Energy Resources (MESA) undertook an investigation in May 1997. It was concluded that there was a failure to fully evaluate the geotechnical data of the sub-sea sediments.

Jack-Up Foundation Assessment

Jack-up foundation site assessment is important for the jack-up platform to decrease the risk of punch-through. The absence of foundation assessment may cause severe punch-through of

the jack-up platform. The jack-up foundation site assessment usually starts with a review of the site geophysical survey report, which includes bathymetry, side-scan-sonar, sub-bottom profiling, and a shallow gas prognosis (Osborne and Paisley 1996). This geophysical survey report may be supplemented with the geotechnical borehole data.

Borehole Data Requirements

For jack-up foundation assessments, it is important to acquire continuous soil data throughout the borehole depth. The borehole depth is the minimum between the anticipated spudcan

penetration depth plus one spudcan diameter and 30 m.

The borehole investigation could be continuous soil sampling test or in situ piezocone penetrometer testing. As thin layers or subtle variations in soil parameters could significantly change the spudcan bearing capacity and penetration performance during unit installation preloading and subsequent operation, the continuity of data is highly important for jack-up foundations. In highly variable soil conditions, it is necessary to conduct both continuous sampling and in situ test.

Applicability of Soil Laboratory Tests to In Situ Condition

The soil laboratory test is a frequently method to acquire the soil strength profile in a specific situation. It is operated in the laboratory, which can use many instruments. However, there is a debate regarding the applicability of the soil strength derived from small-scale laboratory test compared to the in situ soil strength. First, the zone of the soil during the laboratory test has smaller dimension than that of the soil beneath and around a spudcan. The scale effects have to be considered. The presence of fissures and failure surfaces within the soil, which may not be identified during the soil sample test, could make the scale effects worse. Second, the change in pressure from the in situ location may affect the accuracy of the laboratory test. When the soil is brought to the surface, the gas expansion and soil rupture may diminish the usefulness of the laboratory strength test data.

The laboratory soil test should give due consideration to the scale effects and the change in pressure of the soil mass.

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Jellyfish Bloom

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Synonyms

Continuous Plankton Recorder (CPR)

Definition

Jellyfish bloom was an ecological disaster caused by the increasing number of pelagic cnidarians. It was considered as potential problems only relatively after the 1950s (Purcell et al. 2007). In recent years, reports of jellyfish blooms are rapidly increasing from various information sources, e.g., scientific papers, news from TVs or newspapers, data reports of fishery organization or documentary film, and from disparate regions of the world, e.g., Bering Sea, Sea of Japan, Yellow Sea, and Black Sea (Brodeur et al. 2002; Kideys 2002; Uye 2008; Dong et al. 2010). The jellyfish blooms

are widely considered as serious threats to marine fisheries, tourism, coastal power plants, and the whole ecosystem, especially in coastal and offshore areas. The mechanisms of bloom and collapse of jellyfish population are complicated and vary with the environment (Graham et al. 2001; Purcell 2012). The present section will introduce several aspects of jellyfish bloom.

Introduction

“Jellyfishes” are not a clade, either in taxonomy or in phylogeny. Jellyfish is the common name of various medusae for a long time. This common name included the Ctenophore and Siphonophore based on comparable function in ecosystem and status in trophic webs since the 1990s (Mills 1995). The term is so simple and broad that it causes a lot of confusion and misunderstanding in applications (Lucas and Dawson 2014). Thus, it is essential to distinguish the composition in taxonomy. To be precise, “true” jellyfishes equal to pelagic cnidarians, including Scyphomedusae, Cubomedusae, Siphonophore, and Hydromedusae, but the term is also regarded to cover pelagic ctenophores in practice (Lucas and Dawson 2014).

“Jellyfish bloom” is a widely used but easily confusing concept, not only in public media but also in scientific papers or reports. Consistency of term is essential for describing the phenomenon, understanding the causes and consequences, and preparing appropriate crisis response program. Following the summary and discrimination by Lucas and Dawson (2014), a “true” bloom was able to be considered as a part of seasonal life cycles of organisms, which meant normal population of certain species had a potential instinct to bloom when the environmental conditions were suitable. In contrast, an “apparent” bloom generally indicated a local or sudden increase of population, which was usually due to temporary chemical or physical phenomena or longer-term accumulation in enclosed or semi-enclosed habitats (such as aggregation at fronts or local advection to a new location). Besides, we also need to

make a further distinction between “aggregation” and “outbreak” of population. An “aggregation” of individuals indicates a short-term increase of abundance or biomass in the absence of increased population growth, possibly, due to passive drifting in currents or/and active swimming of individuals. An “outbreak” means a sudden increase in population size, showing an unusually high abundance or biomass compared with other years, and usually caused by anthropogenic change in ecosystems or climate change. Thus, an outbreak of population largely equals to a “unnatural” bloom. Below, we discuss the “unnatural” bloom or outbreak of population, which is the narrow concept of jellyfish bloom.

Causes of Jellyfish Bloom

Eutrophication

Eutrophication is widely considered as one of important factors contributing to jellyfish blooms. Most simply, increased nutrients often lead to greater biomass at all trophic levels, through the bottom-up control (Nixon 1995). More food for jellyfish increases the body growth and reproduction rates of them and the growth rate of population (Moller and Riisgard 2007). Besides, eutrophication shifted the N:P ratios of seawaters. That could alter the pelagic food web from diatom-based to flagellate-based, which might be conducive to jellyfish blooms (Purcell 2007). Eutrophication has another additional effect: algal bloom stimulated by abundant nutrients would reduce the transmittance of seawaters, and decreased light penetration may benefit nonvisual gelatinous predators over visually feeding fish (Aksnes et al. 2009).

Hypoxia

Hypoxia in coastal and offshore waters is one serious consequence, which is largely due to eutrophication and hydrodynamic condition in these regions. Fish die in hypoxic environment or escape from hypoxic waters, but many jellyfishes (though not all) showed stronger tolerance than their competitors in food web (Purcell

et al. 2007). This advantage in competition promotes their population growth.

Substrate Additions

Human activities have greatly changed the coastal sediment environment. Harbors and aquaculture rafts are most important and common artificial structures. They provide an ideal habitat for the sessile stages of scyphozoans and hydromedusae, which are found attached to hard surfaces in harbors and aquaculture rafts (Duarte et al. 2013). In addition, these artificial structures also provide eutrophic water and appropriate lagoons for the reproduction and growth of jellyfish (Lo et al. 2008).

Transportation of Exotic Species

There have been many reports of exotic species introduction for jellyfish, e.g., *Phyllorhiza punctata* and *Mnemiopsis leidyi* (Kideys 2002; Verity et al. 2011; Costello et al. 2012). Once these exotic species are introduced into an environment free of their natural predators, the population of invasive species always show rapid or even explosive growth after they adjust to new environment (Parker et al. 2003). The transportation of species included both human influence and natural process. For example, the most damaging introduced jellyfish, *M. leidyi*, was originally considered to transport within ballast waters of intercontinental shipping vessels (Costello et al. 2012). However, a recent and more detailed study showed that the ocean currents played an important role in secondary spread of this species when they arrived at Europe and lead to invasion corridors in western Eurasia (Jaspers et al. 2018). Besides, global warming makes this trend worse (Stachowicz et al. 2002).

Overfishing

Overfishing is regarded as a key environmental driving factor that could stimulate population bloom of jellyfish, probably because of reduced competition from zooplankton when the abundance of forage fish decreases (Purcell 2012). A nice example is that the abundance of jellyfish significantly increased after the herring stock collapsed in the northern Irish Sea by the long-term

data from Continuous Plankton Recorder (CPR) (Lynam et al. 2011).

Climate Change

Climate changes are also considered to affect the population dynamics of jellyfish. Global warming is an important aspect of climate change and causes serious consequences for ocean ecosystems (Danovaro et al. 2004; Doney 2006). Most of the studies about temperate jellyfish species show greater and more rapid production at higher temperature, and salinity also has significant effects, especially for estuarine species (Purcell 2005). In addition to those direct influences, climate change also showed some more complicated and unpredictable indirect impacts on population dynamics of jellyfish. For example, global and local primary production clearly corresponded with sea surface temperature and dissolved iron in ocean (Gregg et al. 2003; Behrenfeld et al. 2006), and it will influence upper trophic levels through the bottom-up control. Climate change also alters the structure of food web, especially related to the predators of jellyfish, e.g., turtles (Hawkes et al. 2009; Carman et al. 2014). And this will make an impact on population dynamics of jellyfish by top-down control.

Global Trend of Jellyfish Blooms

As mentioned above, public perceptions about jellyfish blooms are increasing globally, based on reports, news, and scientific papers. However, some studies suggested that there was no indubitable evidence for a global increase in jellyfish and found worldwide oscillations with an approximate 20-year periodicity in jellyfish populations. A rising phase during the 1990s largely caused the perception of a global increase in jellyfish abundance (Condon et al. 2012, 2013). A recent study also indicated that complex spatial structure of jellyfish population would significantly affect and disturb the density estimate of them in a bay (Shahrestani et al. 2020). But regardless of the global trend, local jellyfish blooms have still caused serious consequences (see below).

Ecological and Societal Threats of Jellyfish Blooms

Threats to Aquaculture and Fishing

The jellyfish blooms pose serious threats to fishing and natural fish populations, through various pathways, including predation on fish larvae and juveniles, competition on food, and hindrance of fishing activities (Purcell and Sturdevant 2001; Uye 2008; Pereira et al. 2014). On the other hand, a growing number of studies also supported the direct damage of jellyfish to aquaculture species (Baxter et al. 2011; Dong et al. 2017) (Fig. 1).

Threats to Tourism

Several jellyfish species are proved to be harmful to humans, such as the moon jellyfish *Aurelia* spp. (Dong 2019). These threats could be more serious for sensitive people (Burnett et al. 1988). Therefore, if these dangerous jellyfish break out near tourist attractions, it will cause potentially significant losses to local tourism industry.

Threats to Power Plants

Although several early warning and monitoring tools have been developed to avoid above interfere (Kawasumi et al. 2001; French et al. 2018), the reports of interfere from jellyfish blocks to

Jellyfish Bloom,
Fig. 1 Giant jellyfish
(*Cyanea* sp.) washed up on
the coast of Zhangzi Island,
China. (Photo by
Dr. Xinfeng Dai)





Jellyfish Bloom, Fig. 2 The bloom of *Rhopilema hispidum* blocked the cooling water inlet of a nuclear power plant on the southeast coast of China. (Photo by Dr. Xiaoyin Chen)

seawater cooling systems of coastal power plants are still increasing in the past several decades (Daryanabard and Dawson 2008, Richardson et al. 2009) (Figure 2).

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K

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Launching Technology

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Definition

Ship launching is the procedure to move the ship from the construction area to the waters to float, by means of a certain launching equipment, after most of the ship construction project is completed.

Introduction

Ship Launching Methods

Ship launching, according to its principle, can be divided into three categories: gravity launching, floating launching, and mechanizing launching.

Ship launching, according to the direction of the ship's entry into the water, can be divided into side launching and stern launching.

Ship launching, according to its process, can be divided into greased slipway launching, steel roller slipway launching, and trolley launching.

By combining the above methods based on feasibility, ship launching can be realized through many ways (Table 1).

The new ship-launching methods include air bag launching, water cushion launching, etc.

Ship Gravity Launching

Gravity launching is a launching method in which the ship enters the water along the slipway of the building berth under its own gravity. Generally, the direction of entry into the water can be longitudinal (end launching) and horizontal (side launching) (Liu and Wang 2011).

End Launching

End launching (Xu 2005) is suitable for ships of different types and launching weights. According to different slide mediums of the slipway, the inclined building berth slipway can be divided into greased slipway and steel roller slipway, so end launching can be divided into longitudinal greased slipway launching and longitudinal steel roller slipway launching.

Longitudinal Greased Slipway Launching

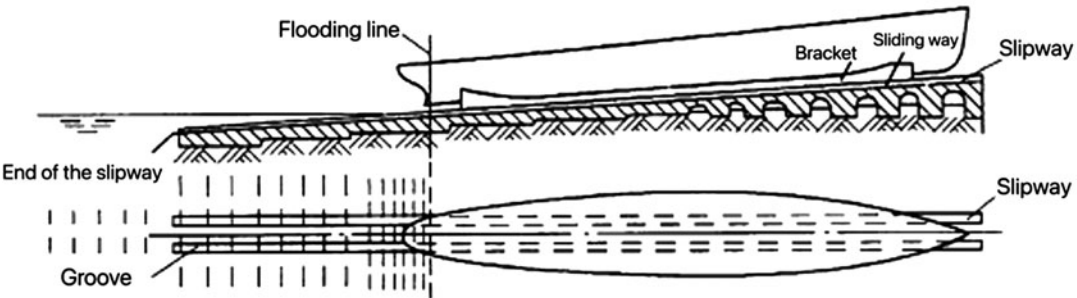
When launching, the keel block, side keel block, and shores are first removed, so that the weight of the ship acts on the sliding way and slipway, and the trigger controls the ship not to slide. When the tide reaches a certain height, the trigger is released and the ship, together with the bracket and sliding way, slides into the water along the slipway. When the stern of the ship reaches the water, the stern of the ship begins to receive buoyancy, and the ship continues to slide until the whole ship floats up by buoyancy (Ji 2005) (Fig. 1).

Longitudinal Steel Roller Slipway Launching

This type of launching uses steel rollers instead of launching grease, by rolling friction instead of sliding friction, thus further reducing the friction between the sliding way and the slipway. Steel roller-launching devices include slipway, steel rollers, distance keeping device, combined type sliding way, etc. (Fig. 2).

Launching Technology, Table 1 Classification of traditional ship launching

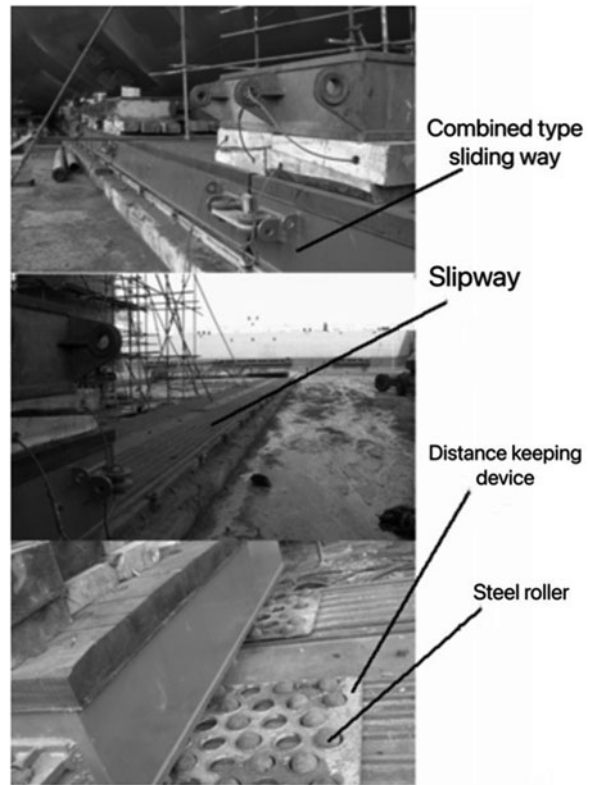
Principle of launching	Water entry method	Launching equipment
Gravity	End launching	Greased slipway, steel roller slipway, etc.
	Side launching	Greased slipway, oak slipway, etc.
Floating	End launching	Shipbuilding dock, floating dock, etc.
Mechanizing launching	End launching	Longitudinal railway slipway, longitudinal inclined cradle slipway, etc.
	Side launching	Side slipway with top and lower railway, sideslipping cradle slipway, etc.
	End launching	Ship lift, crane, etc.



Launching Technology, Fig. 1 Schematic diagram of longitudinal greased slipway

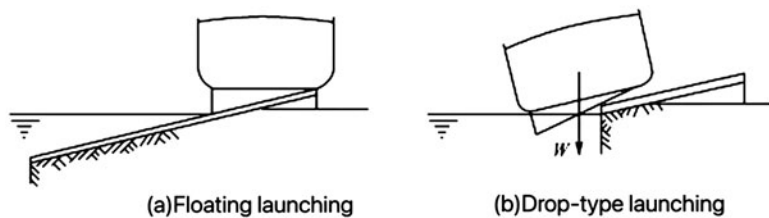
Launching Technology,

Fig. 2 Longitudinal steel roller slipway



Launching Technology,

Fig. 3 Schematic diagram of gravity side launching



Side Launching

Side launching is the sliding of the ship along the width of the ship, which is divided into two types. The first type is the launching with the slipway extending into the water. The ship sits on a wedge-shaped sliding way and slides into the water longitudinally along the slipway. The ship always stays longitudinal during launching, and this type of launching is called side-floating launching. The other type is the launching with the slipway not extending into the water. The end of the slipway is interrupted at the vertical quay wall. When launching, the ship falls into the water, together with the sliding way and poppet,

and balances itself with its own buoyancy and stability, so this type of launching is called side drop-type launching (Xu 2005) (Fig. 3).

Ship Floating Launching

Ship floating launching is a launching method in which water is injected into the ship construction site (dry dock, floating dock, etc.) so that the ship can naturally float on its own buoyancy (Liu and Wang 2011).

Dry Dock Launching

Dry dock (Fig. 4) is a hydraulic structure built on land to facilitate ship launching and jacking onto



Launching Technology, Fig. 4 The dry dock of a shipyard

block based on the principle of floating, which is generally composed of dock chamber (dock bottom and dock wall), dock gate, and water-pumping station.

During ship launching, water is injected into the dock chamber, and the ship floats by water buoyancy. When the water level in the dock is level with the water level outside the dock, the dock gate is opened and the ship is towed out of the dock (Fig. 5).

Floating Dock Launching

The floating dock has a huge concave space with walls on both sides and open ends in the front and rear. It is a trough punt with a special structure. The dock wall and dock bottom on both sides are box-shaped structures, which are divided into several closed cabins along the longitudinal and horizontal directions. Some cabins are called water tanks, which are used for water injection and drainage to make the dock sink and float.

Usage of the floating dock: First fill in the floating dock water tanks with water, and sink

the floating dock to the depth with which the dock could meet the draft requirements of the docked ship. Then use the winch on the top of the dock wall to pull the ship into the dock and remove the water in the water tanks. The floating dock gradually floats up until the top surface of the dock floor is exposed to the surface of the water, and along with the dock floor, the ship to be repaired is also exposed above the water surface, so that the ship repair work can begin (M et al. 2014) (Fig. 6).

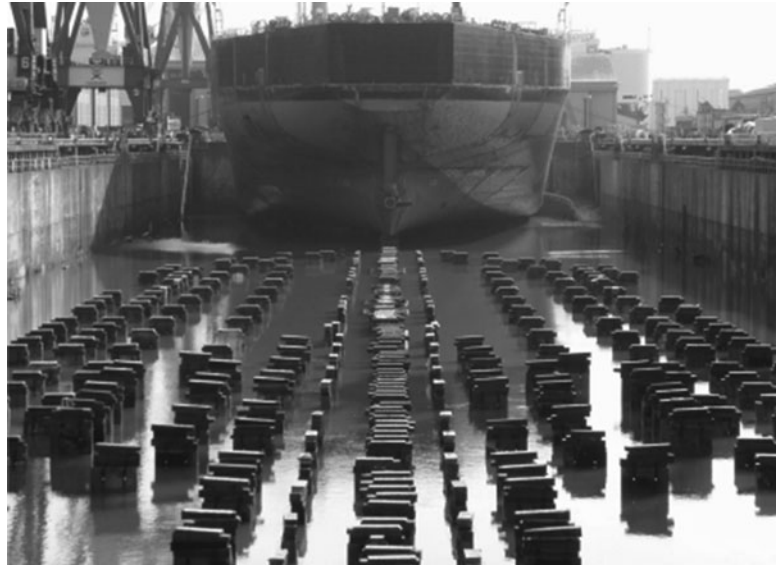
Ship Mechanizing Launching

Mechanizing launching is the use of mechanized facilities to complete the process of ship launching.

Mechanizing Launching with Longitudinal Railway Slipway

The ship is on the integral or sectioned ship slipway with rollers. When launching, the ship slipway that carries the ship is pulled through steel cables and pulleys by the winding engine, and the ship is sent

Launching Technology,
Fig. 5 Dry dock launching



Launching Technology, Fig. 6 Floating dock

into the water along the track on the inclined building berth until it floats successfully (Fig. 7).

Mechanizing Launching with Wedge Type Launching Cradle Slipway

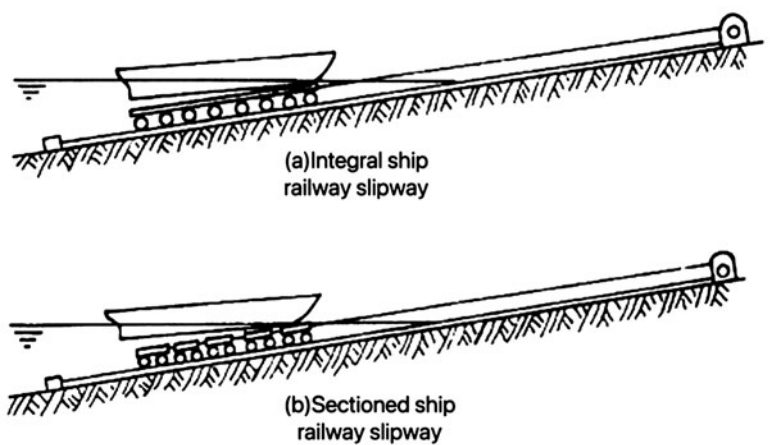
The slipway launching cradle used is a wedge-shaped cradle with a horizontal or approximately horizontal poppet surface. Therefore, bow

pressure will not be generated in ship launching with this method. The launching method is simple and reliable, suitable for the launching and jacking onto block of larger ships (Figs. 8 and 9).

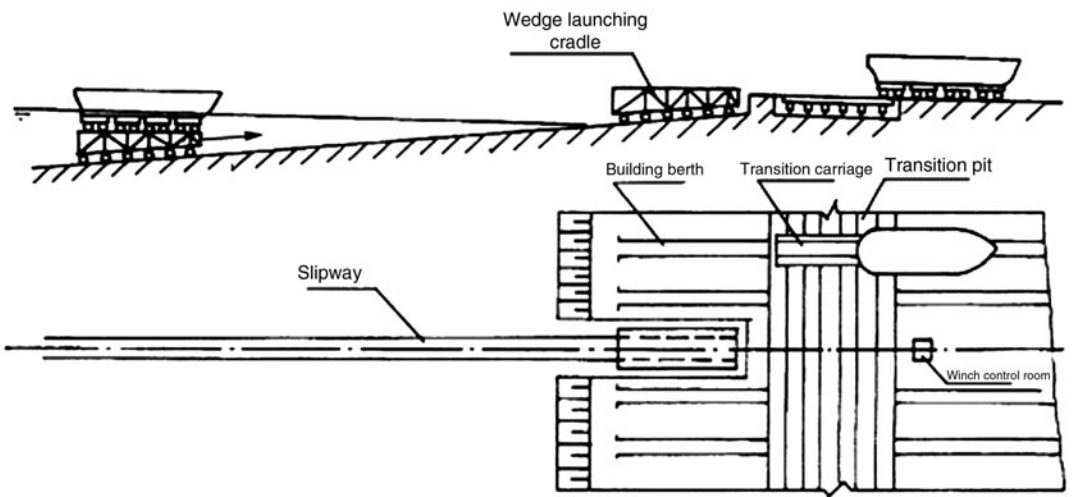
Ship Lift Launching

Ship lift is a lifting platform set up close to the launching wall, where the hydraulic or windlass

Launching Technology,
Fig. 7 Longitudinal
railway slipway



Launching Technology, Fig. 8 Longitudinal slipway of wedge launching



Launching Technology, Fig. 9 Horizontal slipway of wedge launching cradle

Launching Technology,
Fig. 10 Ship lift



winch can be used to vertically lift the platform for ship launching or jacking onto block. In ship launching, the lifting platform is first aligned with the berth slipway and fixed by means of the winch, and the ship is moved, from the building berth to the lifting platform, and is released from the fixation. Then the winch is started, and the lifting platform, together with the launching ship, is lowered into the water. The ship has a draft gradually and floats, and the launching is completed (Ma and Xuexing 2019) (Fig. 10).

Choice of Ship Launching Methods

The ship launching facility is closely related to the layout of the building berth, both of which are extremely important production facilities in the shipyard. When ship launching, the layout and technological process of the shipyard will be comprehensively considered to determine a reasonable launching method and launching facilities.

There are many factors that could affect the choice of building berth types, launching methods and facilities, with the major ones as follows:

1. Plant area and topographic characteristics
2. The coastline of the plant and its geological conditions
3. The water area of the plant and its hydrological characteristics
4. The pattern of water level change
5. Shipyard production program
6. General plan of shipbuilding process

Different launching methods have their own advantages, disadvantages, and scopes of application. When making the choice, we should consider the actual conditions, of the shipyard, and conduct serious analysis and demonstrations to choose a suitable launching method and facilities (Liu and Wang 2011).

Analysis of Ship End Launching Process

In order to prevent possible accidents, launching calculations will be carried out before ship launching, which is an important basis for building berth design, and for corresponding process measures taken during ship launching.

According to the motion state and force condition of the ship's end launching, the end launching process is usually divided into the following four stages:

1. The ship starts to slide until it contacts the water surface (shore slip).
 At this stage, the ship is only affected by gravity, the supporting force from slipway, and friction.

2. From the contact with the water surface to the start of lift by stern (launching).

The hull begins to be affected by water, with its motion direction parallel to the slipway, but it does not move in a straight line with uniform acceleration. When the moment of buoyancy to the bow pivot is equal to the moment of gravity to the bow pivot, the stern starts to float.

At this stage, it is necessary to prevent the tipping of the hull, in which the moment of the functional load to the end of the slipway is greater than the moment of buoyancy to the end of the slipway.

3. From the start of lift by stern to complete floating (lift by stern).

After the ship is lifted by stern, it continues to slide down. Since the moment of buoyancy to the bow pivot is greater than that of gravity, the hull starts to rotate longitudinally around the bow pivot while sliding. When the buoyancy is equal to the gravity of the hull, all floated point of the hull sliding is reached and the whole ship floats.

At this stage, attention should be paid to preventing the dropping phenomena, in which the fore pivot of the hull has passed the end of the slipway before the whole ship starts to float, causing the hull to drop and the bow to collide with the end of the slipway.

4. From complete floating to the ship stopping sliding (all floated).

After the whole ship starts to float, the ship continues to move forward, for about two to three times the length of the ship. Sometimes, due to the narrow waters of the shipyard, the stopping distance requirements of the ship cannot be met, so if the ship is allowed to slide freely, the launching ship may collide with other anchors and facilities in the opposite shore or in the waters.

Cross-References

- [Introduction to Shipbuilding \(Shipyard\)](#)
- [Technical design](#)
- [Underwater Lander](#)

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Layered Space-Time Codes (LSTC)

- [Underwater Acoustic Communication](#)

LDPPs (Long Large Diameter Pipe Piles)

- [Offshore Pile Driving](#)

Level Ice

- [Sloping Structure–Level Ice Interactions](#)

LFC (Low-Frequency Cycle)

- [Semi-submersible Platform](#)

LIDAR (Light Detection and Ranging)

- [Environmental Perception for Underwater Vehicles](#)

Lifting System

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Introduction

In the deep-sea mining system, the lifting system is responsible for transporting the ore collected by the mining robot to the surface mining ship (Shaw 1993). Already the early deep-sea mining research projects and experiences appear that the vertical riser pipe production methodology is the preferred technology for commercial deep-sea mining. Compared with the pneumatic lifting, the hydraulic lifting working process becomes less complicated with large lifting capacity and 50% or more high efficiency. It is considered as the most promising mineral lifting method.

Overview

Already the early deep-sea mining research projects and experience appear that the vertical riser pipe production methodology is the preferred technology for commercial deep-sea mining. Among the mining production methods (such as hydraulic, pneumatic, light medium, heavy medium, and bucket lifting), the riser pneumatic and the hydraulic lifting are the most commonly used ones (Hu et al. 2020). The pneumatic lifting method uses airlift pumps to inject high-pressure air into an underwater mixing chamber at a certain water depth of the riser pipe through an air tubing, the mineral slurry is lifted to the sea surface through the compressed air mixes which is less dense compared to the corresponding column and therefore is displaced upwards through the riser pipe (Doyle and Halkyard 2007). The lifting

efficiency associated with this method is typically less than 20%, which may result in expensive energy consumption. The hydraulic lifting uses one of the following lifting technologies, the clear water pump hydraulic lift or the slurry pump hydraulic lift. The clear water pump hydraulic lift uses a clear water pump installed in a seabed operation platform, the mineral particles are directly fed to the outlet end of the pump, and then lifted to the water surface through the riser pipe. Because the clear water pump and the mineral feeder both are operating in high-pressure conditions, the challenges associated with this method are the design difficulty, low efficiency, and the repair of a malfunctioned system. The slurry pump hydraulic lift conveys the mineral directly to the sea surface through a slurry pump connected in series in the riser pipe. Its working process becomes less complicated with large lifting capacity and 50% or more high efficiency. It is considered as the most promising mineral lifting method.

System Integration and Functions

System Components

A mining lifting system typically consists of a flexible pipe/jumper, a flexible pipe-lifting pump, a rigid riser pipe, and multi-stage electric lifting pumps (Qiu-Hua et al. 2019). The rigid riser pipe is connected to the mining vehicle/collector using a flexible jumper hose, which is suitable for the collector to accommodate the seafloor topography. Buoyancy materials are applied to the flexible jumper to maintain its design spatial configuration. The rigid riser pipe bottom is connected to the flexible jumper through a buffer module, which could effectively ensure the relative independence of the riser pipe system and the mining vehicle (Qiu et al. 2018).

System Functions

The integrated mining lifting system, as the core of the complete mining system, can be mainly responsible for the safety and reliability of the entire system operations. Based on the operational requirements and the features of the mining

system, the main functions of the lifting system are summarized as follows (Liu et al. 2014):

- It provides the important passage and linkage between the seabed mining system and the surface support system.
- It supports the attached power cable and controls the umbilical.
- The flexible pipe is configured to ensure the mining vehicle has enough free-moving space and effectively isolates the disturbances from mining vessels and buffer modules to the mining vehicle operation.
- The buffer module is implemented as a free-hinging container above the seabed to ensure a clogging-free continuously dosing of the riser system.

Classification of Mineral Vertical Transport Systems

Based on the lifting mechanism, mineral vertical transport systems can be divided into hydraulic lifting systems and airlift systems.

Hydraulic Lifting System

Hydraulic Lifting Principle

The mineral hydraulic lifting is conducted in the following steps: (1) the minerals collected by the mining vehicle are pumped into the buffer module through the flexible pipe; (2) the feeder in the buffer module uniformly transports the minerals to the rigid riser pipe; (3) the pressure difference inside and outside the riser pipe provided by the electric pump forms an up-flow in the riser pipe and transports the mineral particles to the surface. The electric pump needs to be designed to overcome the friction caused by the flow of the slurry, and also overcome the potential energy resulted from the density difference between the mineral slurry and the seawater. The pressure inside the riser pipe below the pump elevation is lower than the hydrostatic seawater pressure outside the riser pipe, therefore the minerals are lifted by the net pressure head of the seawater in this riser section. The pressure in the riser pipe above the pump

elevation is higher than the pressure outside the riser pipe, and thus, the minerals are transported by the pump pressure.

Main Components of the System

The key equipment components of the hydraulic lifting system include mainly the lifting electric pump, the rigid riser pipe, the flexible pipe, the flexible pipe lifting pump, the buoyancy materials, and so on.

Rigid Riser Pipe

- Material selection for the rigid riser pipe.

The materials for the rigid riser pipe should be high in strength with corrosion resistant, friction resistant, fatigue resistant, and bending resistant. Generally, the high-yield strength steel is used for the riser pipe, and other materials such as titanium alloy can be also used.

- Diameter of rigid riser pipe.

The outer diameter is mainly designed based on the following factors: the riser self-weight and the dead weight of the lower suspension equipment; the bending stress and deformation from the ship's motion or from the sea current loading. In the preliminary design phase, the riser wall thickness is determined for the static load tensile stress, and in consideration of the dynamic load amplification coefficient for heave compensation and the bending stress coefficient. From experience, in the case of moderate or rough sea state (4 or 5 WMO Sea State Code), the dynamic load amplification coefficient could be around 1.1 for heave compensation and 1.4 if there is no heave compensation, and the bending stress coefficient could be 1.2.

- Riser joint connections.

The rigid riser pipe is made up of pipe bodies and pipe joints. Currently, the oil riser casing is used for deep-sea mining pilot tests. The riser pipe joints are special threaded joints, the upper end of the riser pipe is supported by the ship using a ball joint/flex element, and the lower end is connected to the buffer module using a cross-type swing connector.

Flexible Pipe

- Function requirements.

The main requirements for the flexible pipe are: (1) The flexible pipe should have good bending flexibility/strength and radial rigidity to maintain the stability of the shape of the pipe during mining operations; (2) the flexible pipe configuration should have good flexibility to meet the mobility requirements of the mining vehicle/collector.

- Lifting method.

The flexible pipe system for subsea mining operation uses either a pressure feed type or a suction type of method. For the pressure feeding type, a pump is installed on the mining collector; during operation the pressure inside the pipe is higher than the static seawater pressure outside the pipe, and the pipe is subjected to radial pressure within its stress limit. For the suction type, a pump is installed in the buffer module. During operation, the pressure inside the pipe is lower than the seawater hydrostatic pressure outside the pipe, the wall of flexible pipe is under external pressure. The flexible pipe must be designed to resist the external pressure, which demands the flexible pipe for higher strength and better flexibility. In terms of mineral transport, the fundamental difference between the suction and the pressure type is the different requirements for the flexible pipe. From a perspective point of view, it is recommended to use the pressure feeding method. However, considering the overall layout of the mining system, use of the pressure feeding method, results the pump the installed of on the collector which could inevitably bring much difficulty to the design and operation of the collector and detrimental to the overall layout. In the suction type, although the production cost of the flexible pipe is higher and the manufacturing process is more complicated, the pump is installed on the buffer module instead of the collector which could greatly reduce the design difficulty and the manufacturing cost of the collector, and the installation of the pump on the buffer module can be also used as a counterweight.

- Spatial configuration.

The spatial configuration of the flexible pipe changes relatively to the position between the collector and the buffer module. To maintain a certain spatial configuration, the buoyant materials should be properly placed on the flexible pipe.

- Length and diameter.

To determine the length of the flexible pipe, the free movement range of the seabed collector relative to the buffer module should be considered, and to meet the heave compensation height difference requirement. The longer the flexible pipe, the larger the allowed range of the collector movement, but it also creates greater resistance to the collector, and is more difficult to maintain the flexible pipe spatial configuration.

- Flexible pipe material.

The flexible pipe should have sufficient tensile strength, and internal/external pressure resistance to satisfy the minimum bending radius requirement, having good twisting characteristics, and stay neutrally buoyant in seawater.

Buoyancy Materials The function of the buoyancy materials is to maintain the spatial configuration of the flexible pipe and to reduce the system weight in water. Theoretically, the buoyancy materials must meet the following functional requirements:

- Capable of withstanding the required external hydrostatic water pressure (for every 100 m increase in water depth, the hydrostatic pressure increases by 1 MPa), and will not have damage within the specified working water depth.
- The material should not deform beyond limited allowance during operations.
- The buoyant body can withstand certain impacts and collisions during deployment and retrieving.
- Maintain low water permeability and high bulk modulus of elasticity to ensure its stable buoyancy at high water pressures.

Airlift System

Principle of Airlift

For air lifting, compressed air is injected into the underwater riser pipe, the mineral slurry is lifted upward through the compressed air mixes which is less dense compared to the corresponding column (Pougatch and Salcudean 2008). Above the injection port of the riser pipe is a three-phase flow of mineral, air, and water mixture, below the injection port is a two-phase flow of mineral and water mixture.

Main Equipment of System

The main equipment of the airlift system includes compressors, the airlift tubing/pipe and various valves, and measurement test instruments. The compressor provides the source of power, and the airlift tubing/pipe transports the pressured air.

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Lightweight Construction

- [New Technologies in Auxiliary Propulsions](#)

LIMPET – Land Installed Marine Power Energy Transmitter

- [Power Take-Off System](#)

Liquefied Natural Gas (LNG)

- [Alternative Fuel for Ship Propulsion](#)

Liquefied Petroleum Gas (LPG)

- [Alternative Fuel for Ship Propulsion](#)

LNG (Liquid Natural Gas)

- [FLNG](#)

LNGC (LNG Carrier)

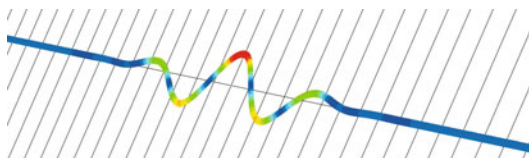
- [FLNG](#)

Local/Global Buckling and Propagation

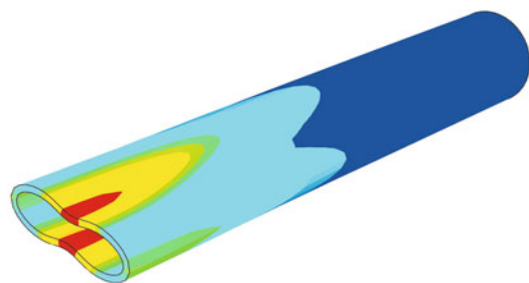
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Definition

Global buckling of a submarine pipeline, similar to Euler's buckling, refers to a sudden large displacement occurs in the sideways direction, when



Local/Global Buckling and Propagation, Fig. 1 Global buckling



Local/Global Buckling and Propagation, Fig. 2 Local buckling

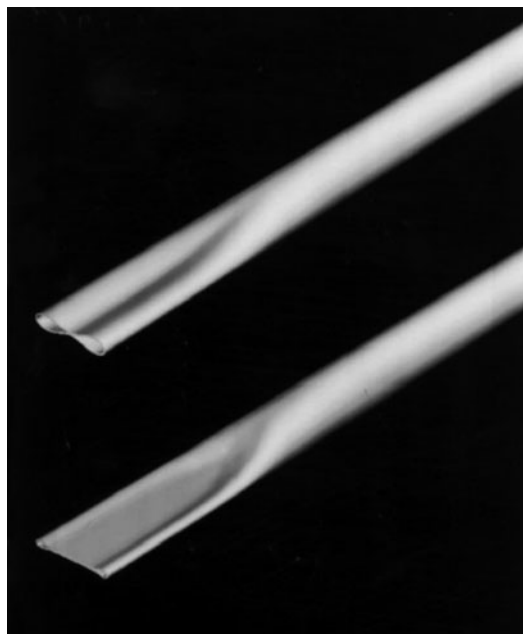
it is subjected to large compressive axial force caused by high temperature and high internal pressure, as shown in Fig. 1. Global buckling damages the structural integrity of a pipeline and results in excessive bending which can lead to more catastrophic failure modes, for example, local buckling, fracture, fatigue, or even buckling propagation (Zhang and Duan 2015; DNVGL-RP-F110 2018).

Local buckling on the other hand is a gross deformation of the pipe cross section due to too high external pressure, as shown in Fig. 2. If the external pressure is larger enough, the local buckling once initiated can quickly transform the pipe cross section into a dumb-bell (or dog-bone) shape that propagates along the pipe driven by external pressure alone in a very short time (Albermani et al. 2011), which is referred to as buckling propagation, as shown in Fig. 3.

Scientific Fundamentals

Cause

Submarine pipelines are often operated with a high temperature fluid to avoid wax or hydrate



Local/Global Buckling and Propagation, Fig. 3 Buckling propagation (Netto and Kyriakides 2000)

problems. Moreover, submarine pipelines have to serve under high internal pressure condition to meet the flow assurance requirement. Both high temperature and high internal pressure result in expansion. The expansion is commonly restrained by friction between pipeline and the seabed. Compressive axial force thereby is induced in the submarine pipeline. Once the compressive axial force exceeds a certain level (critical force), global buckling occurs (Hobbs 1981).

Submarine pipelines are also often operated with high external pressure. Large compressive force is produced in the radial direction of the pipeline because of high external pressure. Under the action of large compressive force, a sudden gross deformation occurs in the section of the submarine pipeline if the section contains initial ovality, local dent, or wall thickness unevenness. For thin wall pipeline (with radius-to-thickness ratio $D/T > 30$), this gross deformation occurs in elastic range of material which is referred to as local buckling. For thick wall pipeline ($12.5 < D/t < 30$), this gross deformation occurs after material yields which is also referred to as collapse (Dyau and Kyriakides 1993).

Essential Feature

Global buckling of submarine pipelines can be classified into two types: (1) vertical buckling and (2) lateral buckling. Vertical buckling is also called upheaval buckling which refers to the global buckling occurs in vertical direction. In the same way, the lateral buckling refers to the global occurs in lateral direction (in the horizontal plane and perpendicular to the axial direction of the pipeline). Vertical buckling occurs in buried submarine pipelines and the lateral buckling occurs in unburied submarine pipelines.

The driving force for global buckling of a pipeline is the effective axial force which represents the combined action of pipe wall tension force, expansion force cause by the pressure between internal pressure and external pressure, and expansion force caused by temperature difference. Effective axial force can be calculated as follows:

$$S = S_T - EA\alpha\Delta T - (p_i A_i) \cdot (1 - 2\nu) \quad (1)$$

where S denotes the effective axial force; S_T denotes the pipeline residual lay tension; E , A , α , and ν are elasticity modulus, cross-sectional area of pipeline, thermal expansion coefficient, and Poisson's ratio, respectively; p_i and A_i denote the internal pressure and the internal bore area of the pipeline; and ΔT denotes the difference in temperature relative to the ambient temperature.

Effective axial force is a key parameter that governs the buckling response of submarine pipelines. Take the lateral buckling for example. Figure 4 shows the effective axial force prior to buckling. For simplicity, it is assumed a pipeline

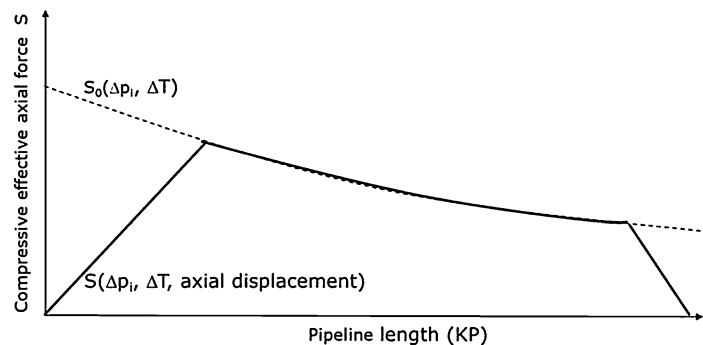
with free end expansion, which is a typical assumption but not always true because of end constraints by PLETs/PLEMs. When the pipeline is started up and the temperature and pressure increase, the effective force will turn into compression. If the pipeline is totally restrained, i.e., not allowed to move axially, vertically, or laterally, the effective axial force will reach its maximum compressive force of S_0 (red dotted line). This force will vary along the pipeline as the temperature and pressure show decreasing profiles along the pipeline. Normally the pipeline ends are free to expand axially, and the effective axial force will therefore end up at zero at the ends (see the black, solid curve in Fig. 4).

Figure 5 shows the general behavior during global buckling. It is split in two, the left part shows the effective axial force, S , over a shorter section of the pipeline. This figure is before the pipeline is started up, so the effective axial force is zero (assuming zero residual lay tension as a simplification). The right part of Fig. 5 shows the typical development of the effective axial force overtime at one location in this pipeline section. For this case, the location is at a significant imperfection where a lateral buckle will be triggered when the effective axial force reach its critical value, S_{init} . Note that this critical value, which is sufficient to trigger a buckle, depends on pipeline properties, pipe soil resistance, and the imperfection. This is also illustrated in the left part of Fig. 5 with the upper hatched line giving the varying S_{init} along the pipeline section.

Figure 6 shows the compressive effective axial force, S_0 , as the temperature and pressure increase during start-up. As seen to the right, the force has

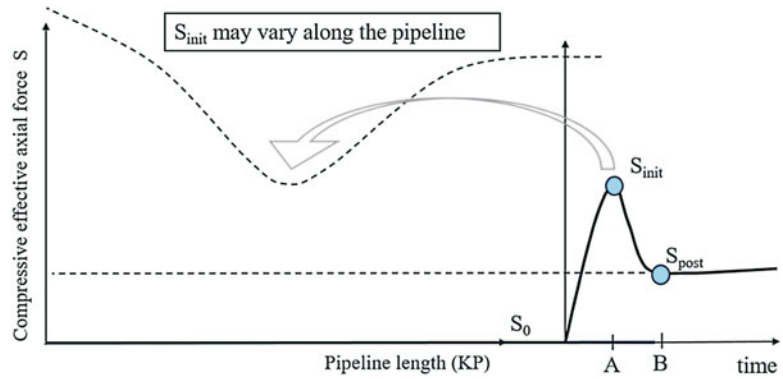
Local/Global Buckling and Propagation,

Fig. 4 Effective axial force, prior to buckling (DNVGL-RP-F110 2018)



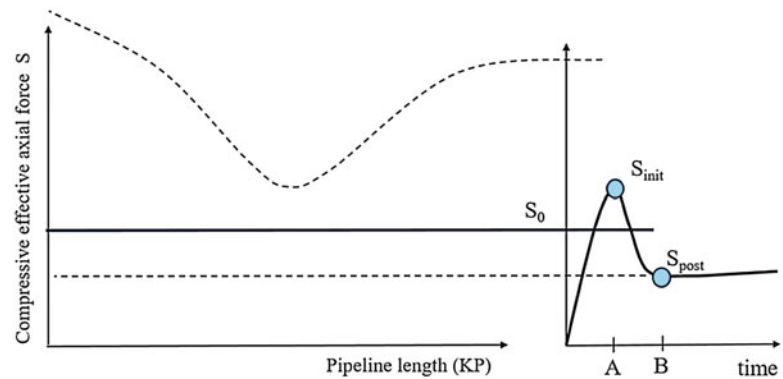
Local/Global Buckling and Propagation,

Fig. 5 General behavior during global buckling – critical buckling force (DNVGL-RP-F110 2018)



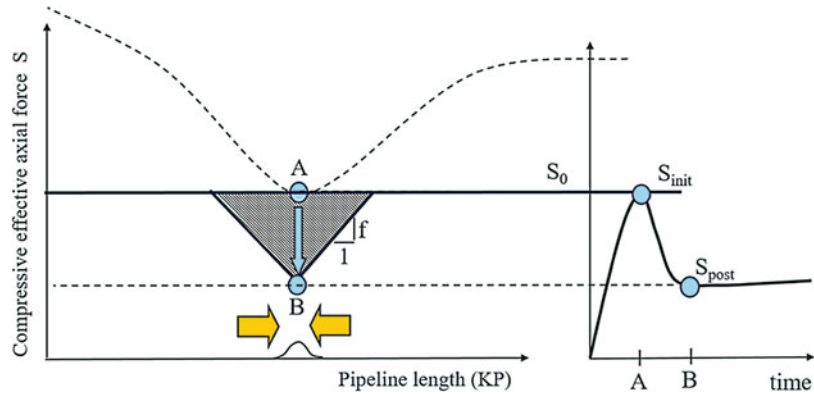
Local/Global Buckling and Propagation,

Fig. 6 General behavior during global buckling – start-up and increase of effective axial force (DNVGL-RP-F110 2018)



Local/Global Buckling and Propagation,

Fig. 7 General behavior during global buckling – effective axial force reaches critical value and a buckle is triggered (DNVGL-RP-F110 2018)



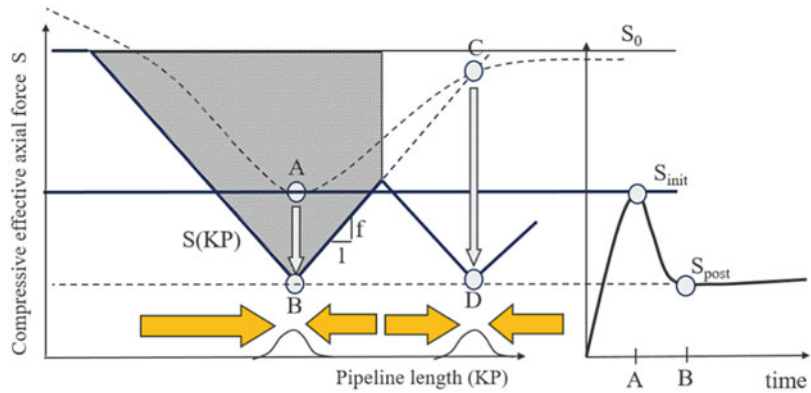
not reached the critical value, S_{init} , and no buckling has been triggered.

In Fig. 7 the temperature and pressure have further increased, and the effective axial force has reached the critical value, S_{init} , (point A). The pipeline buckles, and the effective axial force at the apex of the buckle will drop toward the post-buckling value, S_{post} (point B). The

effective force will gradually build up away from the apex of the buckle due to the axial pipe-soil restraining force or axial friction, f , giving the slope in the effective force diagram. The axial feed-in to the buckle (indicated by the yellow arrows) will be proportional to the shaded area between the solid line and the potential effective force line.

Local/Global Buckling and Propagation,

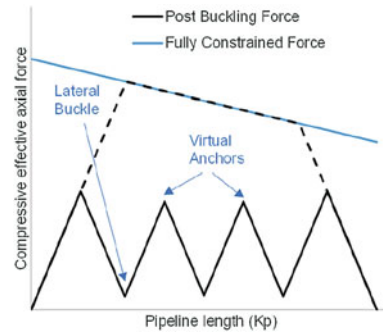
Fig. 8 General behavior during global buckling – pressure/temperature increased further and second buckle triggered (DNVGL-RP-F110 2018)



If the pressure or temperature is further increased, a neighboring imperfection may trigger a buckle and change the effective axial force diagram as shown in Fig. 8. From this point, the post-buckling force in the apex of the buckle (point B) will remain constant, but the axial feed-in to the buckle will increase, proportionally to the shaded area. As this axial feed-in is what drives the lateral buckle, the buckle will increase in deformation as the feed-in continues.

A typical fully developed force profile for a pipeline with multiple buckles and free ends is shown in Fig. 9. At each buckle, the effective axial force reduces as the adjacent pipeline section feeds in to form the buckle. The critical buckling force for each buckle varies, depending on the initial out-of-straightness, seabed frictional response, and buckle spacing. In the feed-in parts of the buckles, the slope of the effective axial force profile is governed by the axial friction. Between adjacent buckles virtual anchors are formed at positions of zero displacement, effectively dividing the system into a series of short pipelines anchored at each end. The virtual anchor spacing (VAS) associated with each buckle varies. The response of the pipeline between virtual anchors is analogous to the response between real anchors, and the loading caused by feed-in in to the buckle is characterized by the VAS.

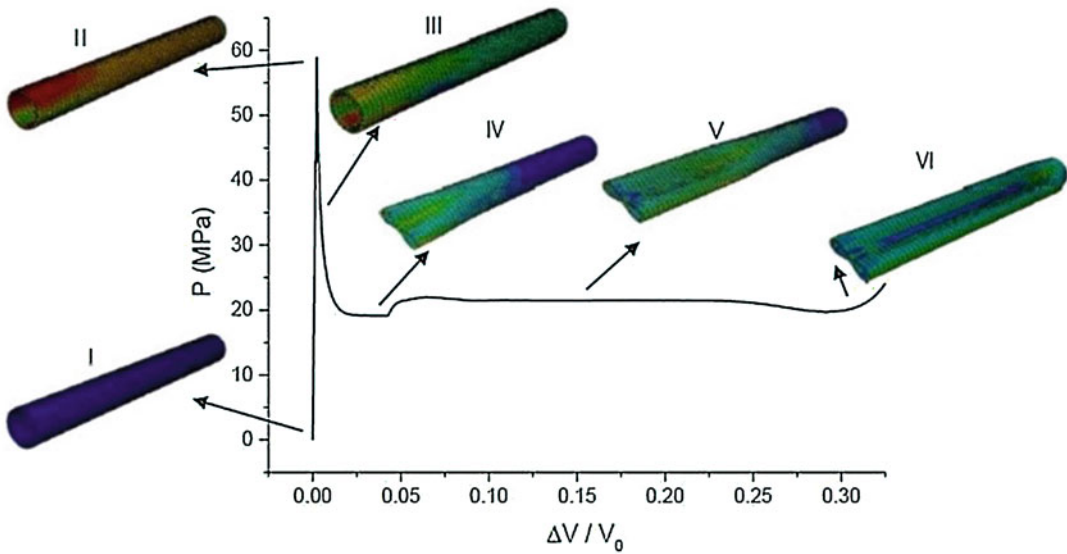
The probability of the initial pipeline configuration to share the available thermal expansion over several buckles or to feed into one single or fewer buckles are determined by the initial pipeline configuration and the pipe-soil interaction parameters.



Local/Global Buckling and Propagation,
Fig. 9 General behavior during global buckling – final configuration with multiple buckles (DNVGL-RP-F110 2018)

Critical buckling pressure is a key parameter that governs the local buckling response as well as buckling propagation of submarine pipelines.

Figure 10 shows a typical response of local buckling and buckling propagation. As seen to the left, the total volumetric change rate increases linearly as the external pressure increase. When the external pressure reaches a certain level is referred to as critical buckling pressure, a gross deformation occurs. The certain value is called critical buckling pressure. After that, total volumetric change rate increases rapidly, whereas the external pressure decreases. When the external pressure drops to certain level, the local buckling propagation is trigger and the local buckling propagates swiftly along the pipeline, as seen to the right in Fig. 10. The certain value is called buckling propagation pressure. The critical buckling pressure and buckling propagation buckling



Local/Global Buckling and Propagation, Fig. 10 A typical response of local buckling and buckling propagation for submarine pipelines

pressure are two key parameters for the design of submarine pipelines.

Key Influence Factors

The two primary factors that affect the global buckling response of submarine pipelines are initial imperfections and pipe-soil interaction.

The initial imperfections may be caused by uneven seabed, crossing arrangement, curvature in the horizontal plane purposely made or randomly imposed during installation, or engineered initiators such as sleepers or increased buoyancy. The initial imperfections are described by three parameters. The initial imperfection amplitude, wavelength, and shape are three primary parameters to describe the initial imperfection of a submarine pipeline. The global buckling is greatly affected by these three parameters as well. This point has been verified by many studies (Taylor and Gan 1986; Taylor and Tran 1993; Karampour et al. 2013; Zeng et al. 2014). The initial imperfection amplitude and wavelength are quite clear, and they have specific physical meanings. However, the initial imperfection shape is an unclear and unquantifiable parameter. A characteristic parameter of initial imperfection shape was

found by Zhang and Duan (2015) which can be quantified and can represent the effect of initial imperfection shape on the global buckling response. The characteristic parameter is the ratio of maximum curvature radius to the initial wavelength. This conclusion has been confirmed by Karampour (2018) and Chee et al. (2018). By now the effect of initial imperfections on the critical force of global buckling has been unified, as follows:

$$S_{\text{int}} = g(w_0/L_0, \rho_{\text{max}}/L_0)(EIq^2)^{1/3} \quad (2)$$

where EI denotes the flexural stiffness of the pipeline; q denotes the vertical load; w_0 and L_0 denote the initial imperfection amplitude and wavelength, respectively; ρ_{max} denotes the maximum curvature radius; and g is a coefficient function which represents the effect of initial imperfections on the critical of global buckling.

Global buckling behavior of exposed and buried pipelines is strongly linked to the pipe-soil interaction. The pipe-soil interaction includes vertical pipe-soil interaction, axial pipe-soil interaction, and lateral pipe-soil interaction. Vertical pipe-soil interaction is of major concern for

upheaval buckling, as it affects the mobilization load. Upward resistance and downward stiffness are two important parameters that influence the vertical pipe-soil interaction. A multi-linear model is normally used to describe the vertical pipe-soil interaction (Schaminee et al. 1990; Palmer et al. 2003; White et al. 2008).

The axial pipe-soil interaction is relevant when any buckling mode is triggered as it affects the post-buckling configuration. The axial feed-in from the straight sections into the buckled region is determined by the mobilized axial reaction. The axial pipe-soil interaction is also important for the axial force build-up, either during buckle development, at the pipeline ends or after a buckle has occurred. A high axial breakout resistance will influence the global buckling initiation, but to ensure conservatism in the design process, this is generally not accounted for. Therefore, residual axial resistance should be used to model thermal expansion and post-buckling behavior. A bilinear model is normally used to describe the axial pipe-soil interaction unless the real data for axial pipe-soil interaction is acquired.

For an exposed pipeline free to buckle laterally, the lateral pipe-soil interaction is one of the key parameters as it influences both the critical buckling load (breakout resistance) and pipeline post-buckling configuration (residual soil resistance after breakout). The lateral pipe-soil resistance is also the key parameter for the lateral buckling on even seabed as it influences both mobilization load (breakout resistance) and pipeline post-buckling configuration. At mobilization, when the pipeline starts to deflect laterally and the displacements are small, the lateral soil resistance is governed by the breakout value. For increasing lateral displacements, the lateral soil resistance may decrease to a lower residual value. In the tail of the buckle, where little lateral movement occurs, the lateral resistance will always be the breakout resistance, and this will influence the post-buckling configuration. A tri-linear model is normally used to describe the lateral pipe-soil interaction (White and Cheuk 2008).

The critical buckling pressure for submarine pipelines is primarily affected by diameter-to-

thickness ratio, elastic modulus and yield stress of material, initial imperfections, and corrosions.

For thin wall pipeline, the critical buckling pressure can be determined by the classic formula (Timoshenko and Gere 1961), as follows:

$$P_c = \frac{2E}{1-\nu^2} \left(\frac{t}{D} \right)^3 \quad (3)$$

For thin wall pipeline, the critical buckling pressure can be determined by experience formula (He et al. 2014), as follows:

$$\frac{P_c}{P_y} = -1.478 \left(\frac{D}{t} \right)^{0.611} \left(\frac{\sigma_y \cdot S}{E} \right)^{0.379} + 2.436 \left(\frac{D}{t} \right)^{-0.094} f_0^{-0.047} \left(\frac{\sigma_y \cdot S}{E} \right)^{0.049} \quad (4)$$

where σ_y is the yield stress of material, and P_y is yield pressure which can be determined by:

$$P_y = 2\sigma_y \frac{t}{D} \quad (5)$$

and S is a parameter describing the anisotropic yielding. It is defined as:

$$S = \frac{\sigma_{ya}}{\sigma_{yc}} \quad (6)$$

Where σ_{ya} and σ_{yc} denote the yield stress in the axial and circumferential directions, respectively.

The elastic modulus and yields stress of material are acquired by material experimental tests. The Ramberg-Osgood model is often used to represent the material constitutive relation (stress-strain curve). It is defined as follows:

$$\varepsilon = \frac{\sigma}{E} \left[1 + \frac{3}{7} \left(\frac{\sigma}{\sigma_y} \right)^n \right] \quad (7)$$

where n denotes the hardening parameter which can be determined by material experimental tests.

Initial imperfection existed in the submarine pipelines in the form of ovality, wall thickness

unevenness, corrosion, or local dent. The ovality describes the out-of-roundness of cross section of a pipeline which is induced during the manufacture of the pipeline and can be measured by:

$$\Delta_0 = \frac{D_{\max} - D_{\min}}{D_{\max} + D_{\min}} \quad (8)$$

where $D_{\max}$ and $D_{\min}$ denote the maximum and minimum diameter of the pipeline. Similarly, the wall thickness unevenness describes the wall thickness variations of a pipeline which is also induced during the manufacture of the pipeline and can be measured by:

$$\Xi_0 = \frac{t_{\max} - t_{\min}}{t_{\max} + t_{\min}} \quad (9)$$

where $t_{\max}$ and $t_{\min}$ denote the maximum and minimum wall thickness of the pipeline.

The loss of metal in a pipeline due to corrosion usually results in localized pits with various depths and irregular shapes on its external and internal surfaces which has a great influence on the critical buckling pressure. Details on this, please see Fatt (1999), Netto et al. (2007), Sakakibara et al. (2008), Netto (2010), and so on.

The buckling propagation pressure for submarine pipelines is primarily affected by diameter-to-thickness ratio and yield stress of material. It can be calculated by the classic result (Palmer and Martin 1975):

$$P_p = \frac{\pi}{4} \sigma_y \left(\frac{t}{r} \right)^2 \quad (10)$$

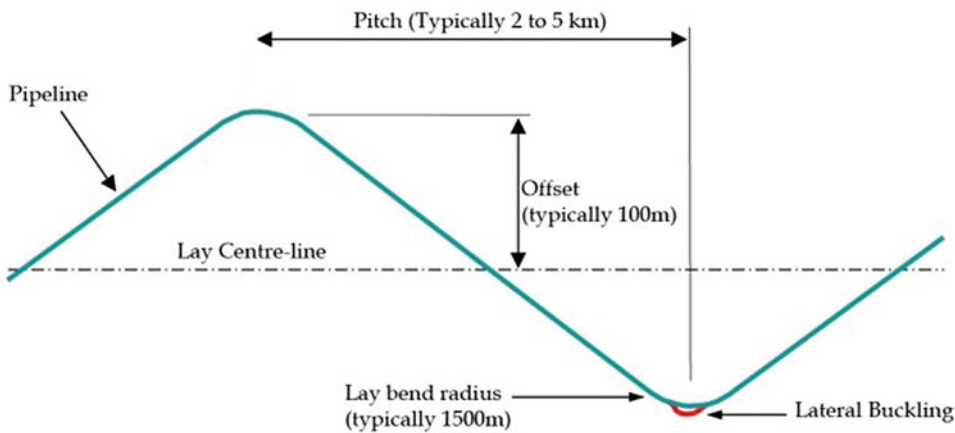
where P_p is the propagation pressure and t is the radius of the pipeline. Some modifications were made by previous studies (Kyriakides and Babcock 1981; Albermani et al. 2011; Gong et al. 2012).

Engineering Measurements

The structural integrity of a pipeline susceptible to global buckling can be assured by two different design concepts: (1) restraining the pipeline with the high compressive forces (buried pipelines) and (2) allowing the pipeline to buckle globally and thereby releasing the expansion forces (exposed/unburied pipelines).

For vertical buckling, burial is the most common method to prevent pipeline buckling. Trenching and backfilling, rock dumping are two primary methods to bury a pipeline against the vertical buckling.

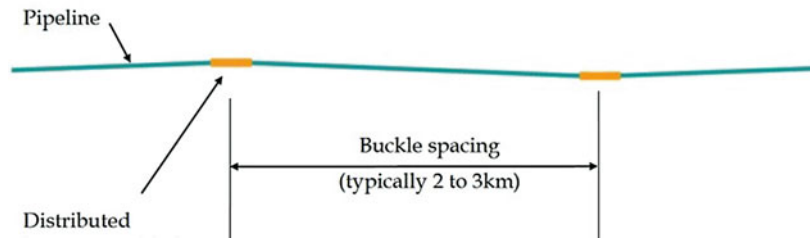
For lateral buckling, snake-lay (Fig. 11), vertical upset (Fig. 12), and distributed buoyancy (Fig. 13) are three primary methods to control the lateral buckling. The critical axial force of lateral buckling for the submarine pipeline is decreased by these ways. Therefore, the lateral buckling is easy to trigger and the expansion forces are released by controlling the formation of lateral buckles along the pipeline. The buckled pipeline can withstand more compressive axial force. The uncontrolled lateral buckling becomes



Local/Global Buckling and Propagation, Fig. 11 Typical snake-lay configuration (Bruton et al. 2005)

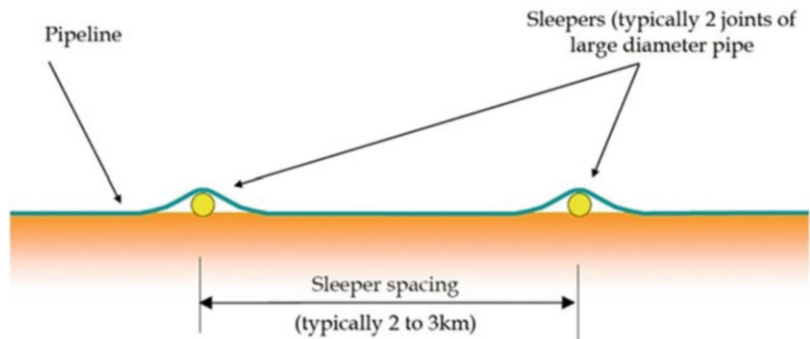
Local/Global Buckling and Propagation,

Fig. 12 Buckle initiation using distributed buoyancy (Bruton et al. 2005)



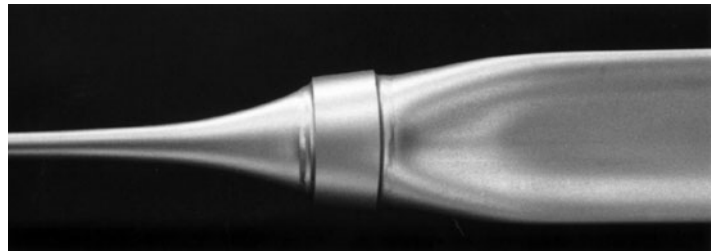
Local/Global Buckling and Propagation,

Fig. 13 Buckle initiation using sleepers (Bruton et al. 2005)



Local/Global Buckling and Propagation,

Fig. 14 Buckling arrestor (Netto and Kyriakides 2000)



controlled. More details please see Harrison et al. (2003), Bruton et al. (2005).

The buckling propagation can be controlled by installing buckle arrestors, at intervals along the pipeline, Fig. 14. The purpose is to limit the extent of damage to the pipeline by arresting the collapse propagation. Buckle arrestors are devices that locally increase the bending stiffness of the pipe in the circumferential direction, and therefore they provide an obstacle in the path of the propagating buckle. More details please see Toscano et al. (2008).

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Localization and Planning

► Simultaneous Localization and Mapping

Long Baseline (LBL)

► Autonomous Underwater Vehicle (AUV)

► Long Baseline Underwater Acoustic Location Technology

► Ultra-short Baseline Underwater Acoustic Location Technology

Long Baseline Underwater Acoustic Location Technology

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Synonyms

Acoustic synthetic baseline (ASBL); Direct sequence spread spectrum (DSSS); Doppler velocity log (DVL); Inertial navigation system (IMU); Inertial navigation system (INS); Long baseline (LBL); Multi-user long baseline (MULBL); Short baseline (SBL); Sparse long baseline (Sparse LBL); Super short baseline (SSBL); Ultra-short baseline (USBL)

Definition

The underwater acoustic positioning system that is used to track underwater vehicles and divers can be divided into a **long baseline LBL (Long Baseline)**, a short baseline SBL (Short Baseline), and

an ultra-short baseline USBL or SSBL (Ultra-Short Baseline/Super Short Baseline) system according to the array spacing. The length of the long baseline underwater acoustic positioning array is generally 100–6000 m. It is arranged on the seabed according to a certain geometric shape. The adjacent three or four array elements form a matrix, which is used to measure the underwater target sound source to each element. Distance determines the location of the target. With the development of underwater acoustic localization technology, distributed underwater acoustic localization technology has become a research hotspot in recent years. Its baseline length is longer, up to tens of kilometers. It uses underwater submerged nodes as positioning beacon arrays. Its form and function are very similar to underwater communication networks (Vickery 1998).

At present, the long-baseline underwater acoustic localization technology is relatively mature. Because of its long range and high positioning accuracy, it has a wide range of applications in underwater shooting range, underwater construction, submarine cable laying, offshore oil exploration, and marine geological monitoring. At the same time, the long-baseline positioning technology system is complicated, the system is difficult to deploy and recycle, and the untimely factors such as the time-consuming array of the array are limited, which limits the application in some scenarios.

System Structure

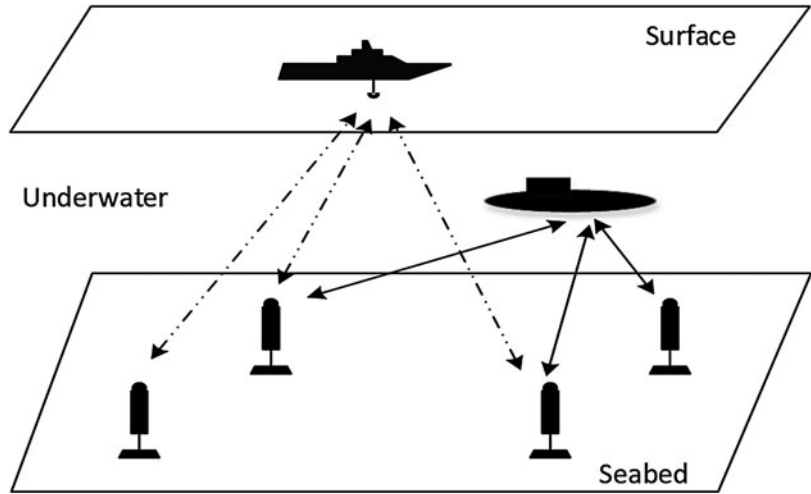
The long baseline underwater acoustic positioning system consists of a question and answer (transceiver) and a transponder (beacon). The transceiver is usually installed on the target to be measured, such as surface ships, submarines, torpedoes, UUVs. When positioning and navigating the target to be measured, multiple transponders are required to form a transponder array, and the measured target motion is generally located in the array.

There are three methods for measuring the array of underwater beacon arrays, such as absolute array, relative array, and joint array. The absolute array is to calibrate the beacons one by one or

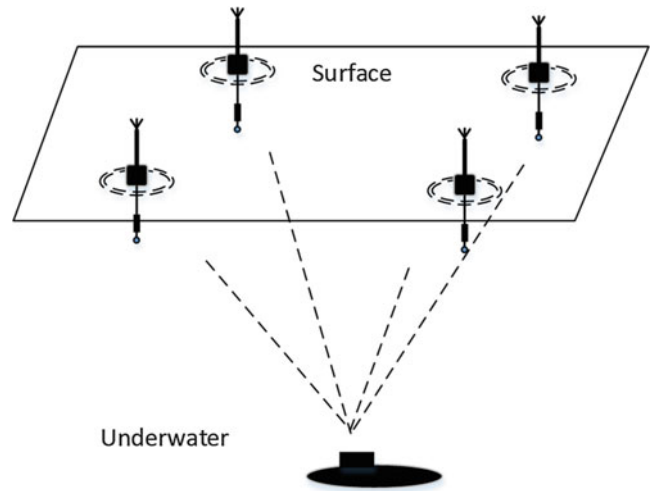
multiple standard groups. Generally, the mother ship uses the range finder to draw a single beacon, and the distance measurement and the angle sensor are used to correct the ranging and angle information. The baseline solution mode solves the beacon coordinate position. This kind of traditional absolute array technology reduces the calibration efficiency with the increase of depth in the deep sea operation, which takes a lot of time, and is greatly affected by the sea state, the mother ship track, and the sound velocity distribution, and it is difficult to obtain high-precision calibration results. With the development of software and hardware technology, beacons have a variety of functions such as range, communication and the development of relative array technology, which uses the long baseline principle to measure the distance between beacons, and finally uploads by the surface unit. The information is solved to obtain the relative formation, and then the ship's position information is used to obtain the geodetic coordinates of the formation (Zhang et al. 2016; Larsen 2000). Because the underwater environment is stable, the beacon is relatively stationary, and the difference in sound velocity changes is small, so the time is less, and the accuracy of the obtained array information is higher. The joint array technology combines the absolute array and the relative array technique. The principle is to calculate the absolute position coordinates of the matrix based on the relative position coordinates of the beacon and the absolute position coordinates of any two beacons.

Both underwater beacons and buoys can act as a matrix of transponders. When the beacon array is formed by the beacon on the seabed, the target to be measured may be an underwater or water surface target, and the system is constructed as shown in Fig. 1. When the array is fixed, the beacon is fixed or anchored to the seabed. If the system is not used for a long time, an acoustic release device can be added between the beacon and the anchor. When working, the question answering machine on the target to be tested first sends an inquiry signal, and the beacon in the underwater beacon array sends back a response signal after receiving the signal, and each beacon response signal is on a different channel. The time taken by the measured target from the issuance of

Long Baseline Underwater Acoustic Location Technology, Fig. 1 Beacon response array system composition diagram



Long Baseline Underwater Acoustic Location Technology, Fig. 2 Buoy response array system composition diagram



the interrogation signal to the receipt of the response signal of the beacon is twice the acoustic propagation time between them, so the relative coordinates between the measured target and the beacon can be calculated. Since the beacon is placed and positionally calibrated, the geographical coordinates are determined, so that the trajectory of the measured object relative to the earth during continuous motion can be further obtained.

For the long baseline buoy positioning system, as shown in Fig. 2, the underwater beacon is replaced by a radio buoy on the sea surface, and the underwater target is equipped with synchronous or asynchronous beacons. When the beacon (sound source) and the buoy (measurement system) on the target are equipped with a high-

precision synchronous clock system, the satellite navigation and positioning system such as GPS is generally used to perform timing and correction on the device, that is, between devices in the system. There is a timing error, and the system formed at this time is called asynchronous positioning system, otherwise, it is called an asynchronous positioning system. The buoy is composed of a building block. The top of the support bar on the water is a wireless communication antenna, a positioning device (GPS/BD) antenna and a beacon light. The lower end is equipped with a floating body to provide buoyancy and then connected down to the electronic compartment. The battery is installed in the cabin, radio stations, positioning, attitude measurement and other equipment,

hydrophones, transducers, and other equipment installed below the electronic compartment.

The communication base station installed on the work ship serves as the central station of the radio station. The data such as hydrophone, sensor, and positioning information are uniformly packaged by the radio buoy and transmitted to the work ship. At the same time, the work ship is equipped with display and control equipment (table). Solve the positioning position of the underwater target. If there is a data confidentiality requirement, the wireless transmission data can be encrypted.

In the specific use of the long baseline positioning system, depending on the actual application and application, the system configuration sometimes uses the underwater beacon array and the surface buoy array.

Positioning Principle

The measurement of the azimuth and distance (or distance difference) is performed by using the pulse signal transmitted by the measured object to arrive at each array element delay (or delay difference) information. The long

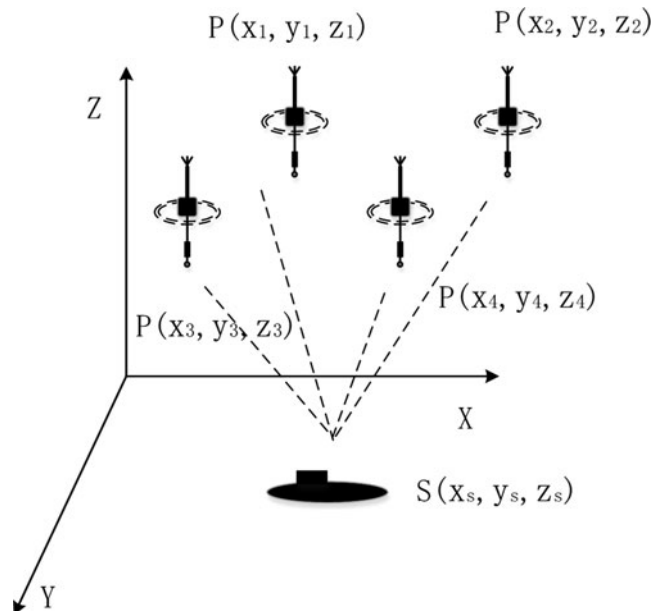
baseline system positioning solution method has multiple solution models, and the classical and representative spherical intersection method and hyperbolic intersection method are compared. Synchronous system solutions generally use a spherical intersection model, and nonsynchronous system solutions generally use a hyperbolic intersection model.

As shown in Fig. 3, where the coordinates of the i -th buoy is $P(x_i, y_i, z_i)$, the measured target position to be solved $S(x_s, y_s, z_s)$, which is determined by measuring the propagation delay of the measured object to each buoy. The distance from the target to each buoy, that is, the slant distance between $P(x_i, y_i, z_i)$ and $S(x_s, y_s, z_s)$, solves the target position by geometric intersection. For a single buoy, the propagation delay determines the spherical surface on which the sound source is located, and the spherical equation determined by the i -th buoy is

$$\begin{aligned} (x_i - x_s)^2 + (y_i - y_s)^2 + (z_i - z_s)^2 \\ = c^2(\tau_i - \tau_b)^2 \end{aligned} \quad (1)$$

where τ_i is the propagation delay of the i -th buoy from the target measured by the system; τ_b is the

Long Baseline Underwater Acoustic Location Technology,
Fig. 3 Long baseline positioning system working principle diagram



delay difference between the signal transmission time of the target (beacon) source and the measurement system (buoy) clock; c is the speed at which sound waves travel in water.

For the synchronous positioning system, the formula (1) becomes the following formula (2)

$$(x_i - x_s)^2 + (y_i - y_s)^2 + (z_i - z_s)^2 = C^2 \tau_b^2 \quad (2)$$

According to the geometric principle, the two spherical surfaces intersect in one circle, and the three spherical surfaces are compared with two points. The intersection of the four spherical surfaces can determine the only point in the space. In general, the target value z_s is known because the target is installed with a depth sensor, and the “spherical intersection model” of Eq. (2) will be transformed into the “circle intersection model.” The two circles intersect at two points (one point when tangent), and if there is a third circle equation, a unique point on the plane can be determined. Therefore, in the case where the depth is unknown, four buoy array elements are needed, and in the case where the depth is known, only three buoy array elements are needed to solve the unique target solution.

For the nonsynchronous positioning system $\tau_b \neq 0$, if it is an unknown number, if the spherical intersection model is still used, a large positioning error will be generated. Although the absolute delay of the measured target to each buoy cannot be known, the delay difference between the measured target and each buoy is a fixed value. According to the analytic geometry, the distance to the two focal points is a fixed value, and the corresponding quadratic equation can determine a hyperboloid. Therefore, the nonsynchronous positioning system solving model is called a “hyperbolic intersection model.” If the measured target depth is unknown, the positive value of the two sets of solutions is the measured target position. When the measured target depth is known, the three array elements can determine the target position double solution. If the double solution is a positive number, the unique solution is obtained by the redundant array element (Janik 2005; Baccou et al. 2001).

Application and Development Trends

The long-term baseline positioning methodology has developed earlier, forming a mature industry, and the products have been serialized. The history of the first long-baseline system can be traced back to 1958. The Applied Physics Laboratory of the University of Washington in the United States built a three-dimensional coordinate tracking underwater shooting range in Dabo Bay, using four transponders placed on the seabed to achieve continuous underwater targets tracking. At present, well-known companies engaged in long-term baseline positioning abroad include French iXblue, Norwegian Kongsberg, and British Sonardyne. In fact, the above companies are involved in the long baseline, short baseline, and ultra-short baseline positioning, as well as some companies specializing in the development of long baseline positioning systems, Benthos, SonarTech, ORCA, etc.

iXblue has a strong technology and R&D team, and its positioning products can combine INS, DVL, GPS, and other equipment to form a comprehensive underwater navigation system. RAMSES is an Acoustic Synthetic Baseline (ASBL) positioning system that makes long-baseline and sparse long baseline (Sparse LBL) positioning easier. The system combines inertial navigation systems to improve positioning with significantly reduced transponder usage. Accuracy and system robustness, the system operating frequency is intermediate frequency and can provide better positioning accuracy than 0.1 m within 4000 m distance. Canopus is the new, most advanced smart transponder. The system's pressure, temperature, and tilt sensors collect environmental information, and the Embedded User Interface allows users to connect to devices using WIFI. The system has three working modes, long baseline positioning, ASBL sparse long baseline positioning and underwater acoustic data telemetry, and Modem functions, and is compatible with third-party acoustic systems, which is a development trend of the foreign acoustic industry. Kongsberg has a number of mature products in the military and civilian fields. The cPAP series transceiver is a high-precision long-baseline positioning system

developed for ROV and cable carriers. It can be connected to the acoustic positioning computer APC or the acoustic positioning console APOS through the data interface. The cPAP series includes an intermediate frequency cPAP 34 and a low frequency cPAP 17, with a maximum working depth of 4000 and 7000 m, respectively. In 2015, the new HiPAP series was introduced to achieve independent array calibration. The new Cymbal protocol applies direct sequence spread spectrum (DSSS) wideband signal processing technology to positioning and communication. HiPAP supports long baseline, short baseline positioning, and multiuser long baseline system (MULBL) mode. Single transponders work in short baseline mode; when working in long baseline mode, at least three transponders are required on the seafloor, typically four to six transponders; multiuser long baseline mode is available. The same beacon array provides positioning for multiple users including ships, ROVs, etc., in the area. Sonardyne has developed a series of long baseline positioning systems, such as the Prospector series, Dunker6 series, Fusion 6G series, and ROVNav series (Zhang et al. 2019). The sixth generation of broadband signal technology claims to provide an unlimited number of users within its system coverage (Sun and Zheng 2015).

With the continuous advancement of hardware, software technology, and the demand for industrial applications, the long baseline underwater acoustic positioning system shows some common development trends (Yang et al. 2006; Sun et al. 2012; Takasu and Yasuda 2010; Sun et al. 2019).

1. The signal system develops from narrowband to broadband, which has higher gain than conventional single-frequency signal processing and stronger antinoise capability.
2. Multisystem combined positioning, the system is not simply a long baseline system, combined with short baseline, ultra-short baseline, forming LBL/SBL, LBL/USBL, LBL/SBL/USBL combined positioning system.
3. Multifunctional integration, in addition to long baseline positioning function, integrated underwater acoustic communication, environmental parameter measurement, and other functions.
4. Multisensors are used in combination, integrating INS/IMU, DVL, etc., in the underwater acoustic positioning system, positioning and solving the data of the fusion multisensor system, greatly improving the positioning ability and accuracy.
5. Multitarget positioning, which can be developed by locating a single target from a single target, and even providing a positioning navigation service to an unlimited number of targets in the coverage area.
6. Strengthen the processing of positioning data, and apply techniques such as least squares estimation and Kalman filtering to improve the quality of positioning data.
7. Convenient operation, the system can automatically calibrate and calibrate the underwater array of water surface, even without calibration, reduce the operating cost, so that users have more energy to focus on the positioning service requirements.

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Long-Baseline (LBL)

- [Integrated Navigation](#)

Long-Baseline System (LBL)

- [Obstacle Avoidance Technology for Underwater Vehicle](#)

Longitudinal Stability

- [AUV/ROV/HOV Stability](#)

Long-Line Aquaculture

- [Traditional Aquaculture Structures](#)

Long-Term Storage

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Definition

Aged or severe damaged vessels or offshore structures are decommissioned or cut into several parts.

Due to some inescapable factors, such as low value, high pollution, radioactive nuclear waste, hard to divide, harmful to people, etc., the structures could not be modified for reuse nor dismantled into components. Hence, the whole structure would be stored at a place for a long period, say usually 10 years or more, waiting for the further processing.

Scientific Fundamentals

Background and Introduction

The oil and gas sector is facing a sharp rise in decommissioning spending, forecasting an increase from around US\$ 2.4 billion in 2015 to almost US\$ 13 billion, per year, by 2040 (Correa and Russel 2017). While in China, there will have 81 decommissioning oil fields, 164 offshore platforms, 3240 km pipelines, and 2310 subsea wells by the end of 2040. From this aspect, it is a great market of dismantling offshore platforms all over the world. It is necessary to rapidly develop the technologies and facilities for decommissioning offshore oil and gas platforms.

The decommissioning process provides an opportunity for financial gains due to the increased resources available for environmental and socio-economic benefits (Sudaia et al. 2018). Briefly, the decommissioning procedure consists of transporting the platform to an onshore dock, where it is dismantled and the parts removed for future sale and reuse (Dexter and Ghorashi 2016; Junior Alves and Souza 2017; Rouse et al. 2018).

Comparing with the onshore structures and pipelines, the issues of decommissioning offshore ones is much more complex. It is not only limited by the domestic and international laws or rules, the cost of decommissioning an offshore structure is more expensive than an onshore one. Besides, for some high pollution or special functional offshore platforms, when they are at the end of their life, they can hardly be exploded and removed directly. Otherwise, severe environmental catastrophe will take place. One possible approach is to tow or transport the whole structure or the divided parts to a certain place, say storage yard, and store them for a long-term period, waiting for the further processing, such

as modification, potential buyer, deep decommissioning, etc.

Before the structures are stored in place, they should be carefully evaluated whether they are valued to be long-term storage. Among different assessments, the potentiality of those retired structures are one of the key factors, and it is also the first step of the old engineering structures recycling. A reasonable determination will directly lead to the benefits in society, economy, environment, and so on.

For those structures or facilities that are being long-term stored, they will be further modified or used on other places. In other words, they are waiting for reusing or recycling. According to previous successful experiences, we found that core of the structure recycling is that they are easy to be constructed. At the beginning stage of decommissioning, due to the usage of the structure is not in decision, the potentiality will determine the chance of its future reuse. Moreover, during the recycling project, the constriction conditions, including the difficulties in modification, costs, and so on, will also affect the final determination.

According to previous investigations, we know that the potential recycling abilities of decommissioning structures should be determined by the following five factors, including geographical position, constriction condition, culture, economic value, and environmental adaptability. These factors are also the key elements on the judgment whether a decommissioning structure could be long-term storage until it is reused in somewhere.

Among various evaluation approaches, the TOPSIS (Technique for Order Preference by Similarity to an Ideal Solution) approach has become one of the most commonly used way to assess potentiality of the retired structures. Although TOPSIS theory is firstly proposed by C. L. Hwang and K. Yoon in 1981, it is still a classic method to estimate the relative superior of the objects. It can obtain the similarity between each alternative and the ideal solution by calculating the weighted Euclidean distance between each alternative and the positive and negative ideal solution, to serve as the criteria for prioritizing the risk level of FMEA failure modes (Liu et al. 2015).

Another potential issue during the long-term storage is the influence on the onshore and

offshore environment. Taking concrete structure as an example, after its decommissioning the structure will be divided into several parts as well as dust or wastes. It is obvious that not all of the decommissioned structured could be reused directly. Hereby, they need to be stored in somewhere and wait for the following command, and others will be buried or further proceeded. However, there may exist some harmful materials to the environment.

On one hand, comparing with the soils on earth, the decommissioned structure sometimes contains heavy metal; such as plumbum or zinc; total organic and inorganic carbon, which may have potential environmental risks for the soil; underground and surface water. On the other hand, there exists a majority of calcium silicate hydrate and calcium carbonate hydrate. If the storage scenarios are wet enough, these compounds will induce strongly alkaline leachate, which pH value is larger than 11. These will undoubtedly pollute the environment.

Therefore, the storage facilities for those potentially contaminative structures should be designed carefully. Two major principles should be obeyed, which are safety and economy, respectively. In fact, the safety factor is the basic principle. From the aspect of avoiding pollution, the storage place should have least effects on the external environment during its construction and operation. This will help to protect the atmosphere, soil, and surface and ground water from pollution. Besides, the economic factor will ensure the storage cost in a relative low level. Hereby, different elements, including site selection, capacity, transportation distance, and so on, need to be taken into consideration simultaneously.

The long-term storage of decommissioned structures can be managed according to classic management modes. Usually these modes consist of three types, which are public owned-private management (POPM), public-private partnership management (PPPM), and franchising management (FM). The differences between these managements are listed in Table 1.

The POPM mode is that the facilities are invested and established by the government. Then they are outsourced to the private or other companies to operate and maintain. That is to say,

Long-Term Storage, Table 1 Long-term storage management

Type		Property	Operation maintenance	Investment	Commercial risk
POPM	Service contract	Public	Both	Public	Public
	Management contract	Public	Private	Public	Public
	Leasing contract	Public	Private	Public	Private
PPPM	Property selling	Both	Both	Both	Both
	Quasi-BOT	Both	Both	Both	Both
FM	BTO	Private-government	Private	Private	Both
	BOT	Private-government	Private (during contract)	Private	Private (during contract)
	TOT	Government-private-government	Private (during contract)	Public	Private (during contract)

the property belongs to the government, while the management right is attributed to the companies. Specifically, it contains three major approaches, which are service contract, management contract, and leasing contract, respectively.

The PPPM mode is that a new company is established by both private company and government. Based on the investment ratio, the commercial risks and profits are both proportionately shared. There usually are two types in this mode, property selling and quasi-BOT mode.

The FM mode is that the major part or all of the investment are provided by the private company. According to specific cooperation ways, the company and government will share the risks and profits. Based on the actual income of the projects, the government will collect the operating expenses/compensate the loss from/to the company. However, because the final ownership belongs to the government, there exists a progress of the ownership and management right transformation. Specifically, there are three major types, including build-transfer-operate (BTO), build-operate-transfer (BOT), and transfer-operate-transfer (TOT) modes.

Key Applications

Long-Term Storage for Normal Offshore Structure Wastes

For normal decommissioning offshore structures, most of the demolished components are hardly

used in the present stage. In other words, they could be classified into structure waste. These structure wastes have the following characteristics, including large in amount, huge variety, high pollution, etc. Besides the giant cost to proceed these wastes, it will also require a big place to store them before they are proceeded or recycled. Therefore, it has induced a series of issues in both environment and society. Moreover, with the development of society, more and more aged offshore structures will be abandoned, which, hereby, will cause a much more severe problem in the future.

Actually, some of the offshore structure decommissioning wastes could be recycled after some working procedure, even though a small part of them cannot be easily handled with current technology. Hereby, some processing experiences of onshore structures decommissioning wastes could be referred.

Taking Shanghai 2010 EXPO construction as an example, based on its construction progress, the decommissioning projects are flexibly adjusted. In order to balance the wastes produced by decommissioning and the requirements from construction, the decommission plan will be adjusted in real time. The committee designed a temporary storage place to arrange the structure wastes. Its main function is buffer and information feedback. When the waste production exceeds the requirement of the construction, the waste will be transported to the storage place, waiting for the

following commands. However, the storage place is area-limited. Thus, when the stored wastes reach threshold, the information will be sent to the executive center, and the decommissioning plan will be adjusted to reduce the waste generation.

We believe that such approach also has significant guidance to the offshore structures' decommissioning and long-term storage. It is possible to adopt this in-time adjusting approach in the offshore structures decommission and constructions in the shipyards. However, it should be noted that this method may not be available for the offshore installations, as the installation cost is in very high level.

Long-Term Storage for Radioactive or Other Polluting Offshore Structure Wastes

Apart from the common wastes of the decommissioning offshore structures, such as concrete, steel, jacket pipe, etc., there are also

some radioactive components in the offshore structures, like floating nuclear plant (FNP) and nuclear power ship. Although their design lives are usually more than 40 years, it will undoubtedly retire in the future. As a special type of energy supplier, nuclear fuel provides a lot of benefit for our life as well as the offshore structures. However, how to proceed its waste has become a worldwide issue.

The 70 MW Akademik Lomonosov Floating Nuclear Co-generation Plant (the first FNP all over the world, see Fig. 1) was established since 2007, and it was towing to its operation area in August 2019. There are two nuclear reactors in this FNP. Each reactor is enclosed in a steel hermetic containment vessel to withstand the pressure. It will have a lifespan of 40 years.

After its lifetime ends, the FNP cannot be easily decommissioned, especially the nuclear cores.



Long-Term Storage, Fig. 1 Floating nuclear co-generation plant. (Source: www.rosatom.ru)

Nuclear decommissioning is the final technical and administrative process in the life cycle of nuclear power operation. As such, decommissioning must strive to ensure public safety. This goal requires many factors be involved in the decision-making process (Suh et al. 2018). Typically, a 40–100 years of storage are usually involved in storage strategy. Most of the wastes are long-term stored in dry casks at various locations (Yano et al. 2018). Licenses for the storage of spent nuclear fuel (SNF) and vitrified highly active waste in casks under dry conditions are limited to 40 years and have to be renewed for prolonged storage periods (Gerold 2018).

At present, the United States does not have a designated disposal site for used nuclear fuel – the nation therefore faces the prospect of extended long-term storage (i.e., >60 years) and deferred transportation of used fuel at operating and decommissioned nuclear power plant sites (Chopra et al. 2014). An investigation on poly-ether ether ketone (PEEK) composite for long-term storage of nuclear waste highlights the effect of radiation and chemical and thermal environments on mechanical and thermal properties of PEEK and presents which of these materials could be an alternative material for long term storage of nuclear (Ajeesh et al. 2015). The results they have obtained are shown as follows: (1) the dispersion of carbon short fiber in the PEEK Composites is significantly uniform from the study under optical microscopy; (2) there are no significant changes in thermal properties of PEEK composite when exposed to aggressive environments from differential scanning calorimeter (DSC) and thermo gravimetric analysis (TGA). As a result, to determine the storage period and material could be the key problem of long-term storage of nuclear waste with offshore structure.

Fatigue Problem During Long-Term Storage

Long-term fatigue assessment of an offshore structure is a challenging practical problem, no matter during its working or storage periods. Various efficient strategies have been proposed in the literature; however, they generally suffer from one or more shortcomings. Low (2016) proposed a new analysis approach, based on using the

classical control variates technique to improve the efficiency of Monte Carlo simulation (MCS). According to his case study, the results show that substantial reduction in the sampling variability of MCS are provided. Time domain simulation is another common method to predict the long-term fatigue damage in offshore engineering. Taking the high-pressure hydraulic power take-off (PTO) machinery in a heaving-buoy wave energy converter (WEC) as example, Yang and Moan (2013) presented the fatigue analysis with time domain simulation. They obtained long-term pressure loads by combining the short-term time simulation results and the occurrence probability of each sea state. Two models (Weibull and generalized Gamma distributions) were applied to fit the long-term rain flow-counted pressure cycles.

Corrosion Problem During Long-Term Storage

Corrosion problem is one of the significant issues during the offshore structure operation, due to the wet and salty ocean environment. During its lifespan, there are some approaches, such as sacrificial anode, polymer anticorrosive material, duty paint, etc., to solve or relieve this slow chemical reaction. However, during the long-term storage, the corrosion protection should be paid attention to, as most of the protection terms are out of work. If severe corrosion occurs on the storage structures, it will clearly affect the following reuse.

Moreover, there is a potential way to store the decommissioned offshore structures in some underground area, if it is not against local law. Subsurface developments have many potential uses ranging from defense installations to transport and retail outlets. Besides, it has great potential ability for the storage. If this storage is adopted, the corrosion issue will be more critical to the structures. Gschwandtner and Galler (2018) assessed the stability of underground structures during a long-time behavior. According to their investigation, geotechnical engineering and geology, hydrogeology in particular, show great influences on the behavior of structure and surface area. They conducted a four-year-length survey, as well as a numerical simulation, to perform the

study. Their results show that the influence of underground and surface waters, especially for those areas that are sensitive to water, cannot be neglected in the consideration of long-term stability.

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Loss of Stability

► Pure Loss of Stability

Low Cycle Resonance

► Parametric Rolling

Low Earth Orbiting (LEO)

► Underwater Acoustic Communication

Low-Density Parity Check (LDPC)

► Underwater Acoustic Communication

LS – Limit States

► Design of Renewable Energy Devices

Luxurious Cruise Ship

► Luxury Cruises

Luxury Cruises

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Synonyms

Brilliance of the Seas; Luxurious cruise ship; Luxury passenger liner

Definition

Luxury cruise is regarded as the “mobile palace in the sea” (Yin 2012), which is the high-valued equipment for the tourism industry. It provides tourists with unforgettable sailing experiences and incredible enjoyment through its different types of facilities.

Scientific Fundamentals

Development History

In the late nineteenth century and the early twentieth century, people traveled across the sea only by cruise ships, which just provided passengers with limited rooms and food services. With the increasing maturity of aircraft technology, in order to save time and money, many passengers choose to travel by airplane. Therefore, in the middle of the twentieth century, the cruise ship industry declined. In order to survive, cruise companies developed the concept of cruise vacation (Duman and Mattila 2005), which transformed the transport-oriented cruise ship into a marine holiday village and built the cruise ship with more luxurious facilities, more programs, and larger displacement. Therefore, luxury cruise has been gradually more and more flourishing since the 1980s (Shan et al. 2017) and has become an independent part of the world tourism industry.

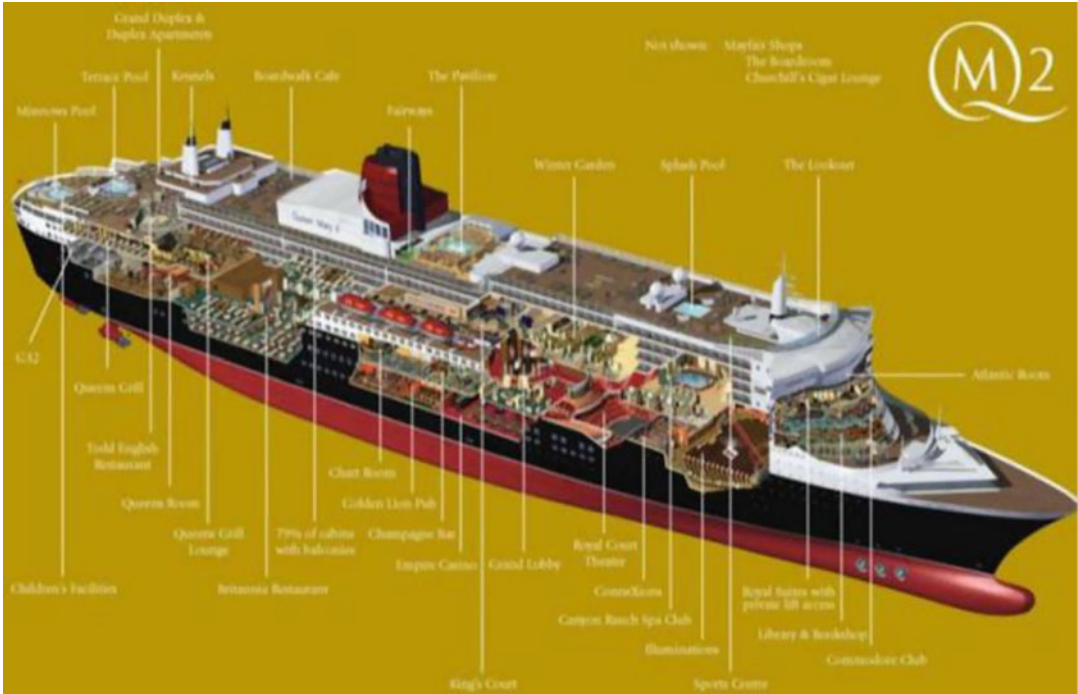
Structural Features

As a typical representative of the high-end product of manufacturing industry, the luxury cruise has integrated the features of advanced manufacturing with the concept of modern services, sightseeing, tourism, and entertainments. The luxury cruise is totally different from mainstream ships in design concept, construction technology, operation, management, and other aspects, which directly reflect the scientific and technological level and comprehensive industrial ability of a country. Compared with the conventional ships, luxury cruise has the following characteristics (Luo and Gan 2017):

1. The molded breadth is relatively large, while the molded depth, relatively small.
2. Double bottom, double hull, long superstructure, square stern, curved nose bow, and underwater bulb.
3. There are many water-tight compartments in the main ship, all of which are plane bulkheads. The deck, board side, and bottom are longitudinal structures. The framework is mostly constructed by flat bulb steel.
4. The superstructure is divided according to its functions, and the public entertainment place is spacious with few pillars.
5. The visible structure focuses on coordination and beauty. The upper and lower structures are designed to be aligned.
6. The high requirements of vibration control and noise elimination.
7. The type of rooms in the luxury cruise involves standard room, sea-view room, superior royal suite, and so on. For the specialty in comfort and function, the structural type and layout in rooms are also different (Fig. 1).

Difficulties in the Structural Design of Luxury Cruise

The structural design of luxury cruise should not only meet the intensity requirements of classification regulations but also comply with its mission of providing high-end services to tourists. The design difficulties can be summarized as follows:



Luxury Cruises, Fig. 1 Basic structure of luxury cruise (<http://image.baidu.com/>)

Plate Welding Distortion Control

High-strength steel has been widely used in shipbuilding. Its beneficial effect is to reduce the weight of the structure and improve the navigation performance and energy efficiency of ships. However, it also brought some negative effects, one of which was that as the thickness of the plate decreased, the welding deformation increased. The thickness of the hull structure of a luxury cruise should be relatively small, especially for superstructures and deck compartments. Most of the plate thicknesses can be among 5–6 mm. Large welding distortions above the waterline seriously affect the aesthetics of the hull, while larger welding distortions below the waterline will not only affect the navigation performance of the vessel but also reduce the ability of the vessel's external plating to withstand the effects of water pressure. The exterior appearance of the luxury cruise ship needs to be aesthetically pleasing. Therefore, the exterior surface of the ship must not exhibit any aesthetic welding distortion.

Vibration and Noise Control

Tourists take a luxury cruise tour to have the relaxing and enjoyable life. However, ship noise can not only influence the normal navigation of the ship but also reduce the comfort of passengers and sailors, causing complaints and dissatisfaction. Therefore, luxury cruises must control the vibration of the ship and the resulting noise (Braghin et al. 2011). At present, almost all classification societies have clear requirements for the vibration and noise of cruises. As the regulation goes, the noise level of all passenger compartments and public spaces shall comply with the requirements of IMO MSC 337 (91): *Ship Noise Level Rules* and ISO 6954 (2000): *Mechanical Vibration Guidelines*. For luxury cruises, the control of vibration and noise needs to be far stricter than normal ships in order to pursue higher comfort.

Structural Strength and Workload of Calculation

As an advanced passenger ship, a large luxury cruise requires high structural strength. The

requirements of the various specifications for cruise ship structures, given by some of the major western classification societies, namely, LR, RINA, BV, and DNV, are mainly concentrated on the calculation of structural strength, which involves the calculation of the longitudinal strength, the ultimate and remaining strength of the hull girder, the fatigue strength, and the strength of the entire ship. For the main supporting parts of the decks, the strength shall be directly calculated to check the tenacity. As for the ultimate and residual strength of the hull girders, besides the standard method, it can also be calculated by the nonlinear finite element method. In addition, the nonlinear finite element method can also be used to eliminate the noise of the ship (Fig. 2).

Structural Details

The luxury cruise should strive for excellence and each part requires exquisite work. Ship design workers in Japan and the West have attached great importance to the design and research of structural details. As a “mobile palace in the sea,” a large luxury cruise boasts a variety of sport, entertainment, dining, and leisure facilities. Every place and facility needs to be dedicated in design and manufacturing. Therefore, detailed design can also be a complex point for cruise design.

Design Specifications

The luxury cruise is not only a transportation vehicle but also a place for amusement. The

main functions of luxury cruise determine that the requirement is stricter than commercial ships in terms of safety, energy saving, and environmental protection.

Safe

Accidents of luxury cruises are generally caused by collision and cruise fire. Therefore, cruise ships must meet the requirements of *the International Convention for the Safety of Life at Sea* (SOLAS) and the requirements of *the Safe Return to Port* (SRtP). As SRtP notes, when a fire or flooding accident occurs, the ship should safely return to the port on its own power. During the process, passengers can safely stay within the “safety zone” and meet basic living needs. Even when the accident exceeds the safety zone, the ship should operate for at least 3 h to ensure orderly evacuation of passengers.

Eco-friendly

In order to create an eco-friendly marine environment, the International Maritime Organization (IMO) launched *Energy Efficiency Design Index (EEDI)* for controlling CO₂, sulfur oxide, and nitrogen oxide (SOX, NOX) emissions (Li et al. 2014). These rules will greatly affect the design of large luxury cruise.

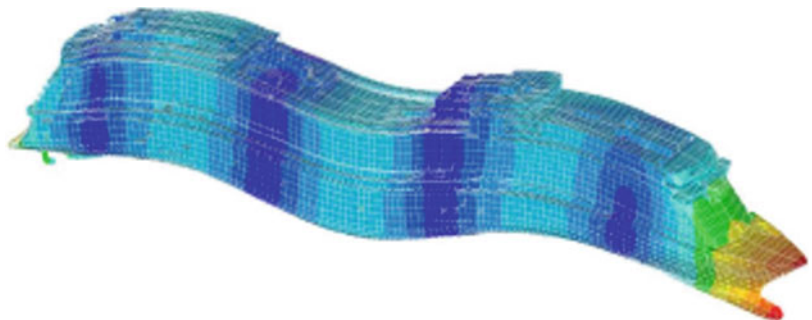
Other Rules

International Labour Convention

The latest *Maritime Labour Convention, 2006* issued by the International Maritime Organization puts forward higher requirements for sailor’s living conditions. For example, the room area for

Luxury Cruises,

Fig. 2 Overall of vibration characteristics (<http://image.so.com/>)



four sailors should be larger than 14.5 m². However, many of the cruises built in recent years cannot meet this requirement.

Other Regulations

Among the cruise associations, *Cruise Lines International Association (CLIA)* is one of the most influential in the industry. The association has established the self-regulation rules widely followed by its members in the area of waste disposal, restricted access to the driving cab, extra life jackets, heavy weight fastening on ships, and passenger prototype exercises (Fig. 3).

The World’s Major Cruise Shipyards

Aker Shipyard, Finland

Aker Shipyard was established in 1738, which has a history of more than 270 years. Remarkably, decades ago, *Aker Shipyard* used to spend 2 years building the famous luxury cruise *Marine Independence*, which made a stir in the whole cruise community and replaced its sister ship *Marine Liberation*, becoming “the largest luxury cruise” in the world at that time (Wang et al. 2005).

Meyer Shipyard, Germany

Meyer Shipyard is one of Germany’s largest and most modern shipyards. It was founded in 1795 and mainly built of wooden boat at first and started the construction of iron ships in 1874. Its first indoor dock was completed in 1987, 470 m long, 101.5 m wide, and 60 m deep. In 2000, the

second giant indoor dock was built, with a specification of 504 m in length, 125 m in width, and 75 m in depth. The shipyard can build three luxury cruises per year.

Fincantieri Shipyard, Italy

Fincantieri shipbuilding corporation in Italy has a long history of designing and building warships for nearly 200 years. The main products are passenger liners, ferries, and submarines, representing Italy’s advanced industrial technology and export around the world. In particular, its passenger ships and luxury cruises reign at the world’s No.1, accounting for 40% of the international market.

Key Applications

The World’s Leading Luxury Cruise Company (Group)

Carnival Corporation & plc

Carnival Corporation & plc (Zhou et al. 2010) is the world’s largest cruise company and has a total of 11 cruise brands, including *Carnival, Aida, Costa, Princess, Holland, Crown, Ibolo, P&O, P&O Australian, Saab, and Ocean Village Classes*. The fleet operates in Europe, the Caribbean, the Mediterranean, Mexico, and the Bahamas throughout the year. Seasonal routes include Alaska, Hawaii, the Panama Canal, the Canadian shipping lines, etc. Its fleet advantage lies in its various leisure facilities and innovative and spacious compartments.

Luxury Cruises,

Fig. 3 Design specifications (<http://www.baidu.com/>)

Passenger ships, Passenger Accommodation Frequency weighted r.m.s values in mm/s from 1 Hz to 80 Hz.			
Location	Comfort rating number (crn)		
	1	2	3
Passenger, top grade cabins	1.5	1.5	2.0
Passenger cabins, standard	1.5	2.0	3.0
Public spaces	1.5	2.0	3.0
Open deck recreation	2.0	2.7	3.5
For passenger ships with comfort rating, no single frequency component within the frequency range 6.3 Hz to 12.5 Hz shall exceed 1 mm/s r.m.s (weight).			

Royal Caribbean Cruises Ltd

Royal Caribbean Cruises Ltd. is the second largest cruise company in the world and owns six major brands (*Royal Caribbean International (RCI)*, *Fine Diamond Cruises*, *Freedom Cruises*, *Pullmantur Cruises*, *CDF Cruises*, and *TUI Cruises*), 41 luxury cruises, and 460 diverse sailing lines. It can visit nearly 300 tourist destinations, covering more than 70 countries and regions including the Caribbean, Alaska, Canada, Europe, the Middle East, Asia, Australia, and New Zealand.

Genting Group

Genting Group is a leading cruise corporation in Malaysia, which is composed of five major subsidiary companies: *Genting Malaysia Berhad*, *Genting Plantations Berhad*, *Genting Singapore PLC*, *Genting Hong Kong Limited*, and *Kien Huat Realty*. Established in September 1993, *Genting Hong Kong* runs the cruise travel business in Asia under the brand of *Star Cruises*. It is considered to be a pioneer in the Asian cruise industry and is committed to develop the Asia Pacific region as an important international cruise destination.

MSC Cruises

MSC Cruises is the leader of the cruise market around the Mediterranean Sea with ships throughout the region all year round. At the same time *MSC Cruises* offers seasonal cruise journey around the world including destinations such as the Nordic Atlantic Caribbean North America South America South Africa and other places. Possessing one of the world's most modern fleets which consists of 11 cruise ships the corporation has achieved the number of 1.3 million tourists annually.

Norwegian Cruise Line

Founded in 1966, *Norwegian Cruise Line* is headquartered in Miami-Dade County, Florida, and has become one of the most well-known brands in the North American cruise industry. Together with the newly joined fleets *Ocean Class* and *Regent Seven Seas Class*, the corporation has a total of 23 cruise ships operating throughout North America, Hawaii, the

Caribbean, Alaska, Europe, the Mediterranean, and Bermuda. The company's major shareholders are *Genting Hong Kong*, *Apollo Global Management*, and *TPG Capital*.

Famous Luxury Cruises in the World

Queen Mary 2

Queen Mary 2 (shown in Fig. 4) is an ultra-luxury ocean liner affiliated to *British Canada Cruise Line Company*. The liner costs more than 800 million US dollars and has a displacement of about 150,000 t. *QM2* can be regarded as a star in the world's cruise and is currently the largest ocean liner, with more than 1250 sailors at service. What is truly impressive is the ship size. She is 345 m long and 72 m tall, which is taller than *the Statue of Liberty* and larger than the largest aircraft carrier in the United States.

Queen Mary 2 is honored for many advanced technologies. A powerful marine mechanical propulsion system enables the sculpting of the giant hull, and advanced navigation and communication equipment can make sailing safer.

MS Voyager of the Seas

MS Voyager of the Seas (shown in Fig. 5) is the lead ship of the *voyager-class cruises* operated by *Royal Caribbean International (RCI)*, with a displacement of 137,000 t, able to carry 3840 passengers. She was the largest five-star luxury cruise in Asia, 311 m in length and 39 m above the waterline. What is worth mentioning, the ship possesses a *royal ballroom*, which is larger than the size of two football field, and a huge theater, with a capacity of 1350 people.

The most famous facility on *MS Voyager of the Seas* is the 120-m-wide deck. It is connected to two 11-story halls at each end side of the deck. Under the *Royal Avenue*, there are special facilities such as skating rinks, basketball courts, mini golf courses, and sea climbing walls.

MS Explorer of the Seas

MS Explorer of the Seas (shown in Fig. 6) is a *voyager-class* cruise ship owned and operated by *Royal Caribbean International*. With a displacement of 138,000 t and 1138 sailors at



Luxury Cruises, Fig. 4 Queen Mary 2 (https://en.wikipedia.org/wiki/Ocean_liner)



Luxury Cruises, Fig. 5 MS Voyager of the Seas (https://en.wikipedia.org/wiki/Ocean_liner)

service, she can accommodate over 3100 guests. The ship is fitted with six powerful *Wartsila* engines and three podded propulsors. The hull is 311 m in length, 38 m in width, and 47 m above the waterline, providing enough activity space for passengers. Passengers can enjoy

skating, board walking, and cliff climbing on the ship.

Sea Adventurer

Sea Adventurer (shown in Fig. 7) is a luxury cruise managed by the Royal Caribbean “Voyager Class,”



Luxury Cruises, Fig. 6 MS Explorer of the Seas (https://en.wikipedia.org/wiki/Ocean_liner)



Luxury Cruises, Fig. 7 Sea Adventurer (https://en.wikipedia.org/wiki/Ocean_liner)

with a displacement of 138,000 t and 1180 sailors at service. She has a length of 311 m, a width of 38 m, and a height of 47.5 m above the waterline, which owns 667 standard compartments.

Adventure of the Seas is best known for her spectacular three-story themed restaurant, under which there is a huge stadium that can

accommodate 1350 spectators. Besides a skating rink, climbing walls, and golf courses, the stadium also features a large gymnasium including basketball, bowling, and volleyball venues. In addition, *Adventure of the Seas* also boasts a wedding chapel, a hallmark of luxury cruise ships.

Navigator of the Seas

Navigator of the Seas (shown in Fig. 8) is one of the cruise liners of *Royal Caribbean International*. She measures about 138,000 GT and carries 3807 passengers plus 1180 sailors. Her length amounts to 311 m with a width of 38 m, and the height above the waterline is about 49 m.

Navigator of the Seas has a learning room equipped with 19 personal computers, the famous “Navigator’s Library” with a collection of thousands of books, a world-renowned caravan piano bar coupled with a nightclub, two swimming pools, as well as a cosmopolitan theater that can accommodate 1359 seats. Other service facilities include the *Cloud Nine* cocktail lounge, a jazz club, and a golf bar. The most considerate design is the large game room outside the library. Here, the popular games and game facilities make young passengers immersed in joy.

Chinese Market Distribution and Future Development Trend

While the ship market continues to be in the doldrums, the large cruise market is still active (Duman and Mattila 2005). Cruise tourism is currently on a large scale in Europe and America. There are 300–400 cruises a day with a large

number of tourists sailing in Alaska, Canada, the Caribbean, Mexico, Bahamas, Hawaii, Bermuda, South America, Panama Canal, Northern Europe and Russia, the Mediterranean, Greece, United Kingdom, and more than 100 countries and regions in Asia and Australia. The voyage spreads over 7 continents and travels more than 260 ports around the world.

With the rapid growth of the cruise economy, the market potential of luxury cruise ships in emerging countries such as China has attracted the attention of global cruise giants. According to the analysis of *2016 China Cruise Industry Development Status and Prospects*, the demand of the cruise market in China is growing rapidly, for its rapid economic growth and large population (Sun et al. 2014). The market scale continues to expand, and the number of cruise ship visitors has experienced an explosive annual growth of over 40% in 2016 (Wang et al. 2018). Considering that the penetration rate of China’s luxury cruise tourism is still less than 0.2%, and the impact of future consumption upgrades and policy promotions, the huge development potential of China’s cruise industry can be optimistic in the future. It is expected that the Chinese cruise market will maintain a growth rate of over 30% in the next 5 years and will have more than nine million cruise visitors by 2020. It can be said that a cruise feast that



Luxury Cruises, Fig. 8 Navigator of the Seas (https://en.wikipedia.org/wiki/Ocean_liner)

focuses on Chinese market will kick off in the near future.

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Luxury Passenger Liner

► [Luxury Cruises](#)

M

Machine Learning

- [Big Data-Based Decision Support Systems](#)

Machinery Maintenance

- [New Technologies in Auxiliary Propulsions](#)

Mach-Zehnder Interferometer (MZI)

- [Fiber Optic Hydrophone](#)

Magnetic Compass (MCP)

- [Integrated Navigation](#)

Magnetohydrodynamics (MHD)

- [AUV/ROV/HOV Propulsion System](#)

Maneuverability of Polar Vessel

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Synonyms

[Discrete element method \(DEM\)](#); [Dynamic positioning \(DP\)](#); [National Maritime Research Institute \(NMRI\)](#); [Planar Motion Mechanism \(PMM\)](#); [The Institute for Ocean Technology \(IOT\)](#)

Definition

Maneuverability of polar vessel refers to the performance that the polar vessel can keep or change its motion state according to the intention of the pilot. It mainly involves the ability of polar ships to maintain or change speed, course, and position.

Introduction

The key principle for ships' successful ice navigation is to keep freedom of maneuver. Once a

Baoshan Wu: deceased.

ship is trapped, the ship goes wherever the ice goes. Ship maneuvering in ice-covered waters is far more complex than that in open water, given the reason that the ice-hull interaction is stochastic in nature, resulting in irregular icebreaking pattern with high uncertainty. To better understand polar ships' maneuverability, the maneuvering tests, maneuvering equations, maneuvering performance, and DP in ice are presented and further discussed.

Nomenclature

L	Ship's waterline length (L_{WL}) [m]	N_v	Derivative of yaw moment due to sway velocity [kg-m/s]
B	Ship's beam [m]	$Y_{\ddot{v}}$	Derivative of sway force due to sway acceleration [kg]
m	Mass [kg]	$N_{\ddot{v}}$	Derivative of yaw moment due to sway acceleration [kg-m]
C_b	Block coefficient	Y_{δ}	Derivative of sway force due to rudder angle [kg-m/s ²]
I_z	Yaw moment of inertia [kg-m ²]	N_{δ}	Derivative of yaw moment due to rudder angle [kg-m ² /s ²]
R_b	Icebreaking resistance [N]	Y_r	Derivative of sway force due to yaw rate [kg-m/s]
σ_f	Ice flexure strength [Pa]	N_r	Derivative of yaw moment due to yaw rate [kg-m ² /s]
H_i	Ice thickness [m]	$Y_{\dot{r}}$	Derivative of sway force due to yaw acceleration [kg-m]
l_{cr}	Crack length [m]	$N_{\dot{r}}$	Derivative of yaw moment due to yaw acceleration [kg-m ²]
D	Turning diameter [m]		
R	turning radius [m]		
R/L	Non-dimensional turning radius		
V	Forward speed [m/s]		
v	Lateral speed [m/s]		
$\dot{v}$	Lateral acceleration [m/s ²]		
R	Turning radius [m]		
r	Yaw rate [rad/s]		
$\dot{r}$	Yaw acceleration [rad/s ²]		
t	Time [s]		
x_G	Position of center of gravity [m]		
Y_0	Sway amplitude [m]		
K	Roll moment [N-m]		
N	Yaw moment [N-m]		
X	Surge Force [N]		
Y	Sway Force [N]		
T	Draft [m]		
x	Forward position from global origin [m]		
y	Lateral position from global origin [m]		
ϕ	Yaw angle [degrees]		
β	Drift angle [degrees]		
δ	Rudder angle [degrees]		
λ	Scale factor		
Y_v	Derivative of sway force due to sway velocity [kg/s]		

Maneuvering in Ice

The extreme environment in Polar Regions poses great challenges to ships' navigation, and ice is the primary factor that affects ship's maneuvering performance, especially to icebreakers. The typical circumstances ships would encounter in ice-covered waters are listed as follows. When the average ice thickness exceeds 50 cm, icebreakers cannot go forward with full speed due to high resistance from ice, and the channel behind icebreakers freezes quickly afterwards, in which case the ship is likely to be trapped. Besides, the shoulder of ship is easily wrecked either in turning or advancing owing to the inhomogeneity of ice. And if a ship encounters a crack, the bow is forced to slide along the edge of crack. This may lead to rudder out of control. When ice-ship interaction is acute, the shipborne equipment would likely to be damaged, and the motion of rudder is encumbered by ice, leading to difficulty of bow turning. Poor maneuverability is one of the most important reasons. Therefore, successful ice navigation calls for good maneuvering performance.

A ship's capability to maneuver in ice-covered waters depends on its available steering forces and its transverse/rotational resistance. Sufficient steering force and propulsive power are key

factors to overcome the increased turning moment generated by the ice force along the broadside during inception of a turn and to reach a steady turning rate.

Steering forces are realized through lift generated by water flow over the rudder(s), differential propeller shaft thrust, directional thrust capability (such as podded-propellers and adjustable propeller nozzle direction), bow thrusters, and lift generated from ship heel angle. A ship's transverse and rotational resistances are directly influenced by its L/B ratio, block coefficient, shape of the icebreaking waterline, and slope of the hull (or "icebreaking angle") along the icebreaking waterline, shape of its underwater hull, and its pivot point.

According to size, age, thickness, and concentration, ice can be classified into four primary types: broken ice, sheet ice, ridged ice, and icebergs (Aboulazm and Muggeridge 1989). Sheet ice, also called level uniform homogenous ice, is the most common and severe ice condition that ships (icebreakers and ice going ships) navigating in polar regions would encounter. Majority of full-scale maneuvering tests were carried out in sheet ice. Events associated with a ship maneuvering in level uniform homogenous ice can be categorized by ice thickness. For very thin ice, a ship's turning radius will normally be very near that of its open water turning radius. This is largely attributed to the icebreaking action of the bow wave. As ice thickness increases, the effect of the bow wave diminishes and the ice sheet begins to interact with the ship's hull. Ice breaking occurs mostly at the bow. The broken channel is wider than the ship's beam so that the stern can move out and point the bow into the turn. The sides and aft of ship also break ice. As ice thickness increases, the ship's ability to clear away broken ice from between its sides and the ice channel edges diminishes. A limiting ice thickness is approached where turning resistance is too high and the ship is no longer able to making turns at all. Quantitative values for ice thicknesses were not provided in the description because thickness alone does not dominate the ship-ice interactions. Ice strength, both in flexural and crushing, and hull-ice friction are also major influential factors. Besides, the authors also take into account the changes of broken channel edges to research the

phenomenon of icebreaking in aft shoulder of ships in level ice (a simplified model of sheet ice in model test and numerical simulation).

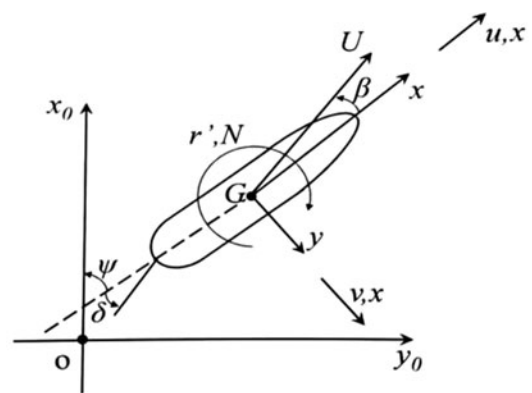
For other conditions, ice distribution (pack or rubble ice, ridges, etc.), ice pressure (e.g., due to wind), snow cover, surface current, and water depth also directly impact a ship's maneuvering performance. Pack or rubble ice is usually a common condition for ice going ships and ice-strengthened ships (Riska 2010). Turning and zig-zag are two main maneuvers in pack ice according to the real navigation. And this was simulated by Zhan et al. adopting DEM method (Zhan and Molyneux 2012).

Equation of Motion in Ice

Coordinate Systems

In ship maneuvering simulations, the ship can be regarded as a rigid body. The co-ordinate systems (Fig. 1) and equations of motion of ship maneuvering in ice are the same as that in open water. And the surge force, sway force, and yaw moment are the three major global forces to determine the ship's maneuvering motions (Zhou et al. 2016).

$$\begin{cases} m(\dot{u} - rv) = X \\ m(\dot{v} + ru) = Y \\ I_z \dot{r} = N \end{cases} \quad (1)$$



Maneuverability of Polar Vessel, Fig. 1 Coordinate systems

External Forces

The external forces include forces from the hull, the thruster, the rudder, and other appendages. During ship's ice navigation, forces on the hull mainly consist of ice forces, hydrodynamic forces, current forces, and wind forces. See Fig. 2.

The forces from appendages and other environmental factors like wind are usually ignored. Until now, most researchers divide external forces into several parts, which is (Lubbad and Løset 2011):

$$\begin{cases} X = X_H + X_P + X_R + X_{Ice} \\ Y = Y_H + Y_P + Y_R + Y_{Ice} \\ N = N_H + N_P + N_R + N_{Ice} \end{cases} \quad (2)$$

H, P, R, and ice represent forces and moment generated from hydrodynamic force, propeller force, rudder force, and ice load, respectively.

Ice Force

The basic premise to get ice force is to understand the mechanics of ice-ship interaction. Two categories, analytical approach and numerical approach, are used to classify the existing models of ice-ship interaction process. In terms of analytical approach, the methodology in which the total

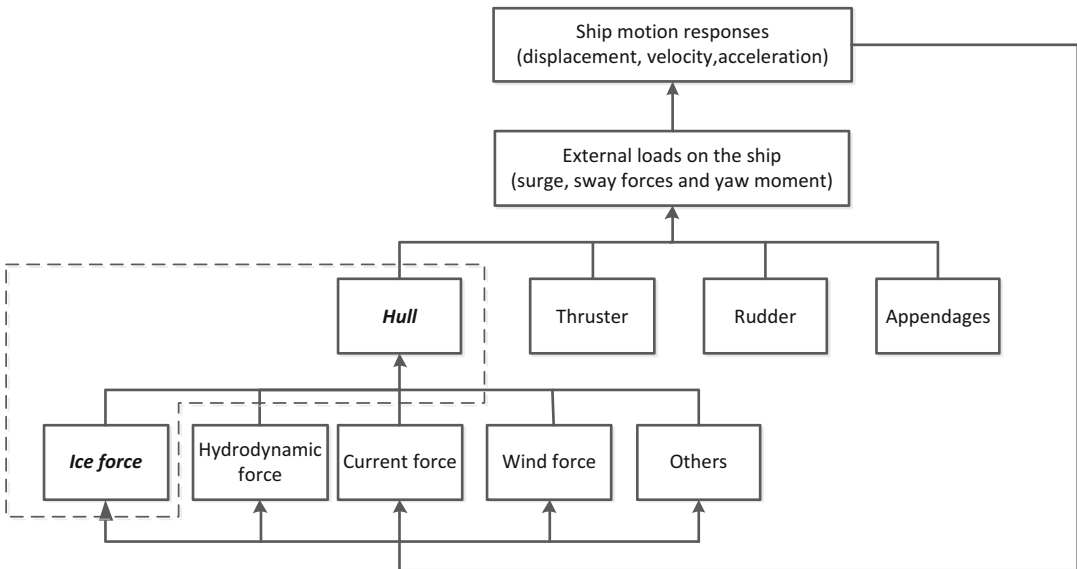
ice force is divided into several independent force components that represent the corresponding physical processes during the continuous ship maneuvering in level ice has been widely used. The global ice force is equal to the linear sum of the three force components, that is, the breaking, buoyancy, and clearing force components. It can be written as:

$$\begin{cases} X_{ice} = X_{br} + X_{cl} + X_{bouy} \\ Y_{ice} = Y_{br} + Y_{cl} + Y_{bouy} \\ N_{ice} = N_{br} + N_{cl} + N_{bouy} \end{cases} \quad (3)$$

The ice breaking force component represents the contributions from the ice breaking process which is stochastic in nature resulting in irregular icebreaking pattern with high degree of uncertainty in predictability during ship maneuvers. To get ice force by numerical approach is mainly depends on its numerical model.

Overview of Ship Maneuvering Prediction in Ice

To predict ship maneuvering performance in ice, a model applying the analytical approach with



Maneuverability of Polar Vessel, Fig. 2 Ice-ship interaction in maneuvering (Liu et al. 2007)

numerical implementation for simulating various ship maneuvers in level ice is established. The model is used to simulate real-time maneuvering and support maneuvering simulator for its efficiency. Also, numerical methods have witnessed great advancement with the development of the computer technology. They can be used to solve analytical equations or to simulate the continuous ship-ice interaction process in various ice conditions (Liu et al. 2007).

Analytical Approach

The analytical approach is subdivided into empirical, semi-empirical models.

With a large number of ship model tests and full scale ship trials having been carried out, experimental data (e.g., coefficients) can be obtained through sorting and fitting test results. Majid and Menon et al. (1983) built a time domain model for ship maneuvers in ice based on conventional maneuvering equations. The Taylor's series expressions with coefficients obtained from model tests were adopted to estimate the ice forces on the hull due to the ship motion in the level ice. Williams et al. (1996) proposed an ice-hull force model for ship maneuvers in level ice from a series of PMM model tests. Robert et al. (2002) conducted a series of resistance and maneuvering tests using the R-class icebreaker model (scale ratio 1:40) and MV Arctic Bulk Carrier model (scale ratio 1:80), in which the damping coefficient and the acceleration control coefficient of the roll were calculated. Lau et al. (2004) discussed the effect of breaking channel width on ship maneuverability from PMM tests. And the effect of external moment on ship-ice interaction was also studied. Li et al. (2013) conducted a series of model tests on icebreaker in ice basin. The change of model speed and heading angle in different ice thickness were analyzed. Phenomena such as bending broken on ice and ice crushing on hull surface were observed.

The analytical models applying semi-empirical approaches take into account more details of the pertinent ship-ice interaction processes. Tue-Fee et al. (1987) proposed a semi-empirical model for predicting ship's steady turning maneuvers in level, unbroken, homogeneous ice fields. The

formulas for calculating sway force and yaw moments caused by pure ice forces are theoretically derived from basic assumptions. Aboulazm (1993) theoretically analyzed the turning performance of the ship in the block ice condition, and the external force of the ship in steady turning was calculated.

Numerical Approach

Discrete element method (DEM), which divides the continuous ice materials into many small elements, is widely applied in ice mechanics issues. Based on DEM, Lau (2006) simulated advancing and turning motion of TERRY FOX numerically using commercial discrete element code software. The ship was regarded as rigid body, and both level ice and ship model were established in discrete elements. The numerical results were compared with PMM model test results, which agreed very well. The shortcoming is that ice model was greatly simplified, and the failure of old discrete elements and the generation of new discrete units were not considered. Sawamura (2010) also calculated ship maneuvering in level ice using DEM model. The level ice in waterline surface was divided into a number of small circles, and the disappearance of circles indicated the breaking of the ice. This method is a good way to observe the maneuvering trajectory, yet it only paid attention on the ice force near the waterline, ignoring the friction of the broken ice along the hull. Many other researchers have also contributed greatly to this problem, such as Zhan et al. (2010), Zhan and Molyneux 2012, Derradji (2003), Valanto (2001).

Another common numerical model was polygon model. This method is a 2D numerical solution, in which the broken channel edges and the ice breaking in ships' aft shoulder during turning are observed in detail. This method can be found in Martio (2007), Su et al. (2010), and Zhou (2016).

Maneuvering Tests in Ice

Maneuvering test is a collective name for all those tests in which the rudder is turned or the turning forces and moments are obtained by other means,

for example, azimuthing thrusters, tunnel thrusters in bow, different rpms at twin propellers. The primary purpose of maneuvering tests is to investigate the effectiveness of the hull in making turns. The other considerable purpose is to investigate the effectiveness of the turning devices like rudders, thrusters, or azimuthing propulsors. Maneuvering tests are also applied to investigate the flow of ice floes around the hull in various maneuvers. The design of aft shoulders down to baseline is especially important for turning capability. Quinton et al. (2005) had collected test data of all publicly available full-scale and model-scale ship-ice maneuvering data before 2005.

Full-Scale Ice Maneuvering

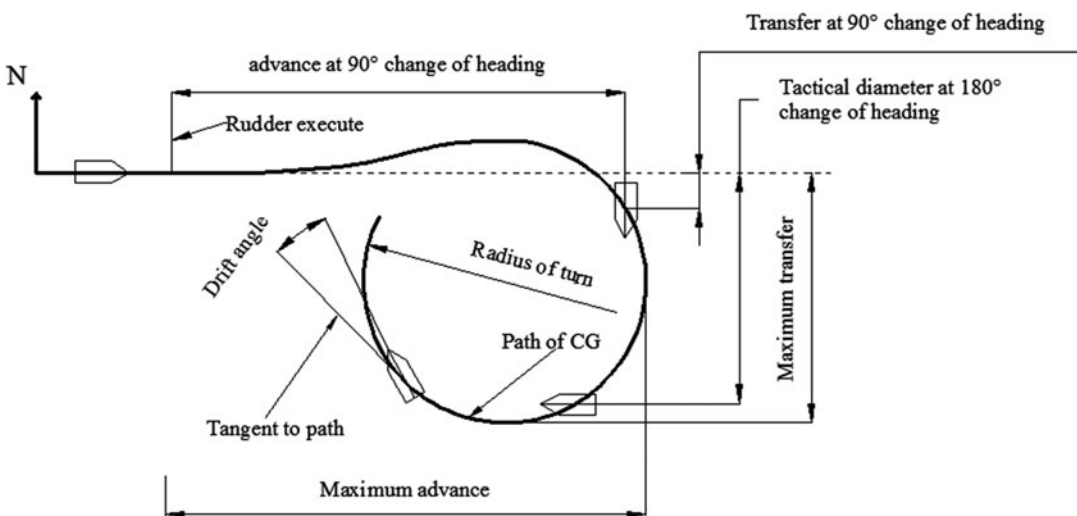
In terms of full-scale ice maneuvering, common maneuvers conducted by icebreaking ships are: (1) ahead progress, (2) channel breakout, (3) turning circle, (4) captain's turn, (5) modified captain's turn, (6) Kempf maneuver (or zig-zag), (7) breaking out a beset ship, (8) close coupled escort, and (9) ridge ramming. The turning circle and the Kempf maneuver are the most common of all in a "maneuvering in ice" performance evaluation.

The purpose of the turning circle test is to find out how much area is needed to turn the ship. In practice, the circle may not be a perfect circle, but

a spiral. The progress of this test is just like that in open water (Fig. 3), which involves four phases. However, the detail of external force acting on the hull is different because of the ice. In open water, steady turning radius R is the simplest measure of a ship's maneuverability, while the situation is different in ice condition. Because variations in ice thickness and strength also affect the ship's maneuverability, R/L is not as reliable a representative of "maneuvering performance in ice" as it is for open water. Yet, most researchers consider this parameter as a standard. Generally, the smaller the R/L values for a ship, the greater its ability to maneuver while keeping continuous motion. Some modern icebreaker designs are able to turn the ship in level ice so effectively, that the diameter of the broken area in ice is almost the same as the waterline length of the ship.

The turning ability is not the only measure for the maneuvering performance of an ice-going ship. It is important to perform different maneuvers (e.g., zigzag and star maneuvers) in shortest time possible. A good maneuvering performance can be achieved by a proper hull form design, having a large turning rate of the rudder(s) or azimuthing thruster(s) and providing a large transverse force.

Breaking out of channel test is mainly conducted by merchant ships. Those ships



Maneuverability of Polar Vessel, Fig. 3 Turning circle schematic (ITTC Dictionary of Hydromechanics)

navigate mostly in channels made by icebreakers. Therefore, the turning circle is not a relevant description of the agility of the ship but the ability of the ship to turn out from a channel. The test can be performed from zero speed or some other specified speed. The channel width and local pattern of ice floes and shapes of the channel edges (before the test) have big influence on the results. If this test is performed from zero speed in the channel made by the test in presawn ice, the result may differ from that in naturally broken channel.

A lot of full-scale maneuvers in ice were conducted, for example, Max Waldec, Coast Guard 140 Foot Icebreaker, Canmar Kigoriak, Robert LeMeur, USCGC Polar Star, USCGC Healy, AHTS/IB Tor Viking II. The database of tests can be found in Quinton et al. (2005).

Model-Scale Ice Maneuvering

Ice maneuver tests in model scale can be grouped into two categories, that is, free running model test and captive model test.

The free maneuver tests involve the use of a remotely operated model that is unrestricted and capable of self-propulsion and steering. An operator directs the model via remote control and a model tracking system recording its motions. Two maneuvers in particular are used to test the maneuvering performance of free running models in ice: turning circles and Kempf maneuvers. The general process is to place the model on one side of the ice sheet, and then setting the power. The rudder angle will be given until the ship model runs straightly and steadily, and the turning diameter can be measured by running out of the 1/4 turn path. The self-propelled model test can directly observe the maneuverability of the ship model in different ice condition, but the scale of ship model is usually limited by the scale of the ice tank. And it costs a lot. A free maneuver test conducted by NMRI (National Maritime Research Institute) in Japan (Izumiyama et al. 2005) found the difference of ice load distributions in the turning-going. In the turning mode, significant ice loading occurs in the aft-body in the outside of the turn, while the inside hull receives very little load.

Captive maneuver tests involve the use of some form of Planar Motion Mechanism (PMM)

or a combination of devices that cause the same motion control (e.g., a rotating arm in combination with other devices). The purpose of PMM test is to obtain a series of hydrodynamic and derivatives which govern the maneuvering performance of a ship. Six standard PMM tests, including static rudder, yaw sweep, pure sway, pure yaw, and pure yaw with drift, are carried out to get forces and moment of the model. Ship maneuvering characteristics can be predicted afterwards. The Institute for Ocean Technology (IOT) in Canada has performed a series of model tests using its PMM. A PMM test (Lau 2007) was conducted by the institute in manifold conditions, and the process of PMM test was described in detail.

DP in Ice

With an increasing interest of operations in Arctic and subarctic areas, the station-keeping capabilities in ice have attracted wide attention. Vessels operating in various ice conditions require suitable dynamic positioning (DP) systems to help course-keeping or position-keeping. DP is widely used in ice-free waters and open seas where the impacts from wind, waves, and currents are involved in the force balance algorithms.

DP in ice conditions infers additional parameters such as the local and global ice loads on hull and the performance variations of propulsion systems. To understand and predict the ice forces acting on a DP ship, a wide range of numerical simulation studies and development projects have been initiated throughout the years. The numerical simulations focus on a wide range of applications from hull and propulsion designs, DP capability limits, ice management, and oil spill response capabilities evaluation to operational training for navigational officers.

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Maneuvering Tests

► Towing Tank Test

Manned Submersible (MS)

- Atmospheric Diving Suit (ADS)
- Human Occupied Vehicle (HOV)
- Submersible

Marine Controlled-Source Electromagnetic Method (MCSEM)

► Deep Tow System

Marine Ecological Red Line

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Synonyms

Ecological sensitive areas; Ecological vulnerable areas

Definition

The marine ecological red line is a marine spatial planning tool, referring to the institutional arrangement of important marine ecological functional areas, ecological sensitive areas, and ecological vulnerable areas as key control areas which implement strict classified management and control measures in order to maintain marine ecological health and security.

Scientific Fundamentals

The Origin and Development of the Red Line System

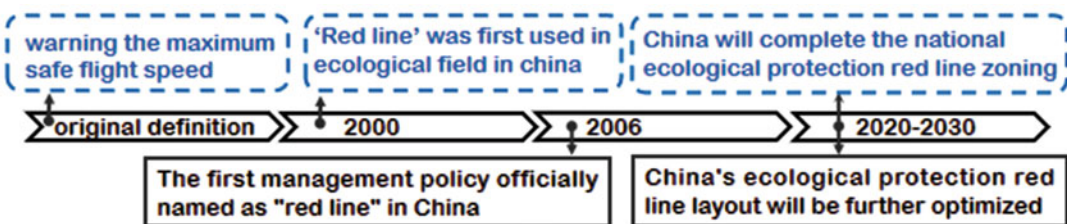
The red line was originally defined as a red sign warning the maximum safe flight speed or a red line marking other hazardous bases. “Red line”

as a spatial planning tool was first used in China by a Chinese scholar Dr. Gao Jixi when he was doing ecological planning in Anji County, Zhejiang Province, in 2000 (Gao 2014). In 2006, the State Council formulated the cultivated land red line of 1.8 billion mu (1 mu = 666.6666667 square meters), which was the first management policy officially named as “red line” in China. Subsequently, the concept of red line was gradually introduced into the field of resources and environment, and related concepts such as water resources red line and forestry red line were derived. The connotation of red line also expanded from spatial constraints to quantitative constraints (Rao et al. 2012). By the end of 2020, China completed the national ecological protection red line delineation; by 2030, China’s ecological protection red line layout will be further optimized (Fig. 1). So far, the ecological red line system has risen to the height of national strategy and has become a new system to guide China’s social and economic development. It is also an innovative measure in regional ecological protection and management in China.

Principles of Marine Ecological Red Line Zoning

There are five key principles in marine ecological red line zoning.

1. Regional conjugation. Any marine ecological red line area must be a relatively complete, scarce (unique) natural geographical unit, and its scope should include the key areas to maintain the integrity and connectivity of the



Marine Ecological Red Line, Fig. 1 Timeline of red line policy development in China

- ecosystem, so as to ensure the flow and transmission of ecosystem material, energy, and information.
2. Ecological orientation. The scientific knowledge of ecosystem and its dynamics should be used to guide the division of marine ecological red line, with emphasis on the specific ecosystem and the influence range of various activities.
 3. Integration of land and sea development. It is necessary to fully consider the internal relationships between ocean and land resources, natural environment and ecology, correctly handling the links between ocean and land ecological protection, as well as formulating strategic thinking of the integrated development of sea and land.
 4. Coordination principle. Marine ecological red line zoning should be connected with land ecological red line zoning as far as possible, coordinating with the boundaries of various marine ecological protection areas, adapting to the needs of economic and social development as well as the local regulation ability, and reserving appropriate future development space and environmental capacity space.
 5. Dynamic principle. Marine ecological red line zoning should be forward-looking, and the area should only be increased but not be decreased. Whenever the boundary and threshold of the ecological red line zone has changed due to the disturbances from the external environment, it should be adjusted in time to ensure the continuity of its basic ecological process and function.

Theoretical Basis of Marine Ecological Red Line

Ecological succession (Wikipedia 2021a) is the process of change in the species structure of an ecological community over time. The time scale can be decades (e.g., after a wildfire), or even millions of years after a mass extinction (Sahney and Benton Michael 2008). It is a phenomenon or process by which an ecological community undergoes more or less orderly and predictable changes following a disturbance or the initial colonization of a new habitat. Succession may be

initiated either by formation of new, unoccupied habitat, such as from a lava flow or a severe landslide, or by some form of disturbance of a community, such as from a fire, severe windthrow, or logging. Succession that begins in new habitats, uninfluenced by preexisting communities, is called primary succession, whereas succession that follows disruption of a preexisting community is called secondary succession. Succession is among the first theories advanced in ecology, and the study of succession remains at the core of ecological science.

A coupled human-environment system (Wikipedia 2021b) (known also as a coupled human and natural system, or CHANS) is an integrated scientific framework for studying the interface and reciprocal interactions that link human (e.g., economic, social) to natural (e.g., hydrologic, atmospheric, biological) sub-systems of the planet (NSF, Marina et al. 2011). The phrase “coupled human-environment systems” appears in the earlier literature (dating back to 1999) noting that social and natural systems are inseparable (Sheppard and McMaster 2004, NRC 1999). Research into CHANS builds on the disciplines of human ecology, ecological anthropology, environmental geography, economics, and other eco-bio-geo-physical fields. Moving beyond some of the traditional research methods in social and natural sciences, CHANS tackles broader investigations into the complex nature of reciprocating interactions and feedbacks between humans on the environment and the effect of the environment on humans.

The theory of human-sea relations. In a narrow sense, it means that human society, in order to survive and develop, constantly expands and deepens the transformation and utilization of the marine environment (Kappel et al. 2012), enhances the ability to adapt to the marine environment, and changes the appearance of the marine environment; in turn, the marine environment also affects human activities, resulting in sea area differences. There is a unity-contradiction between them.

Sustainable development (Wikipedia 2021c). Economic development ensures that environmental depletion and degradation do not occur. In the

late 1980s, the term came to have a wider application as concern with environmental issues grew. Criteria of sustainable development came to be applied to all forms of economic development. Hitherto those economic developments had ignored environmental effects that could undermine those developments. So sustainable development would be seen as economic and social development that could avoid pollution, conserve nonrenewable energy forms, and in general would not cause future problems for which there were no easily available solutions.

The carrying capacity (Wikipedia 2021d) of a biological species in an environment is the maximum population size of the species that the environment can sustain indefinitely, given the food, habitat, water, and other necessities available in the environment. In population biology, carrying capacity is defined as the environmental maximal load (Hui 2006), which is different from the concept of population equilibrium. The carrying capacity is the number of individuals an environment can support without significant negative impacts to the given organism and its environment. Below carrying capacity, populations typically increase, while above, they typically decrease. A factor that keeps population size at equilibrium is known as a regulating factor. Population size decreases above carrying capacity due to a range of factors depending on the species concerned but can include insufficient space, food supply, or sunlight. The carrying capacity of an environment may vary for different species may change over time due to a variety of factors, including food availability, water supply, environmental conditions, and living space.

The balance of nature (Wikipedia 2021e) is a theory that proposes that ecological systems are usually in a stable equilibrium (homeostasis), meaning that a small change in some particular parameter (e.g., the size of a particular population) will be corrected by some negative feedback that will bring the parameter back to its original “point of balance” with the rest of the system. It may apply where populations depend on each other, for example, in predator/prey systems or relationships between herbivores and

their food source. It is also sometimes applied to the relationship between the Earth’s ecosystem, the composition of the atmosphere, and the world’s weather.

Theoretical paradigm of delimitation of marine ecological red line. The delimitation of marine ecological red line should be based on the ecological integrity of marine resources and environment, and natural interference and human interference should be regarded as external interference. The delimitation of marine ecological red line should be based on finding the threshold between disturbance pressure and ecological integrity in the process of system development and formulate the threshold value of spatial resources, environmental elements, and ecological protection.

Key Applications

National Marine Red Line Policy in China

Review and Comparison of National Policies

In recent years, the state has paid more and more attention to the delimitation, management, and supervision of marine ecological red line and promulgated many important policies. In 2018, the State Oceanic Administration (SOA) promulgated the “measures for supervision and administration of the red line of marine ecological protection (Draft for comments),” pointing out that the implementation of the national marine ecological protection red line should be uniformly guided, coordinated, and supervised by the State Oceanic Administration. The provincial marine administrative departments shall, in accordance with the law, organize and implement the management and control of marine ecological protection red line areas, marine ecological environment monitoring, and ecological protection and remediation. This method mainly makes detailed requirements for sea-related review, assessment and supervision, regular evaluation, monitoring, and so on. In 2019, the Ministry of Ecological Environment issued the “measures for the management of ecological protection red line (Trial)” (Draft for comments). It pointed out that local governments at all

levels are responsible for delimiting and strictly abiding by the ecological protection red line, and the department in charge of ecological environment under the State Council, together with relevant departments, shall formulate policies and standards for delimiting and strictly observing the ecological protection red line. The departments in charge of ecological environment at all levels shall exercise unified supervision over the red line of ecological protection in their respective administrative areas and carry out unified monitoring and evaluation, supervision and law enforcement, and supervision and accountability. Relevant departments in charge of development and reform, finance, natural resources, water conservancy, agriculture and rural areas, forestry, and grassland at all levels shall, in accordance with their respective responsibilities, do a good job in protecting and managing the red line of ecological protection. The measures mainly regulate and restrict the delimitation and adjustment of red line, human activity control, protection and restoration, and ecological compensation. In 2020, the Ministry of Natural Resources issued the “ecological protection red line management measures (Trial)” (Draft for comments). The measures pointed out that the department in charge of natural resources under the State Council, together with relevant departments, should formulate and improve policies for delimiting and managing ecological protection red lines, establish and improve relevant technical standards and regulatory systems, and guide the delimitation and management of ecological protection red lines in all provinces (regions and cities). According to the land and space planning, the local natural resources departments at all levels shall establish a red line coordination mechanism for ecological protection and uniformly carry out the work of use control, monitoring and evaluation, supervision and law enforcement, and assessment and evaluation. The measures have detailed regulations on the red line delineation and adjustment, limited human activity control, supervision, and implementation, covering the main aspects of the two measures issued in 18 and 19 years, with relatively complete and comprehensive contents.

Detailed Introduction of National Policies

Measures for Supervision and Administration of the Red Line of Marine Ecological Protection (Draft for comments) in 2018. The “red line of marine ecological protection” determined the boundary line of geographical area and the control line of relevant management indicators of important marine ecological functional areas, marine ecological sensitive areas, and marine ecological vulnerable areas. The red line of marine ecological protection is the bottom line of maintaining marine ecological function and marine environmental quality, which cannot be adjusted in principle. After the delimitation of the red line of marine ecological protection, the red line area, the retention rate of the natural coastline of the mainland, the retention rate of the natural shoreline of the island, the sandy coastline of the island, and the quality of the sea water can only be increased, but not reduced. In terms of monitoring and inspection, it is stipulated that the implementation of the red line of marine ecological protection should be included in the content of national marine supervision. The provincial marine administrative department shall establish a dynamic monitoring platform for the marine ecological protection red line in the administrative area to monitor the development and utilization activities and ecological environment status of the islands in the marine ecological protection red line area and the surrounding sea areas and report the annual monitoring results to the marine regional Branch Bureau before the end of December of each year. The State Oceanic Administration carries out assessment on the implementation of marine ecological protection red line of coastal provincial government. The assessment contents mainly include the implementation of marine ecological protection red line control indicators, implementation of control measures, and protection and restoration of marine ecological protection red line areas. In terms of public supervision, any unit or individual has the right to report to the marine administrative departments at all levels any behavior that violates the red line system of marine ecological protection, and the marine administrative department receiving the report shall deal with it according to law and regulations.

Measures for the management of ecological protection red line (Trial) in 2019 (Draft for comments). It is stated that areas with special important ecological functions within the scope of ecological space must be strictly protected. It is the bottom line and lifeline to ensure and maintain national ecological security. It usually includes important ecological function areas with important water conservation, biodiversity maintenance, soil and water conservation, wind prevention and sand fixation, coastal ecological stability, soil erosion, land desertification, rocky desertification, salinization, and other sensitive and fragile areas of ecological environment. In order to delimit and strictly observe the ecological protection red line, we need to establish the ecological protection red line system, ensure the national ecological security, and build a beautiful China, and these measures are formulated in accordance with the requirements of several opinions on delimiting and strictly observing the ecological protection red line and relevant laws and regulations. In principle, the ecological protection red line shall be managed according to the requirements of prohibited development areas. This method follows the principles of ecological priority, strict control, and equal emphasis on rewards and punishments. All kinds of development activities that do not conform to the main function orientation are strictly prohibited. According to the orientation of dominant ecological function, differential management should be implemented to ensure that the ecological function, area, and nature of ecological protection red line will not be reduced. In terms of human activity control, urbanization and industrialization activities are prohibited within the ecological protection red line, and all kinds of development activities that do not conform to the main function orientation are strictly prohibited. The measures list specific items of prohibited activities and permitted activities and formulate approval rules for human activities or construction projects within the ecological protection red line. The management of existing human activities and construction projects within the red line of ecological protection should follow the principles of respecting history, seeking truth from facts, handling according to

law, and gradually solving problems and strictly investigate and deal with illegal construction projects. In addition, the measures also require all localities to formulate ecological protection red line protection and restoration plans, give priority to the protection of good ecosystems and important species habitats, repair damaged ecosystems, build ecological corridors and important ecological nodes, and improve the integrity and connectivity of the ecosystem. In accordance with the principle of overall planning and comprehensive management of land and sea, it shall carry out ecological renovation and restoration of the ecological protection red line of marine land and space and focus on strengthening the comprehensive renovation of estuaries, coastal zones, islands, and polluted sea areas within the ecological protection red line.

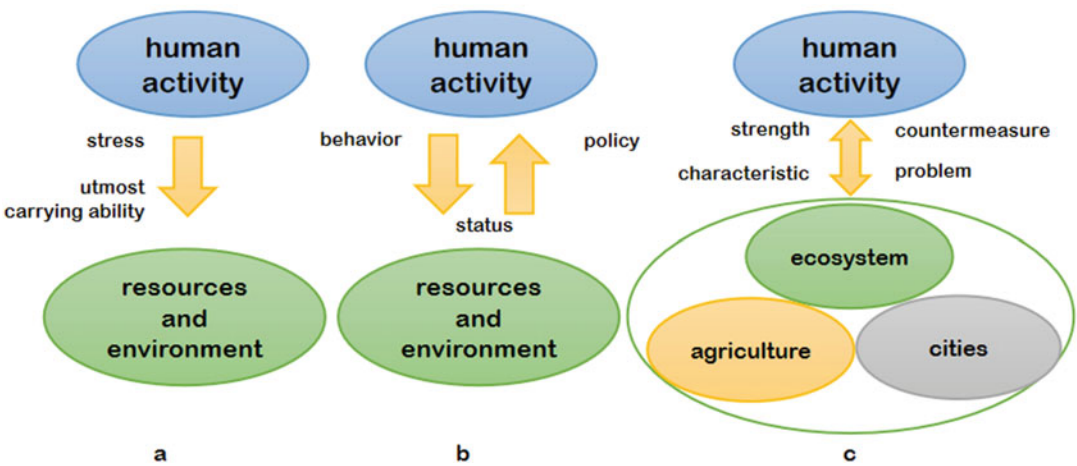
Ecological protection red line management measures (Trial) in 2020 (Draft for comments). They are the areas with special important ecological functions in the land and marine ecological space which must be strictly protected under compulsion. These areas include water conservation, biodiversity maintenance, soil and water conservation, windbreak and sand fixation, coastal protection and other ecological functions of extremely important areas, soil erosion, land desertification, rocky desertification, coastal erosion and sand source loss and other ecologically vulnerable areas, as well as other areas with potential important ecological value which cannot be determined at present. In order to formulate and strictly observe the ecological protection red line, establish the ecological protection red line system, ensure the national ecological security, and build a beautiful China, these measures are formulated in accordance with the requirements of “several opinions on delimiting and strictly observing the ecological protection red line” and relevant regulations. According to the national and regional ecological security pattern and on the basis of scientific assessment, all provinces (districts and cities) shall identify areas with extremely important ecological functions and extremely fragile ecology, and other areas with potential important ecological value that cannot be determined by assessment at present, and shall be included in

the red line of ecological protection, so as to make sure that they should be fully delineated and protected. The demarcation of ecological protection red line should coordinate the contradictions and conflicts with permanent basic farmland, urban development boundary, and existing land space development and utilization activities, so as to ensure that the three control lines do not cross and overlap. In order to maintain the continuity and integrity of the ecosystem, scattered cultivated land, garden land, artificial commercial forest, artificial grassland, improved grassland, transportation, communication, energy pipeline, power transmission and other linear infrastructure, wind power, photovoltaic, marine energy and other facilities, and military, cultural relics, religious, funeral, and other special land in areas with extremely important ecological functions and extremely fragile ecology can be designated to enter the red line of ecological protection. Within the red line of ecological protection, in principle, human activities are prohibited in the core natural reserves, and development and productive construction activities are strictly prohibited in other areas. Once the red line of ecological protection is delimited, it shall not be adjusted without legal procedures. All localities are not allowed to adjust the ecological protection red line by modifying the land spatial planning at the city, county, and township level without

authorization. The regulations and requirements for transfer in and adjustments of nature reserves and transfer out (major national projects involving land/Sea Island) in the adjustment of red line are introduced in detail. At the end of the “measures,” social supervision, performance appraisal, and accountability are emphasized. The whole management method provides a good system guarantee for ecological protection red line. Due to the institutional adjustment and the state’s attention to the red line, it is believed that the state and local governments will issue further regulations on delimitation, management, and supervision in the next few years.

Red Line Delineation Methods: “Double Evaluation”

“Double evaluation,” the abbreviation of the evaluation of resource and environmental carrying capacity and territorial development suitability, is the important basis and prerequisite for the implementation of spatial planning (Fig. 2). The evaluation of resources and environment carrying capacity is a comprehensive evaluation of natural resources endowment and ecological environment background. The evaluation of territorial development suitability, on the premise of maintaining ecosystem health, comprehensively considers the suitability of human activities such as agricultural production, urban construction, and so on, by



Marine Ecological Red Line, Fig. 2 Technological system evolution in the field of “double evaluation” in China (Tian and Liu 2020)

comprehensively considering the resources and environment elements, location conditions, and specific land space. Based on the integrated evaluation results of “double evaluation,” ecological protection red line could be delineated.

According to the “requirements of several opinions on the establishment of land space planning system” and supervision of implementation issued by the CPC Central Committee and the State Council as well as the “notice on comprehensively developing land and space planning” issued by the Ministry of Natural Resources, the “double evaluation” has become the basis for optimizing the spatial pattern of land and the prerequisite for the compilation of land and space planning. As an important basis of land and space planning, “double evaluation” plays an important role in finding out the basic background of land space, ensuring the scientificity and rationality of land space planning, supporting ecological restoration and land space policy-making. “Double evaluation” is the reference basis for delimiting marine ecological protection red line and determining land and sea use planning indicators.

Due to the richness of the connotation of ecological red line, the demarcation methods of ecological red line are diverse, and there is no unified standard for dividing ecological red line. Based on the importance of ecological function and ecological vulnerability, the existing method for dividing ecological red line is to use geographic information system (GIS) for spatial analysis and processing.

Relationship and Difference Between Marine Ecological Red Line Zoning and Existing Marine Zoning

Marine ecological red line zoning is closely related to marine main function zoning, marine reserve, and marine functional zoning, but there are some differences (Table 1). As the basic and restrictive planning of marine space development in China, marine main function zoning has a wider scope of action and stronger effectiveness than other plans. Marine main function zoning plays an important role in guiding other marine-related planning and is an important carrier and way to

ensure the implementation of marine ecological red line zoning, and marine ecological red line zoning and marine function zoning are the important basis of marine main function zoning.

Marine protected areas (MPAs) refer to the areas designated in accordance with the law for special protection and management of coasts, estuaries, islands, wetlands, or sea areas, including the protected objects, for the purpose of marine natural environment and natural resources protection. The core factor of the value of marine reserves is the inherent scarcity and typicality of the protected objects, which is also applicable to the marine ecological red line areas.

Marine functional zoning (MFZ) is a type of marine spatial planning implemented widely in China and one of the three major systems defined in the Law of the PRC on the Administration of Sea Area Use. China adopts “top-down management” for MFZ, in which upper management levels impose clear constraints and restrictions on lower levels. The sea area is divided based on its geographical location, natural resources, natural environmental conditions, and social needs. It is used to guide and restrict the practical activities of marine development and utilization and ensure the economic, environmental, and social benefits of marine development. Marine functional zoning refers to all kinds of functions based on human needs. It emphasizes the economic attribute that the ocean can be used by human beings to produce use value, and it is a man-made overall arrangement of sea area use positioning. Therefore, marine functional zoning is essentially a management method to implement protection in development.

Case of Study

Currently, coastal regions in China are carrying out marine ecological red line zoning. Most of the regions have carried out extensive scientific research for zoning.

The people’s Government of Guangdong Province promulgated and implemented “the outline of the environmental protection plan for the Pearl

Marine Ecological Red Line, Table 1 Differences and relations of several current marine zoning in China

Zoning type	Marine principle function zoning	Marine functional zoning	Marine ecological red line zoning	Marine protected area
<i>Resource development and protection</i>	The optimized and key development zones focus on development, while the restricted and prohibited development zones focus on ecological protection	Develop and protect according to function plan	Strict protection	Restrict human activity
<i>Zoning basis</i>	Environmental carrying capacity, current development density, and development potential of marine resources	Sea area location, natural resources, environmental conditions, and requirements for development and utilization	Marine natural attributes, resources and environmental characteristics, regional ecological sensitivity and vulnerability, ecological importance	Intertidal zone and subtidal zone protected by law or other effective means, including the plants, animals, historical sites, and cultural characteristics inside the area
<i>Purpose</i>	Establish a rational development order, optimize the allocation and utilization of marine resources, protect the environment, and promote sustainable development	Guide the use of sea areas, protect and improve the marine ecological environment, and promote the sustainable utilization of resources	Maintain the integrity and connectivity of regional marine ecosystem	For a conservation purpose, typically to protect natural or cultural resources (MPA 2016)

River Delta” (2005), which put forward a complete ecological red line for the first time in the field of environmental planning. In 2014, Tianjin divided the red line area into six categories, mountains, rivers, lakes, wetlands, parks, and forest belts, and determined the proportion of different zones, and then strictly controlled the corresponding ecological regions. So far, the urban ecological red line system has basically formed. Based on the analysis of the system structure of ecological, water, atmosphere, and other environmental elements, the environmental protection intensity level of each region is determined based on the sensitivity and vulnerability of the areas affected by pollution formation and transmission process, and the important difference of the protection area, and the management is carried out with hierarchical management and control measures. In February 2014, the Ministry of Environmental Protection issued “the national ecological protection red line - Technical Guide for

delimiting ecological function baseline (Trial),” and in June 2016, the Oceanic Administration issued “the technical guide for delimiting marine ecological red line.” These guidelines have a good guiding role for the development and practice of ecological red line theory and delimitation technology in China.

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Marine Operations

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Definitions

Marine operations are nonroutine operation of a limited defined duration related to handling of

object(s) and/or vessel(s) in the marine environment during temporary phases (DNVGL 2017). A marine operation is a process involving interactions between dynamic systems, operational procedures, environmental actions, and human intervention. A marine operation shall be designed to bring an object from one defined safe condition to another.

Examples of Marine Operations

Marine operations include a large variety of activities. The definitions of the following commonly used operations are provided (DNVGL 2016).

- **Towing:** The transfer at sea from one location to another location of a self-floating structure or a structure resting on a barge by pushing/pulling by tugs.
- **Launching:** The activities comprising cutting of sea fastening of a structure resting on a launch barge, the structure's slide-down on the launch rails on the barge and diving into the water until the structure is free floating.
- **Lifting:** The activities necessary to lift or assist a structure by a crane.
- **Load transfer operation:** The operation to transfer the load (i.e., an object) from/to vessel(s) without using cranes, that is, by using (de-)ballasting. Typical load transfer operations are load-out, lift-off, mating, and float-over.
- **Pipelay:** The operation of assembling and laying the pipeline on the seabed, from start-up point to lay-down point.

General Terms Related to Marine Operations

The relevant terms on marine operations are defined as follows (DNVGL 2016):

Operation Reference Period

Duration of marine operations shall be defined as an operation reference period, T_R .

$$T_R = T_{POP} + T_C \quad (1)$$

where T_{POP} is the planned operation period, and T_C is estimated maximum contingency time. T_{POP} shall be based on a detailed schedule for the

operation. In early planning phase, where a detailed schedule is not available, T_{POP} can be based on experience with similar operations. T_C should cover the general uncertainty in T_{POP} , the unproductive time during the operation, and possible contingency situations that will require additional time to complete the operation.

Weather Restricted and Weather Unrestricted Operations

An operation shall be defined as weather restricted or weather unrestricted based on its reference period.

Marine operations with a reference period (T_R) less than 96 h and a planned operation time (T_{POP}) less than 72 h may normally be defined as weather restricted. However, in areas and/or seasons where the duration of the reliable weather forecast is less than 96 h, the maximum allowable T_R is the duration of the reliable forecast. A weather restricted operation shall be planned to be executed within a reliable weather window. If the weather restricted design environmental condition is too low, severe waiting on weather delays can occur.

Marine operations that cannot be defined as weather restricted shall be defined as weather unrestricted operations. Environmental criteria for these operations should be based on long-term statistics. If found beneficial, operations of shorter duration may also be defined as weather unrestricted.

Operational Limiting Criteria

Operational limiting environmental criteria (OP_{Lim}) shall be established and clearly described in the marine operation manual. OP_{Lim} shall not be taken greater than the minimum of: (a) the environmental design criteria, (b) maximum wind and waves for safe working and object handling or transfer conditions for personnel, and (c) weather restrictions for equipment.

Forecasted and Monitored Operational Limits, Alpha Factor

Uncertainty in both the monitoring and the forecasting of the environmental conditions shall be considered. This should be done by defining a

forecasted (and, if applicable, monitored at the operation start) operational criteria, OP_{WF} :

$$OP_{WF} = \alpha \times OP_{Lim} \quad (2)$$

where the alpha factor, α , depends on the planned operation period, T_{POP} , the operational limiting OP_{Lim} significant wave height, the forecast levels and whether or not meteorologists or measurement equipment are available on site (DNVGL 2016).

Assessment of Operational Limits

Operational limits need to be assessed during planning of marine operations. The limits can be expressed in terms of environmental conditions (sea states, wind, and current conditions) or motions that could be monitored on-board the installation vessels before executing the operation. The operational limits depend on the type of operation and the property of the dynamic system. The operational limits can be used to improve the system performance in the planning phase, that is, to select vessels and equipment, to optimize the procedure, and to propose contingency or mitigation actions. They can also be used together with the weather forecasts to support on-board decision making in the execution phase. Traditionally, operational limits have relied mostly on practical marine operation experiences. However, it is important to quantify the responses (forces, motions) and corresponding operational limits for complicated operations with strict requirements. A systematic method to establish the operational limits is required.

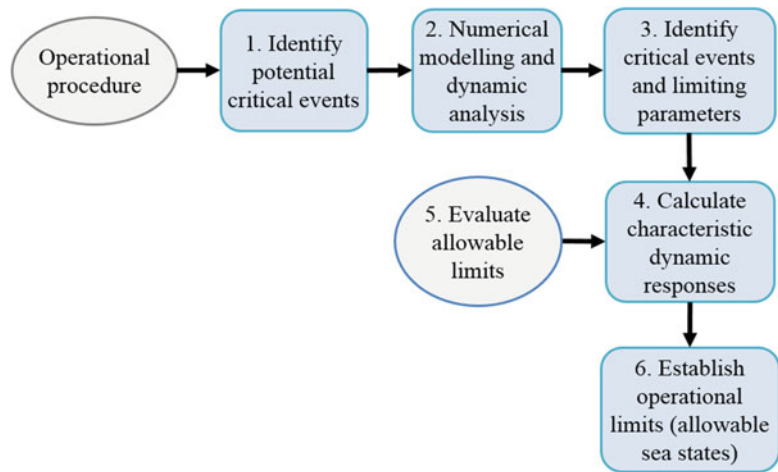
Li (2016) and Guachamin-Acero (2016b) established a general methodology to establish the operational limits (Li 2016; Guachamin-Acero 2016). It includes six main steps, see Fig. 1.

1. Identification of potentially critical events.

Based on the operational procedure, a preliminary selection of activities which could lead to critical events is required. The preliminary selection requires qualitative risk assessment

Marine Operations,

Fig. 1 General methodology to establish the operational limits. (Li 2016)



based on experience, guidelines, and by reviewing of relevant operations.

2. **Numerical modeling of operational activities.** Numerical modeling and analysis of the potential critical events are required to evaluate the dynamic responses of the installation systems. A quantitative assessment of the dynamic responses under reasonable environmental conditions will indicate which response parameters may reach high levels and thus limit the operation.
3. **Identification of critical events and limiting parameters.** By assessment of the dynamic responses, the events governing each offshore activity and leading to failure events are identified. They are identified as critical events, and the governing dynamic responses are defined as limiting parameters.
4. **Calculation of characteristic dynamic responses.** Once the limiting parameter is identified, a characteristic value of its dynamic response needs to be calculated. The dynamic responses should be treated in a probabilistic way to take into account the stochastic nature of the environmental conditions (e.g., wind, wave, and current conditions).
5. **Evaluation of the allowable limits.** The allowable limits are the maximum values that the limiting parameters may reach to remain within acceptable safety margin. It needs to be specified based on safety criteria to avoid structural failure and the exceeding of specified

installation requirements. For events related to structural failure, structural analysis or finite element modeling to determine the strength or the capacity of the structure is normally required to provide the allowable limits.

6. **Assessment of the operational limits.** By comparing the characteristic dynamic responses and their allowable limits for all possible sea states, the allowable sea states at which the responses are less than the allowable limits can be found. These sea states can be transformed into allowable motion responses. In general, both the allowable sea states and responses can be used as operational limits.

This general methodology uses response-based criteria to determine the allowable limits of sea states and to assess the safety of the operation. The operation is considered to be safe if the characteristic value of the governing response is less than the capacity. Often, safety factors need to be considered to take into account the uncertainties associated with the analyses or the data in the assessment of both the load effect and the resistance. Uncertainties on the weather forecasts need to be included by implementing alpha factor. Other uncertainties related to the wave spectra should also be implemented (Guachamin-Acero and Li 2018). This general methodology has been applied in analyses of various marine operations (Li et al. 2016a; Gao et al. 2016; Guachamin-Acero et al. 2017).

Installation Methods for Offshore Wind Turbines

The installation of an offshore wind farm includes the transportation and installation of foundations, turbine components (tower, rotor, and nacelle assembly, RNA), substations, and cables. The most commonly used method for foundation installation is heavy lifting operation using either floating cranes or jack-ups. Turbine components are normally lifted and installed using jack-up vessels in shallow water depths.

A review of the current installation methods for bottom-fixed offshore wind turbine (OWT) foundations, turbine components, as well as floating wind turbines is presented here. Recently developed installation concepts are also discussed.

Installation Methods for Bottom-Fixed Foundations

Methods for installing foundations depend on the foundation type. The most commonly used bottom-fixed foundations include monopiles, gravity-based foundations, jackets, and tripods.

Monopiles

Monopiles are transported either on-board of a barge or an installation vessel or capped and wet towed (Herman 2002). The wet tow of a single floating monopile has been applied during installation of two wind farms (Npower Renewable 2006; Ballast Nedam 2011). The transportation of more than one monopile per trip can be achieved using proper connection between the monopiles.

Installation of monopiles using crane vessels in general includes three main steps: upending, lowering, and hammering operations. A combined wet-tow and upending in water can be performed by lower capacity cranes than those required for transporting and upending on board. However, the upending of long monopiles in water is more weather sensitive than upending on board. Moreover, the verticality of the monopile during hammering operation should also be carefully controlled. Figure 2a shows a monopile lifting operation by the heavy lift vessel (HLV) “Svane.” Li (2016) studied extensively the

monopile lowering operation by considering the nonstationary features of the installation process (Li et al. 2014, 2015, 2016b). The dynamic responses during the hammering process were also investigated (Li et al. 2016a). The allowable sea states in terms of significant wave height (H_s) and wave spectral period (T_p) and the operability at one reference site for the whole monopile installation have been derived (Guachamin-Acero et al. 2016b).

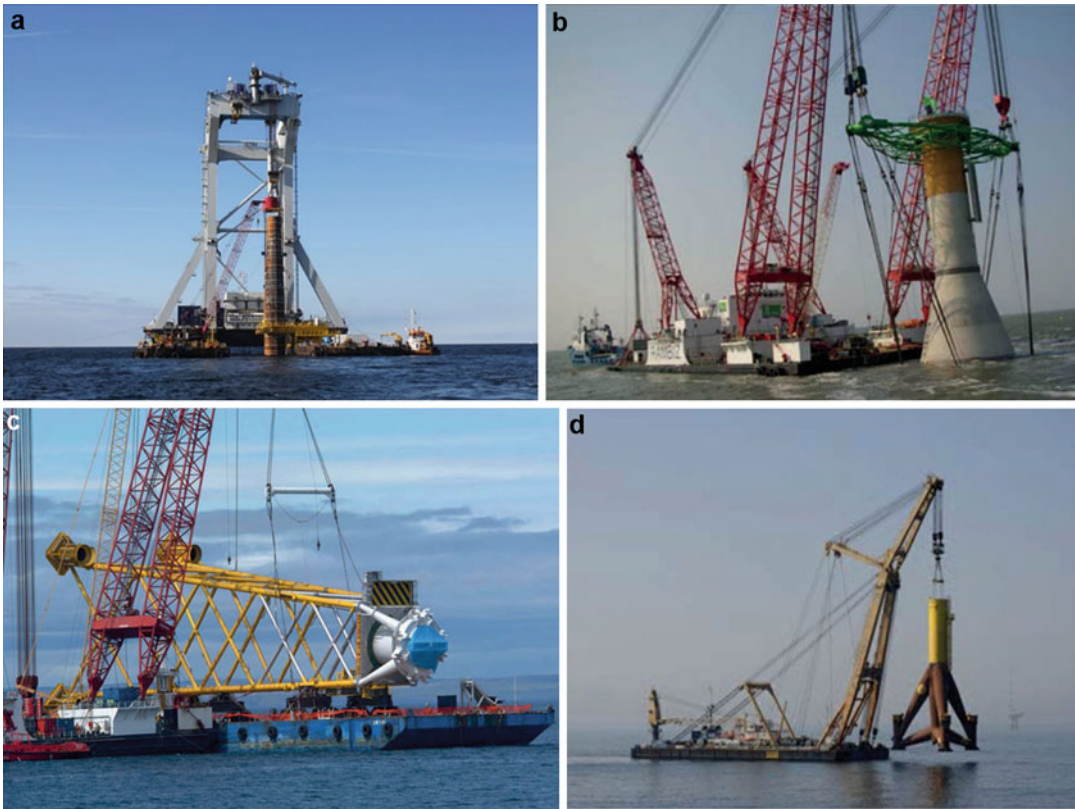
Gravity-Based Foundations

Gravity-based structures (GBS) for wind turbines weigh over 2500 tons. They can either be wet-towed or dry-transported on a barge or an installation vessel. Most of the existing GBS in very shallow waters are dry-transported to the offshore site. Wet-tow can reduce the installation costs by chartering lower capacity lifting vessels. Although the wet-tow method is widely used for transporting gravity-based platforms, it has not been applied for installing GBS of OWTs because their weight is still within the crane capacity of available HLVs. However, if larger GBS are applied for offshore farms in deeper waters, the wet-tow method may be more cost-efficient. Figure 2b displays a lifting operation of a GBS by a HLV.

A novel installation concept was proposed to transport and install a fully assembled gravity-based foundation OWT using double-barge supports and standard tugboats (Wasjø et al. 2013). The concept aimed to reduce the costs by avoiding the use of heavy lift crane vessels. Model tests and numerical studies have been conducted to assess the feasibility of this concept (Bense 2014).

Jackets and Tripods

Jackets and tripods for OWTs can range from 400 to 1000 tons and with a height of 30–90 m in water depth of around 20–70 m. Figure 2c and d shows typical installation activities for jacket and tripod. The jackets can be transported in either an upright or horizontal position depending on the size of the foundations and the available transport barges. The installation of piles could be carried out either after positioning the jackets (*post-piling*) or before jackets installation (*pre-piling*) (LORC 2013).



Marine Operations, Fig. 2 Installation of bottom-fixed OWT foundations. (a) Monopile lifting operation. (Source: <https://www.offshorewind.biz/>), (b) GBS installation.

(Peire et al. 2009), (c) Jacket upending operation. (Source: <http://www.rechargenews.com>), (d) Tripod lifting operation. (Source: <http://www.4coffshore.com/>)

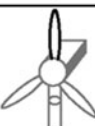
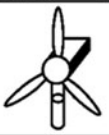
Post-piling is traditionally used in the oil and gas industry and was applied for the jackets in Beatrice wind farm. Experience with pre-piling is limited, but it was first used for the jackets at Alpha Ventus wind farm (Østvik 2010). Pre-piling is considered to be a faster method than post-piling. With pre-piling, smaller vessels can be employed for the piling operation and the HLVs are only required to lift the jackets into the preinstalled piles. With post-piling, the expensive HLVs are used to install both the jacket and the piles. In addition, a considerable amount of steel can be saved using pre-piling because the sleeves for the piles are unnecessary.

Installation Methods for Turbine Components

Transportation and installation methods for turbine components are very different from those for foundations. They must be dry-transported

on-board an installation vessel or a feeding barge. The turbine, particularly the nacelle and the rotor, have sensitive components which only tolerate very limited accelerations during transportation and installation. Moreover, lifting operations are weather-sensitive especially at large lifting heights, and the mating between the components is challenging.

There are many alternatives for turbine installation. Figure 3 shows the commonly used installation strategies. The weight of each component for the NREL 5 MW reference turbine (Jonkman et al. 2009) is included in the figure. The choice of method is related to the vessel size, distance from port to site, size of the wind turbines, and the lifting capacity of the crane. By increasing the amount of onshore assembly, the offshore construction work can be reduced. However, the assembled components reduce the efficiency in

Components and weights	Installation method	Number of lifts					
		1	2	3	4	5	6
2 tower sections:  173.75 ton each	1						
	Lift weight	173.75	173.75	300.5	16.5	16.5	16.5
nacelle:  240 ton	2						
	Lift weight	347.5	300.5	16.5	16.5	16.5	
hub:  110 ton	3						
	Lift weight	173.75	173.75	240	110		
3 blades:  16.5 ton each	4						
	Lift weight	173.75	173.75	333.5	16.5		
	5						
	Lift weight	347.5	333.5	16.5			
	6						
	Lift weight	697.5					

Marine Operations, Fig. 3 Installation alternatives for turbine components including number of lifts and lift weights. (Kaiser and Snyder 2011)

using the deck space and increase the number of trips for transportation when installing a large wind farm. In addition, the weight of the fully assembled turbine with the large lift height requires very large and expensive crane vessels (Ku and Roh 2014).

Among the methods for blade installation, single blade installation is most frequently used for offshore installation in recent years, due to small deck space requirement and flexible blade orientations during installation (Ahn et al. 2017).

During the installation process, the blade is lifted and installed in a feathered position, which is kept during the whole installation operation (Kuijken 2015). Recently, extensive studies on a single blade installation have been carried out. (Zhao et al. 2018a, b) developed an integrated aero-hydro-soil-elastic-mechanical tool for simulating installation of a single blade for wind turbines. (Jiang et al. 2018a) studied the final blade installation phase onto a monopile wind turbine. The effects of mean wind speed, wind turbulence,

significant wave height, wave spectrum peak period, wind-wave misalignment, and water depth on the blade installation were evaluated. Moreover, some studies have been focusing on developing active control schemes to control the tugger lines used for blade installation, with the purpose to increase the weather window for the final mating phase between the blade root and the hub (Ren et al. 2018a, b).

In addition to the methods shown in Fig. 3, a few innovative concepts to install fully assembled tower and RNA have been proposed. Sarkar and Gudmestad (2013) designed a floating substructure to transport the fully assembled structure, but the concept was only applicable for telescopic tower (Sarkar and Gudmestad 2013). Guachamin Acero et al. (2016a) proposed a new concept based on the principle of the inverted pendulum (Guachamin Acero et al. 2016a). The feasibility of this installation concept was assessed by numerical analyses.

Installation Methods for Floating Wind Turbines

Installation methods for floating wind turbines are quite different from those for bottom-fixed wind turbines due to larger water depth of the installation site. For turbine installation in shallow water, jack-up vessels are normally used to provide stable platform for lifting and offshore bolting, which could hardly be carried out by a floating vessel with motions in six degrees of freedom. However, for deep waters, if onshore assemble is chosen, the installation can only be performed by floating vessels, and it was proven to be difficult and costly.

The installation method differs with different floating structures. The Hywind Scotland, which was the first offshore floating wind farm, applies a spar-type floating wind turbine. The spar buoy was dry-towed in a horizontal configuration using a transportation vessel and upended to a vertical position by ballasting the spar. The tower, nacelle, and blades were preassembled onshore and installed by a single lift using a large heavy lift vessel. Both the upending of the spar and the mating between the tower and the spar buoy are critical operations. In order to

increase the operability, the installations were done at a well-sheltered location in deep water. Then, the fully assembled floating unit was towed to the site and hooked up to the mooring system (Lien 2016).

The installation of Windfloat, on the other hand, was the first offshore wind turbine to be deployed without the use of any offshore heavy lift vessel. Due to the shallow draft of the semi-submersible foundation, the turbine could be fully assembled onshore prior to the unit being towed offshore (Roddier et al. 2010). In this way, all the weather-sensitive lifting operations can be carried out onshore, which could reduce the installation costly significantly when installing a large number of wind turbines with this type of foundations. Once at shore, the floating structure was hooked up to the preinstalled mooring system.

In order to reduce the installation cost to make floating wind turbine more cost-effective, innovative installation strategies are required. In the Hywind installation challenge campaign (Equinor 2018), and among the proposed innovative installation concepts, there is a tendency to favor novel installation vessels and facilities to reduce offshore lifts and operation time. A novel wind turbine installation concept was proposed to use a catamaran installation vessel for installations of fully assembled OWTs (Jiang et al. 2018b, c). This concept avoids extremely weather-sensitive high lifts from a floating vessel by carrying the wind turbine on the catamaran and a gripper device. This installation method was deemed to be suitable for both offshore bottom-fixed or floating foundations.

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Marine Protected Area (MPA)

► [Marine Protected Areas in Areas Beyond National Jurisdiction](#)

Marine Protected Areas in Areas Beyond National Jurisdiction

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Synonyms

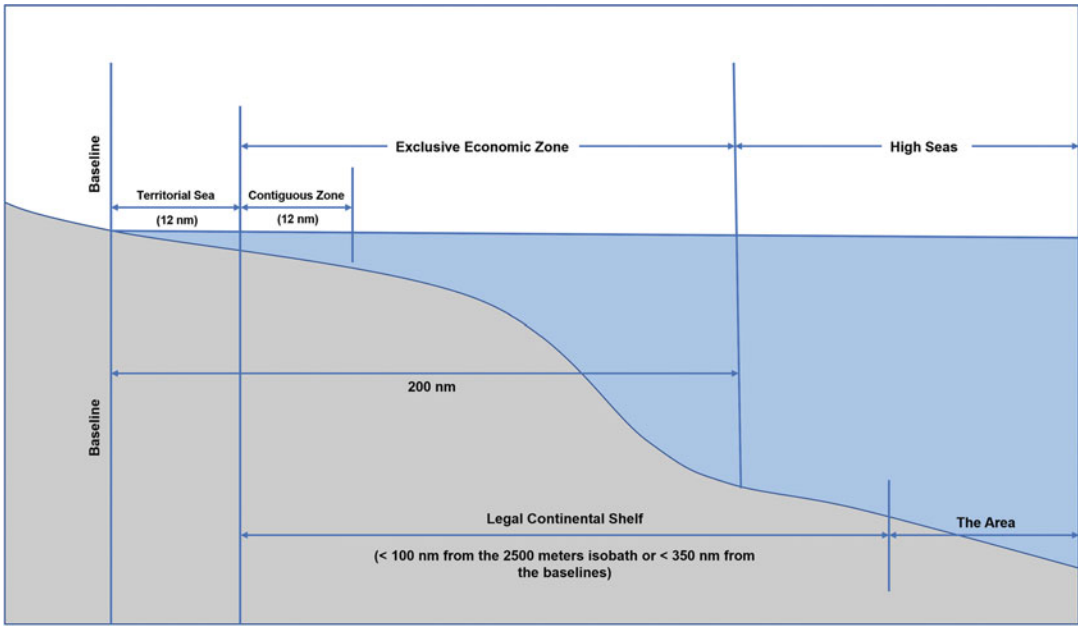
[Areas beyond national jurisdiction \(ABNJ\)](#); [Marine protected area \(MPA\)](#); [Marine protected areas in areas beyond national jurisdiction \(MPA in ABNJ\)](#)

Definition

A marine protected area (MPA) is “a clearly defined geographical space, recognized, dedicated and managed, through legal or other effective means, to achieve the long-term conservation of nature with associated ecosystem services and cultural values” (Muraki Gottlieb et al. 2018). If a part of or whole MPA is located on the ABNJ, then we call it “marine protected areas in areas beyond national jurisdiction (MPA in ABNJ).” MPAs do not mean absolutely no fishing or other human activities; in fact, most MPAs in the world are multipurpose areas. The no-fishing MPAs are often referred to as “no-take areas.”

Scientific Fundamentals

Areas beyond the national jurisdiction (ABNJ) amount to approximately 61% of the oceans, 44% of the surface of the Earth, and 65% of the volume of the biosphere but remain the least protected space on the planet (Gjerde et al. 2016). However, these areas are increasingly under threat from anthropogenic activities such as pollution, overfishing, mining, and geoengineering. At the UN Conference on Sustainable Development in Rio de Janeiro in June 2012, the world’s political leaders committed to the conservation and sustainable use of marine biological diversity in areas beyond national jurisdiction (ABNJ) (UN 2012). After that, the General Assembly decided to convene an Intergovernmental Conference to elaborate the text of an international legally binding instrument under the United Nations Convention on the Law of the Sea (UNCLOS) on the conservation and sustainable use of marine biological diversity of areas beyond national jurisdiction. The Conference will meet for four sessions. The first three sessions were conducted in 2018 to 2019. By decision 74/543 of March 11, 2020, the General Assembly decided to postpone the fourth session of the conference to the earliest possible available date to be decided by the General Assembly (UN 2020) (<https://www.un.org/bbnj/>). After



Marine Protected Areas in Areas Beyond National Jurisdiction, Fig. 1 Diagram of maritime zones by UNCLOS

this, a new regime about the marine protected areas in ABNJ will be expected.

ABNJ legally comprise the “high seas,” which indicated the waters beyond the exclusive economic zones (EEZ) of national jurisdiction, and the “international seabed area” (“the Area”), which indicated the seabed, the ocean floor, and subsoil thereof beyond national jurisdiction (UNCLOS Articles 1 and 86) (Fig. 1). If a part of or whole MPA is located on the ABNJ, then we call it “MPA in ABNJ.”

Objectives of Establishing MPAs in ABNJ

For the past few decade years, the ABNJ were under increasing and intensive threats from human activities, such as overfishing, pollution, maritime shipping, and climate change. Besides, it will also be strongly threatened by potential deep-sea resource exploitation, including oil, gas, and metallic ores, in the future (Sharma 2015; Gjerde et al. 2016). These issues not only threaten the biodiversity in ABNJ (Oleary et al. 2020) but also depress the ecosystem function and service in

these areas (Worm et al. 2006; Rogers et al. 2014) (Table 1).

MPAs are widely considered as an important tool for the conservation of biological diversity and marine ecosystems in the past several decade years (Lester et al. 2009). By the end of August 2020, 17,237 MPAs were established in the world, and they covered near 27 million square kilometers in area, about 7.45% area of world’s oceans. However, only less than one in a thousand MPAs were established in ABNJ (Smith and Jabour 2018). It meant that the ABNJ were largely under absolutely unprotected conditions (Gjerde et al. 2016).

The Potential Cost of MPAs in ABNJ

The establishment of MPAs in ABNJ, especially the “no-take” MPAs, could be also with some cost, at least in short term. First, fisheries data show that 6% of global fish catch is obtained in ABNJ (www.seaaroundus.org/data/#/global). Thus, if the current and potential fishing areas in ABNJ are protected and all forms of fishing are strictly prohibited in these areas, the global supply of fish will be

Marine Protected Areas in Areas Beyond National Jurisdiction, Table 1 The current and potential threats to biodiversity and ecosystems in ABNJ, from human activities and climate change

Activity	Impacts or threats to biodiversity and ecosystems in ABNJ
Fishing	Fishing is the most significant direct threat to biodiversity in ABNJ now and also with high and widespread environmental impacts to habitats and ecosystems (Pauly et al. 2005)
Maritime shipping	Given the large spatial extent of shipping activities, environmental impacts are likely to be high. For example, anthropogenic noise from ship was considered to be with great impacts on marine life (Nowacek et al. 2010; Williams et al. 2015)
Climate change and associated effects	Ocean ecosystems are already experiencing significant change mainly because of ocean acidification, hypoxia, and warming, and changes will continue to extend across ABNJ in the future (Doney et al. 2012)
Land-based pollution	Environmental impacts from land-based pollution are high and affect all aspects of marine life, even in deep seafloor (Van Cauwenberghe et al. 2013)
Deep-sea mining	Environmental impacts are likely to be high and broadly distributed and impact the entire seabed to surface ecosystem. Although deep-sea mining is not a current threat in “the area,” it is widely considered as a reality in the near future (Durden et al. 2018)
Oil and gas exploration and exploitation	The local and regional impacts are experienced which can have high negative environmental impacts to the entire seabed to surface ecosystem (Gray et al. 1999). No offshore oil and gas drilling operations currently take place in ABNJ, but it could happen accompanied by shortage of energy and technological advancement in the future
Bioprospecting for marine genetic resources and scientific research	Bioprospecting and scientific research have a small scope and temporal and spatial footprint. Thus, the environmental impacts to biodiversity and ecosystems are considered to be weak
Aquaculture	Currently, aquaculture activities are mainly concentrated near to shore in EEZs, and the expansion of aquaculture offshore is likely to be contained within areas of national jurisdiction in the future. But, some ecological and environmental impacts of aquaculture activities are still largely unknown, such non-native species escapes from aquaculture (Ju et al. 2020)
Renewable energy	Currently, engineering construction of renewable energy in marine remains in EEZs, and the expansion is likely to be contained within areas of national jurisdiction in the future
Military	Because of its secrecy, less data are able to be used to evaluate the environmental impacts on marine biodiversity and ecosystems. However, a limited amount of evidence showed the potential impact from military, especially the underwater noise, could not be ignored (Tyack et al. 2011; Morell 2015)
Submarine cables and pipelines	The submarine cables only cover 0.00002% of seabed in ABNJ, and no significant and widespread environmental impacts were found (Taormina et al. 2018). The oil and gas pipelines are considered to be with great potential environmental impacts in offshore and continental shelf (Dey et al. 2004), and it was unlikely given the lack of oil and gas exploration and exploitation in deep sea of ABNJ

Adapted from Oleary et al. (2020)

reduced, especially the tunas and squids (Sala et al. 2018). Second, a large proportion of maritime shipping routes are located on the ABNJ. Environmental protection terms of MPAs in ABNJ, such as

control of air pollution emissions and underwater noise levels of ships in the protected area, will inevitably increase the costs of maritime shipping (Gallucci 2018). Third, some underwater

engineering, such as laying or maintaining of submarine cables, pipelines, and other facilities, may face stricter environmental impact assessment.

Key Applications

The Current Status of MPAs in ABNJ

MPAs in ABNJ: Lessons from Two High Seas Regimes
Although there is still much debate about the details, establishing a network of MPAs in

ABNJ is viewed as an important measure to protect marine biodiversity and widely accepted (Smith and Jabour 2018). To date, 13 MPAs have been established in ABNJ: 2 in the Southern Ocean, 10 in the North-East Atlantic region, and 1 in the Mediterranean Sea (Table 2).

The Future Harder Task

The world’s ocean represents the largest ecosystem on Earth comprising 1.3 billion km³ of water. Considering the strong and widespread

Marine Protected Areas in Areas Beyond National Jurisdiction, Table 2 Thirteen MPAs in ABNJ (including wholly or partly) have been established to date

	Name of MPAs	Location	Remarks
1.	South Orkney Islands southern shelf MPA	Southern Ocean	It is the world’s first wholly high-seas MPA (in 2009). And it is also the first no-take marine reserve in the CCAMLR’s network of Southern Ocean MPAs
2.	Ross Sea region MPA	Antarctica and Southern Ocean	The world’s largest MPAs in ABNJ (It covers 1.55 million square kilometers, of which 72% is “no-take” area.). It is also in the CCAMLR’s network of Southern Ocean MPAs
3.	Pelagos Sanctuary	Mediterranean Sea	Maybe, it is the first MPAs in ABNJ (in 2002), because it currently is located in part on the high seas. The Sanctuary has been designated and is jointly managed by France, Italy, and Monaco
4.	Charlie-Gibbs South MPA	North-East Atlantic Ocean	Entirely within ABNJ. The seabed, the subsoil, and the water column are protected by all OSPAR CPs
5.	Milne Seamount Complex MPA		
6.	Mid-Atlantic Ridge north of the Azores High Seas MPA		
7.	Altair Seamount High Seas MPA		
8.	Antialtair High Seas MPA		These four MPAs are located within an area subject to a submission by Portugal to the UN CLCS for an ECS. Portugal has expressed the intention to assume the responsibility to take measures for the protection of the seabed and the subsoil within these areas. Upon invitation by Portugal, the OSPAR Commission agreed to collectively protect the water column of these MPAs
9.	Josephine Seamount High Seas MPA		
10.	Charlie-Gibbs North High Seas MPA		
11.	Rainbow Hydrothermal Vent Field SAC		This MPA is partly situated within an area subject to a submission by Iceland to the UN CLCS for an ECS. The water column is protected collectively by all CPs. The seabed and the subsoil remain unprotected
12.	Hatton Bank		
13.	Hatton-Rockall Basin		
			These MPAs are situated within areas subject to a submission by a CP to the UN CLCS for an ECS. The seabed and subsoil of these sites are protected by the respective CP, while the water column remains unprotected

CCAMLR: the 1980 Commission for the Conservation of Antarctic Marine Living Resource
OSPAR: the 1992 Convention for the Protection of the Marine Environment of the North-East Atlantic (also referred to as the OSPAR Convention)
UN CLCS: Commission on the Limits of the Continental Shelf, United Nations
ECS: Extended Continental Shelf
CP: Contracting Party of OSPAR

connectivity of marine ecosystem and the multiple threats on marine biodiversity, more MPAs in ABNJ are absolutely necessary in the future. However, there are still some barriers about more MPAs in ABNJ. First, recent study showed that 54% of the present fishing grounds in ABNJ would be unprofitable at current fishing rates, if the subsidies from governments disappear. The first five countries which provide largest subsidies are Japan, Spain, China, South Korea, and the United States. All of these countries also organize the largest high-seas fishing fleet in the world (Sala et al. 2018). Thus, we must more rationally regulate high-seas fishery subsidies and management, along with the process of establishing more MPAs in ABNJ. Second, several independent frameworks to manage and protect the marine ecosystems in ABNJ have already existed. For example, fishing is managed regionally by some regional fishery management organizations (RFMOs), shipping is managed by various conventions under the International Maritime Organization (IMO), and deep-sea mining in “the areas” is managed by the International Seabed Authority (ISA) (Altwater et al. 2019). Nonetheless, cooperation and coordination among these different frameworks are largely limited, which challenges the protection of marine ecosystems in ABNJ. Third, our understanding of deep-sea biodiversity and deep-sea ecosystem functions is far from enough until now (Appeltans et al. 2012, Thurber et al. 2013). This gap in knowledge is a great barrier in establishing of MPAs in ABNJ. To overcome it, more new methods, such as data-driven planning tools (Visalli et al. 2020), are needed to instructing the planning model for MPAs.

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Marine Protected Areas in Areas Beyond National Jurisdiction (MPA in ABNJ)

- [Marine Protected Areas in Areas Beyond National Jurisdiction](#)

Maritime Ocean Engineering Research Institute (KRISO)

- [Ice Tank Test](#)

Material Takeoff (MTO)

- [Steel Pipelines and Risers](#)

Maximum Likelihood Sequence Detection (MLSD)

- [Underwater Acoustic Communication](#)

Maximum Likelihood Sequence Estimation (MLSE)

- [Underwater Acoustic Communication](#)

MBS – Multibody Simulation

- [Analysis of Renewable Energy Devices](#)

MDO Method

- [Multidisciplinary Design Optimization \(MDO\)](#)

MDO-Computing Framework

- [Multidisciplinary Design Optimization \(MDO\)](#)

ME – Morison’s Equation

- [Analysis of Renewable Energy Devices](#)

Medium Access Control Layer

- [Underwater Acoustic Sensor Network](#)

Mega-Float

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Synonyms

[Mobile offshore base \(MOB\)](#); [Very large floating structures \(VLFS\)](#)

Definition

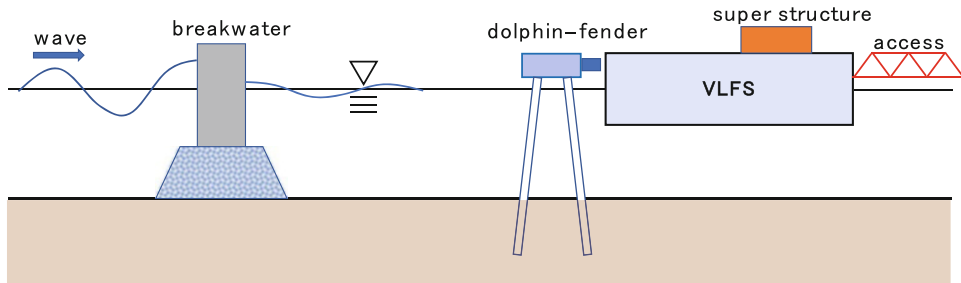
Mega-Float is a type of VLFS whose concept was developed, particularly in Japan. Mega-Float

originates from a coined word meaning “Mega” (huge) + “Float” (floating structure). It is sometimes referred to as “Mega-Float.” Technological Research Association of Mega-Float (TRAM) in Japan initiated the project with the same name “Mega-Float” in the 1990s. Mega-Float also denotes such a pontoon type very large floating structures (VLFS) for ocean space utilization as studied and developed by TRAM. The objective of Mega-Float is ocean space utilization, being deployed for long years in a sheltered sea area, which is contrary to Mobile Offshore Base (MOB) deployed in rather offshore sea area for sometimes tentative use. The Mega-Float project by TRAM started in 1995 and continued for 6 years. The focus of Phase I project (1995–1996) was on the development of fundamental technologies such as design and construction of VLFS while it was on the demonstration as an airport in Phase II (1997–2001). A VLFS with the length 1000 m was designed, constructed, and tested to demonstrate its function as a floating runway in Phase II project. This VLFS was regarded as the largest man-made floating island and certified in 1999 by the Guinness World Records. A semi-submersible type Mega-Float (SSMF) for offshore deep waters was also studied by TRAM. However, the investigations into SSMF were limited. This article mainly overviews the Mega-Float concept and the technologies developed in Mega-Float projects that were carried out in Japan between 1995 and 2001. Reference may also be made to the literatures such as Suzuki (2005) and Fujikubo and Suzuki (2015).

Scientific Fundamentals

Mega-Float Concept

The concept of Mega-Float developed by TRAM is illustrated in Fig. 1. Mega-Float is installed in a sheltered sea area (such as bay), and it consists of a large floating structure and a mooring system. An access from land can be a part of it. The breakwater to protect Mega-Float from severe waves is also constructed around it if necessary. The structure is constructed by the at-sea joining



Mega-Float, Fig. 1 Conceptual sketch of Mega-Float system (Taken and reproduced from SRCJ web page)

and welding of the steel structural units that have been pre-constructed in shipyards. It can create artificial ocean space on the sea, of an arbitrary shape fitting for its function.

The advantages of Mega-Float over the land reclamation may be summarized below.

- Mega-Float can be installed in the sea area with soft ground and/or deep waters.
- Mega-Float can comply with high tidal variation
- Mega-Float is extensible, movable, and removal. Design has a high flexibility.
- Mega-Float gives less impact to marine environments.
- Mega-Float construction costs lower and is made in a shorter period.
- Mega-Float is stronger against ground motions in earthquakes.

Design Technology of Mega-Float

Numerical simulation methods had to be developed for the design since Mega-Float was an unprecedented structure and the design need to be proceeded together with the numerical analysis (Design-by-Analysis). The key issue is hydroelasticity. Several hydroelasticity codes for VLFS have been developed and cross-validated in the Mega-Float project since the design of VLFS must be carried out together with the numerical simulation tool which takes account of the hydroelasticity of Mega-Float. Most of the codes are based on the diffraction theory in frequency domain. Structural modeling is different among the codes; some used beam elements, while the other plate theory, shell FEM, etc.

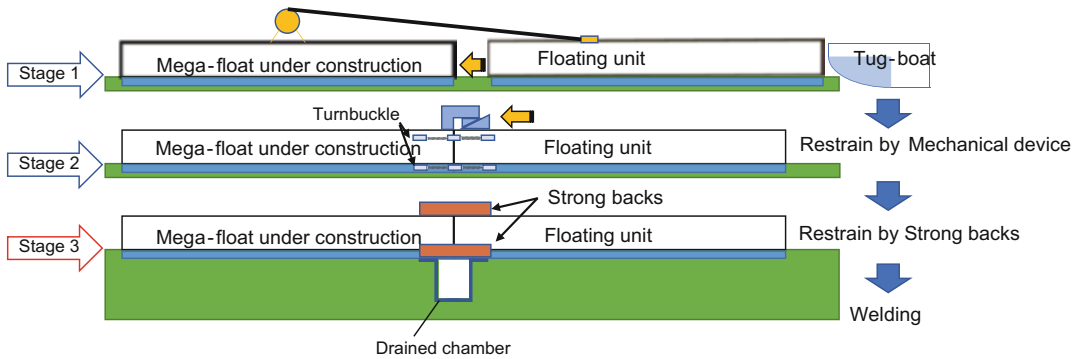
Other technological issues have also been investigated. They include the optimum arrangement of mooring system, and safety evaluation in the extreme event and catastrophic event (drop object, collision, earthquakes, damaged condition). Hydraulic tests using scaled models are also performed to validate the numerical simulations.

Environmental impact has also been investigated in terms of flow blockage, salinity change, and temperature distribution. The impact on ecosystem around and below Mega-Float is clarified by the numerical simulations and field measurement in Mega-Float project. The results indicate only little effect, except for a small reduction in dissolved oxygen adjacent to Mega-Float.

Construction Technology of Mega-Float

Mega-Float is finally assembled and constructed at sea. The construction process is as follows (Yamashita et al. 2007; see Fig. 2):

- Unit structures (modules) are pre-fabricated/ built in shipyards. The maximum size of the unit structure depends on the size of the dock, however, typically ranging from 100 m to 300 m in length. The ship-building technologies can be applied.
- The units are towed out from the dock to the site by using towing boats. The relative motion among the units are restrained and stiffened gradually by using mechanical joints and finally welded together. When joining two structures, at least, one of the two is moored to the “dolphin.”



Mega-Float, Fig. 2 Construction process of Mega-Float

- At the beginning of the restraining process, mechanical devices such as jacks, turnbuckles and wooden wedges are used to reduce the relative motion. In the later stage, the degree of restraints is strengthened. Steel strong-backs and fitting pieces are welded to the units, in order to conduct the welding works with good quality between the two units.
- For the underwater welding, a specially designed chamber attached to the bottom of a hold compartment is drained of the water by using blowers to pump in the compressed air.
- The above process is repeated until Mega-Float is formed at sea.
- Mega-Float is moored to the mooring facility in and after the construction process.
- The thermal deformation to the sunlight and welding deformation are to be considered in the fabrication and construction.

Mooring Technology of Mega-Float

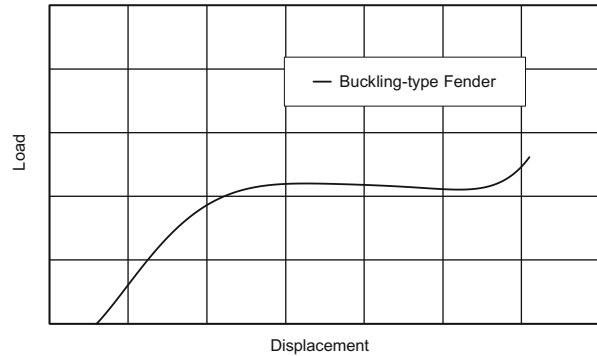
The mooring system of Mega-Float may be selected and designed from various options depending on the water depth and environmental conditions. For deep waters, conventional catenary mooring system may be adopted. For shallow to mid-water depth (20–40 m), the mooring system consisting of steel pile/jacket structure and fender is typically adopted. The marine jacket for mooring is sometimes called “dolphin.” The dolphin is piled to sea floor. Mega-Float is equipped with steel “arms” to hold the dolphin. Between the arms and dolphin, the fenders are inserted for shock-absorbing the contact loads between the

dolphin and Mega-Float. The fender is a rubber-made structure, with the diameter 1–2 m, which can bear the large drift force on Mega-Float due to waves, wind, and current. It is attached either to the dolphin or the floating structure side.

Figure 3 shows a sample of dolphin-fender system used for Mega-Float (left) and the non-linear reaction force characteristics of the mooring system (right). Normally, there is a gap between the fender and Mega-Float, of few tens of centimeters as the excess play. Within the excess play, Mega-Float moves freely. No reaction force is supplied in this region. When the contact starts between the fender and the float, or the fender or the dolphin, the fender gives the reaction force to the structure in proportion to the contact displacement. When the reaction force reaches a certain level to the increased displacement, the fender undergoes buckling of the rubber. Then, the reaction force becomes almost constant to the further increase of displacement. This reaction force characteristics are beneficial; the mooring system allows large displacement and dissipate the energy, while the constant load does not damage the dolphin structure due to the further increase of the reaction force.

Historical/Societal Background of Mega-Float

Japan is an island country surrounded by seas. About 80% of the land is mountainous, while the rest of 20% is used as the industrial or city areas, and they are concentrated near the shorelines. Ocean space utilization has been considered necessary, particularly in the metropolitan areas.



Mega-Float, Fig. 3 Mooring system for Mega-Float and its mechanical characteristics. (Reproduction from Iijima et al. 2002)

However, the seas around the metropolitan areas that had good conditions for reclamation in terms of water depth and soil condition was already reclaimed. In the 1970s, the feasibility of a floating airport was investigated for the phase 1 construction of Kansai airport using a semisubmersible type VLFS, as a gateway to Osaka (Takarada 1982). Even though the proposal of building the floating runway for Kansai airport was not accepted, then the ship-builders of Japan and academia started to pursue the technologies for VLFS. With these backgrounds, TRAM was founded in 1995 and started their investigations (Okamura 2004).

Key Applications

Mega-Float Phase I Project

TRAM was comprised of 17 leading shipbuilding companies and steel companies in Japan and was founded in April, 1995, on the initiative of Japanese government. The research activities were carried out in Phase I and Phase II projects. Phase I project focused on the development of the fundamental technologies of design and construction. Phase II project focused on the demonstration (Technical Research Association of Mega-Float 1999, 2000, 2001; Sato and Inoue 2003; Yoshida 2003).

The main aims of Phase I were (1) establishment of technologies for design, (2) establishment of technologies for the construction, (3) evaluation

of impact on the upper structures, (4) establishment of long-term durability, and (5) establishment of evaluation method for environmental impact. In Phase I project, the application of Mega-Float was not specified to the airport. The construction technology as explained in the “Construction Method of Mega-Float” section was pursued.

For establishing the joining technology, a small prototype Mega-Float structure with the length 300 m, the width 60 m, and the depth 2 m was constructed as sea off Yokosuka, Japan. The Mega-Float consisted of nine units, each of which had the size 100 m $\times$ 20 m. For the welding technique development, the welding shrinkage and deformation were measured in the at-sea test. The dry welding using the drainage system was found to be effective in securing the welding quality rather than wet welding. The small prototype Mega-Float in Phase I was later reinforced and extended to the Mega-Float with the length 1000 m.

Mega-Float Phase II Project

The application as an airport, whether or not airplanes can land and take-off, was questioned for Phase II project. TRAM designed and constructed a floating runway (see Fig. 4) using Mega-Float technologies off Yokosuka (the same site as Phase I). The size of the structure was 1000 m in length, 60 m in breadth (121 m partially), and 1 m in draft. The structural depth was 3 m. The size was equivalent to the existing small commuter airport. The frequency domain simulation was extended to



Mega-Float, Fig. 4 1000 m class Mega-Float off Yokosuka, Mega-Float Phase II project (By the courtesy of Shipbuilding Research Centre of Japan)

M

1000 m airport case and used for checking its function. Time domain simulation was developed, and the taking-off and landing of an airplane was simulated for safety.

For functionality check as an airport, several considerations on the facilities were further necessary. They include the following:

- The dolphin top must be below the line extending at a predetermined angle from the end of the runway known as “Obstacle Limitation Surface.” Low-headed dolphin for mooring system was developed.
- Approach lights arranged on fixed piles and Mega-Float must be in line within a certain allowance to guide the airplane landing.
- The surface of the steel Mega-Float structure which has dynamic deformation to waves need to be paved with asphalts. Pavement method was investigated, and durability test was performed.
- Navigation equipment such as “Instrumental Landing System (ILS),” “Precision Approach Pass Indicator (PAPI),” and “Future Air Navigation System (SAS)” must function on Mega-Float.

After the completion of the at-sea tests, the Mega-Float structure in Phase II was dismantled (actually, dismantling works were part of the research activities) and removed from the test site. Some parts were reused as “Data Backup Base” in case of disaster, “World Cup Mega-park,” and marine fishery parks. TRAM was dissolved in 2001, and the activities were succeeded by the Shipbuilding Research Centre of Japan (SRC).

Haneda Airport Extension Bid

Haneda airport located in Tokyo Bay is famous as a gateway to Tokyo and one of the world largest and busiest airports. The Japanese Government

decided to construct the fourth runway to Haneda airport which was planned to be in operation in 2009. The fourth runway was to be constructed near the river mouth of Tama River. There were three concepts proposed for the runway: Mega-Float, pile-supported wharf, and a hybrid of reclamation and pile-supported wharf. The Shipbuilders' Association of Japan bid for the construction and proposed the Mega-Float with the length 4000 m (Sato and Inoue 2003; Yoshida 2003). The design service life was 100 years. The design wave height at the site was 2.3 m in significant wave height and the significant wave period 5 s for operation condition. Even for the storm condition, the significant wave height was 3.7 m considering a storm with 200 years of return period. The technical committee to the fourth runway of Haneda airport concluded that all of the three construction proposals have the possibility of being adopted with not serious technical defects. Finally, the consortium for the hybrid of reclamation and pile-supported wharf made a successful bid.

Other Applications

Mega-Float may be used as airport, container terminal, recreation spot, emergency rescue base, storage facility, marine renewable energy platform, solar energy platform, etc. There are various application proposals for these applications. However, no Mega-Floats have been realized so far, except the ones in the Mega-Float projects. Recently, Mega-Float seems to be re-gaining attentions by the engineers. The development of autonomous ship, hydrogen-driven ships, etc. may imply a need for the offshore base, or an ocean space in deeper waters. Mega-Float may also be utilized in such a context.

Cross-References

- Connectors of VLFS
- Station-Keeping System for VLFS
- Structural analysis and design of VLFS
- Very Large Floating Structures (VLFS): Overview

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Melt Inert Gas Welding (MIG)

- Welding Technology

Metacentric Height

- AUV/ROV/HOV Stability

Metal Net

► Net Structures: Design

Metallic Tubes

► Umbilical Cable

Meteorological Monitoring and Measurement Buoy

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Definition

Meteorological monitoring and measurement buoys (aka weather buoys) are the buoys to measure parameters for monitoring and weather forecasting. The measurements include air temperature above the ocean surface; wind speed and direction; barometric pressure; sea water temperature and salinity; wave height, period and direction; and current etc. The weather buoys can be an important part of the global weather monitoring and prediction system (see the ocean data buoy in Fig. 1).

Generally, raw data can be processed and logged on board of the buoy and then transmitted via radio, cellular, or satellite communications to meteorological centers for use in weather forecasting and climate study.

Scientific Fundamentals

Meteorological buoys are instruments which collect weather and ocean data within the world's oceans, as well as aid during emergency response to chemical spills, legal proceedings, and engineering design. Moored buoys have been in use since 1951, while drifting buoys have been used

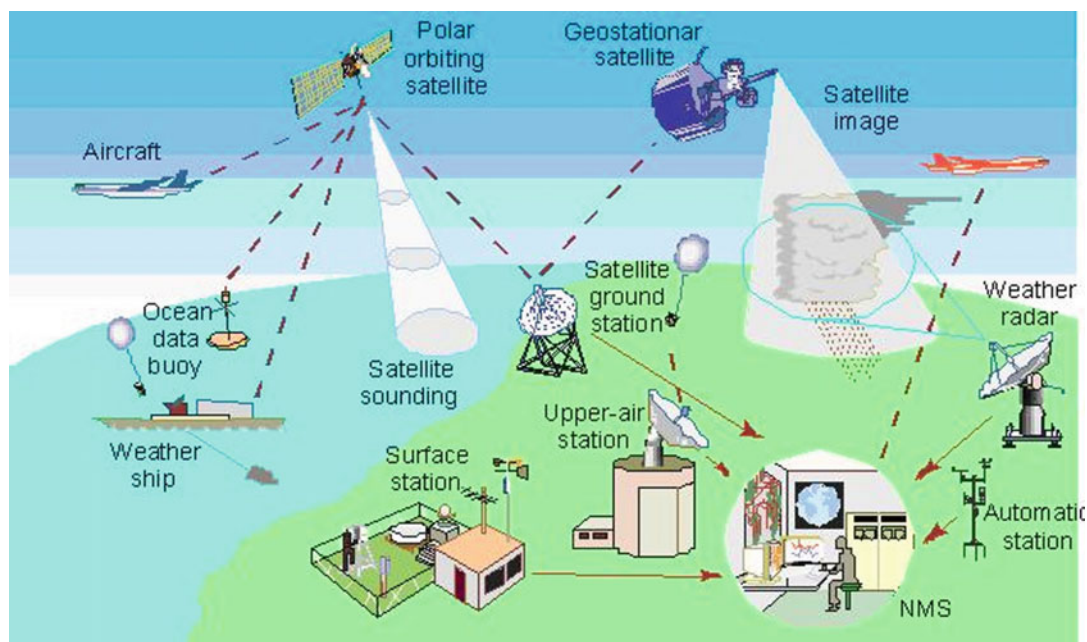
since 1979. Moored buoys are connected with the ocean bottom using either chains, nylon, or buoyant polypropylene. Different types of weather buoys are used by different organizations (see Figs. 2 and 3).

The first known proposal for surface weather observations at sea occurred in connection with aviation in August 1927, and starting in 1939, United States Coast Guard vessels were being used as weather ships to protect transatlantic air commerce.

During World War II The German Navy deployed weather buoys (*Wetterfunkgerät See* – WFS) at fifteen fixed positions in the North Atlantic and Barents Sea. They were launched from U-boats into a maximum depth of ocean of 1000 fathoms (1,800 m), limited by the length of the anchor cable. Overall height of the body was 10.5 m (of which most was submerged), surmounted by a mast and extendible aerial of 9 m. Data (air and water temperature, atmospheric pressure and relative humidity) were encoded and transmitted four times a day. When the batteries (high voltage dry-cells for the valves, and nickel-iron for other power and to raise and lower the aerial mast) were exhausted, after about 8–10 weeks, the unit self-destructed.

The Navy Oceanographic Meteorological Automatic Device (NOMAD) buoy's 6-m (Fig. 4) hull was originally designed in the 1940s for the United States Navy's offshore data collection program. The United States Navy tested marine automatic weather stations for hurricane conditions between 1956 and 1958, though radio transmission range and battery life was limited. Between 1951 and 1970, a total of 21 NOMAD buoys were built and deployed at sea. Since the 1970s, weather buoy use has superseded the role of weather ships, as they are cheaper to operate and maintain. The earliest reported use of drifting buoys was to study the behaviour of ocean currents within the Sargasso Sea in 1972 and 1973. Drifting buoys have been used increasingly since 1979, and as of 2005, 1250 drifting buoys roamed the Earth's oceans.

Between 1985 and 1994, an extensive array of moored and drifting buoys was deployed across the equatorial Pacific Ocean to monitor and help



Meteorological Monitoring and Measurement Buoy, Fig. 1 WMO global observing system (http://www.wmo.int/pages/prog/www/TEM/WMO_RFC/index_en.html)



Meteorological Monitoring and Measurement Buoy, Fig. 2 Oceanographic buoy operated by the marine data service



Meteorological Monitoring and Measurement Buoy, Fig. 3 Weather buoy operated by the National Data Buoy Center



Meteorological Monitoring and Measurement Buoy, Fig. 4 A 6-m NOMAD

predict the El Niño phenomenon. Hurricane Katrina capsized a 10 m buoy for the first time in the history of the National Data Buoy Center (NDBC) on August 28, 2005. On June 13, 2006, drifting buoy 26,028 ended its long-term data collection of sea surface temperature after transmitting for 10 years, 4 months, and 16 days, which is the longest known data collection time for any drifting buoy. The first weather buoy in the Southern Ocean was deployed by the Integrated Marine Observing System (IMOS) on March 17, 2010.

Key Applications

Measurements

Weather buoys, like other types of weather stations, measure the parameters that the conventional weather station measures, including:

- Air temperature above the ocean surface
- Wind speed (steady and gusting)
- Wind direction
- Barometric pressure

and unlike the conventional land weather stations, they measure more oceanographic parameters, including:

- Measure sea water temperature
- Sea water salinity density
- Wave height, and dominant wave period and wave direction
- Ocean/tidal current

Raw data is processed and can be logged on board the buoy and then transmitted via radio, cellular, or satellite communications to meteorological centers for use in weather forecasting and climate study.

Types of Buoys

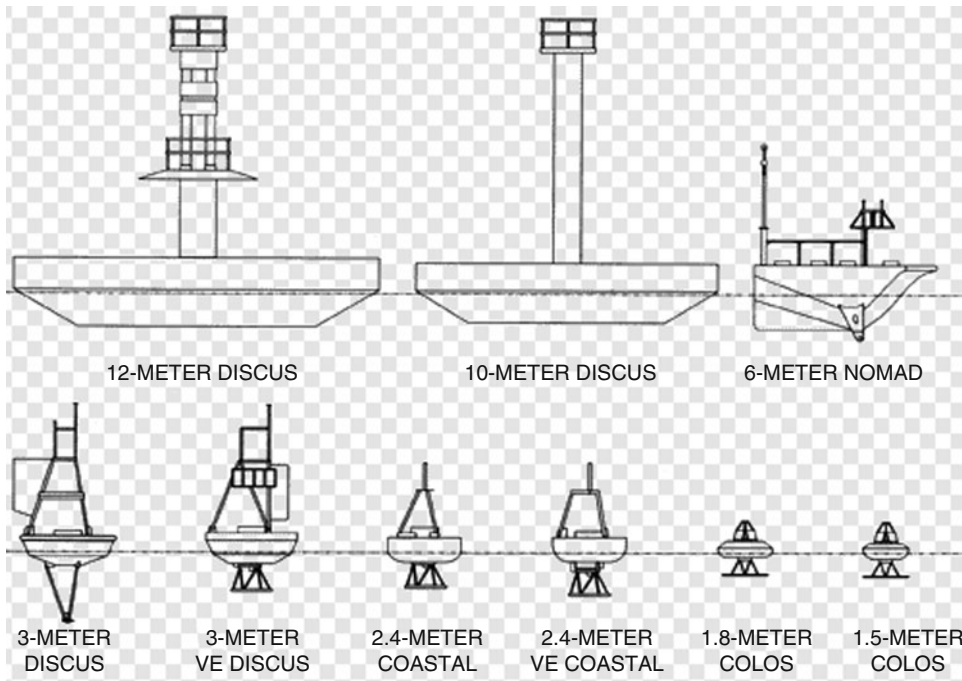
Moored Buoys

Weather buoys range in diameter from 1.5–12 m (Fig. 5). Those that are placed in shallow waters are smaller in size and moored using only chains, while those in deeper waters use a combination of chains, nylon, and buoyant polypropylene. Since they do not have direct navigational significance, moored weather buoys are classed as special marks under the IALA scheme, are coloured yellow, and display a yellow flashing light at night.

Discus buoys are large round and moored in deep ocean locations, with a diameter of 10–12 m. The aluminum 3-m (10 ft) buoy is a very rugged meteorological ocean platform that has long-term survivability. The expected service life of the 3-m (10 ft) platform is in excess of 20 years and properly maintained; these buoys have not been retired due to corrosion. The NOMAD is a unique moored aluminum environmental monitoring buoy designed for deployments in extreme conditions near the coast and across the Great Lakes. NOMADs moored off the Atlantic Canadian coast commonly experience winter storms with maximum wave heights approaching 20 m (66 ft) into the Gulf of Maine.

Drifting Buoys

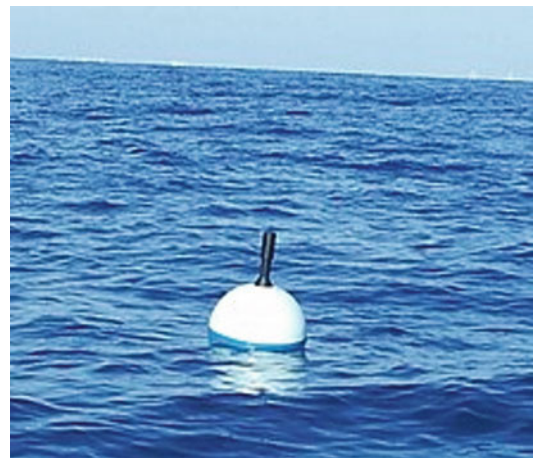
Drifting buoys are smaller than their moored counterparts, measuring 30–40 cm in diameter.



Meteorological Monitoring and Measurement Buoy, Fig. 5 Weather buoy operated by the National Data Buoy Center (NDBC)

They are made of plastic or fiberglass and tend to be either bicolored, with white on one half and another color on the other half of the float, or solidly black or blue (Fig. 6). It measures a smaller subset of meteorological variables when compared to its moored counterpart, with a barometer measuring pressure in a tube on its top. They have a thermistor (metallic thermometer) on its base, and an underwater drogue, or sea anchor, located 15 m below the ocean surface connected with the buoy by a long, thin tether.

Many different drifting buoys exist around the world that vary in design and the location of reliable temperature sensors varies. These measurements are beamed to satellites for automated and immediate data distribution. Other than their use as a source of meteorological data, their data is used within research programs, emergency response to chemical spills, legal proceedings, and engineering design. Moored weather buoys can also act as a navigational aid, like other types of buoys.



Meteorological Monitoring and Measurement Buoy, Fig. 6 Drifting buoy

TAO Buoy/Global Tropical Moored Buoy Array

The TAO/TRITON Array was designed to better understand and predict climate variations related to El Niño and the Southern Oscillation

(ENSO). ENSO, the warm phase of which we refer to as El Niño and the cold phase La Niña, represents the strongest year-to-year climate fluctuation on the planet. ENSO events significantly disrupt normal patterns of weather

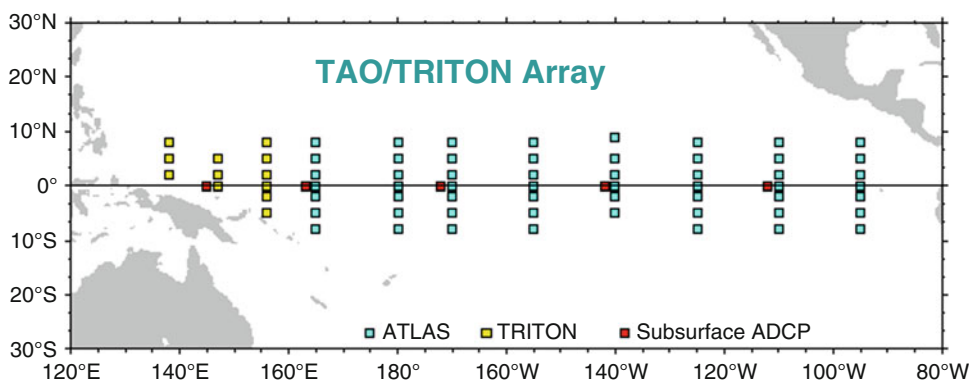
variability, affecting agriculture, transportation, resource management, energy production, and the lives of millions of people around the globe. ENSO also affects Pacific marine ecosystems and commercially valuable fish stocks such as tuna and anchovy. The project was motivated by the 1982–1983 El Niño event, and it took over the 10 year period 1985–1994 to deploy the buoys (Fig. 7), and the full TAO/TRITON system was built (Fig. 8) and is presently supported by the USA (NOAA/National Weather Service/National Data Buoy Center) and Japan (Japan Agency for Marine-Earth Science and Technology).

The TAO/TRITON array was extended to the Global Tropical Moored Buoy Array (GTMBA, see Fig. 9) after mid-1990, with the additions of the Prediction and Research Moored Array in the Tropical Atlantic (PIRATA), and the Research Moored Array for African-Asian-Australian Monsoon Analysis and Prediction (RAMA) in the Indian Ocean (McPhaden et al. 2009). Now the GTMBA is a global system with its interest spanning intraseasonal-to-decadal and longer timescales on global basis, including:

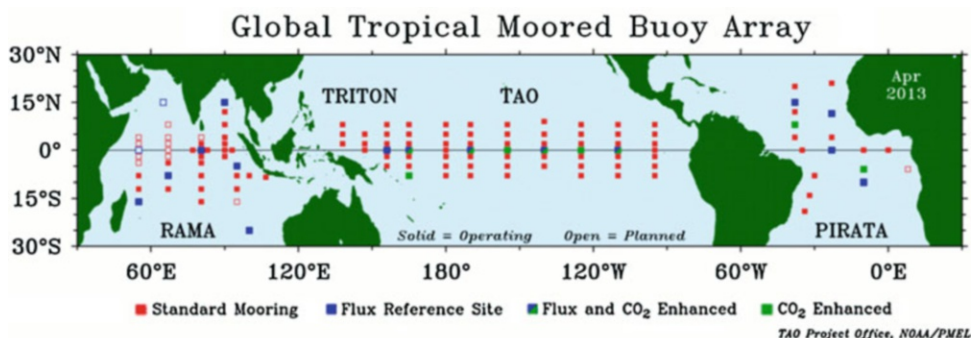
- El Niño/Southern Oscillation and its decadal modulation in the Pacific
- The meridional gradient mode and equatorial warm events in the Atlantic



Meteorological Monitoring and Measurement Buoy, Fig. 7 A TAO buoy



Meteorological Monitoring and Measurement Buoy, Fig. 8 The TAO/Triton array (<https://www.pmel.noaa.gov/gtmba/>)



Meteorological Monitoring and Measurement Buoy, Fig. 9 Global Tropical moored buoy array (<https://www.pmel.noaa.gov/gtmba/>)

- The Indian Ocean Dipole
- The mean seasonal cycle, including the Asian, African, Australian, and American monsoons
- The intraseasonal Madden-Julian Oscillation, which originates in the Indian Ocean but affects all three ocean basins
- Trends that may be related to global warming.

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Methanol

- [Alternative Fuel for Ship Propulsion](#)

Microplastic Particle

- [Microplastics](#)

Microplastics

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Synonyms

[Microplastic particle](#); [Micro-polymer particle](#);
[Micro-synthetic organic particle](#)

Definition

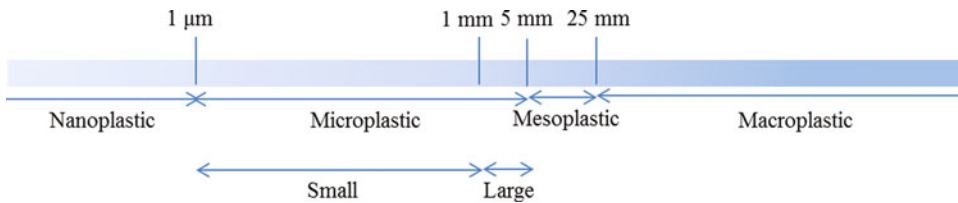
Plastics are polymers, which are chains of molecules derived from small monomer molecules that are extracted from fossil origin (crude oil, gas, etc.), renewable (sugarcane, starch, vegetable oils, etc.) or even mineral base (salt). Microplastics are usually defined as plastic fragments with sizes less than 5 mm.

Scientific Fundamentals

Plastic materials have been widely used in many applications since 1970. The common types of plastics include polyethylene (PE), polypropylene (PP), polystyrene (PS), polyvinylchloride (PVC), and polyethylene terephthalate (PET) (Table 1). The global production of plastics has increased rapidly over the past 60 years, reaching almost

Microplastics, Table 1 Plastic types and examples of items commonly found in marine debris

Type of polymer	Common examples of marine debris	Specific density
Polyethylene (PE)	Plastic bags, bottles, gear, cages, and pipes for fish farming	0.91–0.94
Polyethylene terephthalate (PET)	Bottles, strapping, gear	1.34–1.39
Polyvinyl chloride (PVC)	Film, pipe, plumbing pieces, containers, buoys	1.16–1.30
Polypropylene (PP)	Rope, bottle caps, gear, strapping, drinking straws	0.90–0.92
Polystyrene (PS)	Disposable cutlery, utensils, containers, packaging	1.04–1.09
Expansive polystyrene (EPS)	Bait boxes, floats, cups, expanded packaging	0.01–1.05
Acrylonitrile butadiene styrene (ABS)	Electronics and electrics, car interior	1.03–1.11
Polymethyl methacrylate (PMMA)	Architecture, instrument and meter parts, automobile lamp, optical lens, transparent pipe	1.15–1.19
Polyamide or nylon (PA)	Gear, fish farming nets, rope, toothbrushes	1.13–1.15
Polyurethanes (PU)	Packing of fragile goods, aviation, aerospace, automobile manufacturing, liquefied natural gas transport vehicle (ship) manufacturing	1.2–1.25
High density polyethylene (HDPE)	Cleaning product	0.91–0.94
Low density polyethylene (LDPE)	Preservative film	0.94–0.98

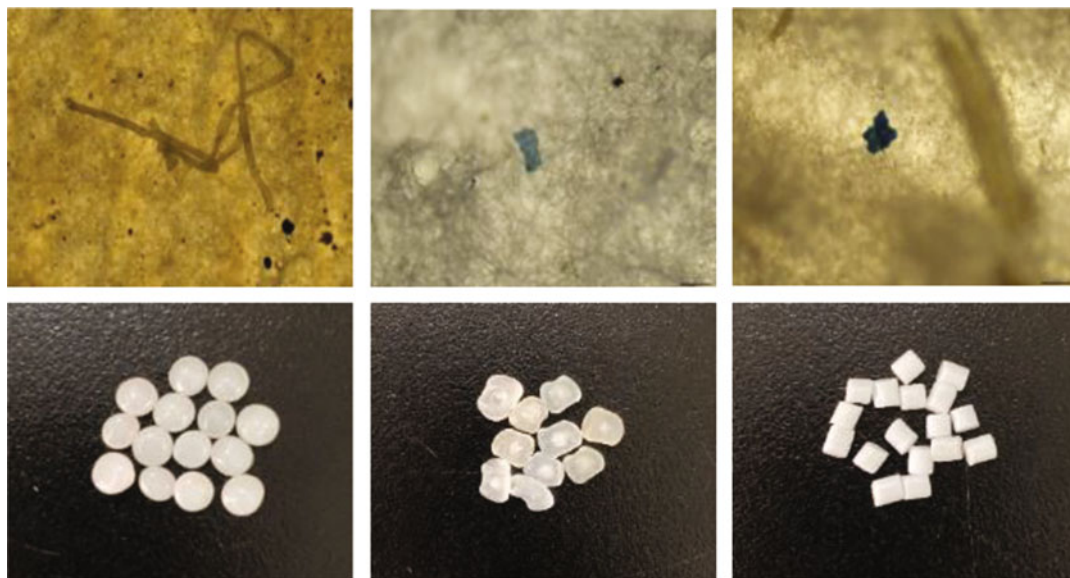


Microplastics, Fig. 1 Overview of relevant size ranges for plastic particles

360 million tons in 2018 (PlasticsEurope 2019). However, due to improper plastic disposal across the world, the majority of the plastic litter ends up in marine ecosystems transported via wind, river flow, and effluent discharge from sewage treatment plants. This results in global plastic pollution, particularly in estuarine and marine environments. Plastics in the sea are exposed to ultraviolet (UV) radiation, heat, oxidation, biodegradation, and mechanical disruption and eventually disintegrate into small debris (Shim et al. 2017). Microplastics are widely defined as synthetic organic polymers with an upper size limit of 5 mm (Fig. 1) (Lusher et al. 2013; Zhang et al. 2020), which are derived from the polymerization of monomers extracted from oil or gas (Thompson et al. 2009). These microplastics can be divided

into two categories according to their sources: secondary microplastics, as described above, and primary microplastics. The latter are produced intentionally as microbeads, microfibers, and in other forms (Fig. 2) (Dris et al. 2016) for specific domestic and industrial applications.

Since the 1940s, when plastics were mass-produced, the volume of plastics has increased rapidly. Consequently, the amount of plastic debris entering the marine environment rose in parallel with rates of production. Microplastics have been detected in aquatic systems almost all over the world, including freshwaters, oceans, polar environments, and even pristine mountain lakes or beaches of remote and unpopulated islands (Kettner et al. 2017). When one looks at vertical distribution, microplastics exist in bottom



Microplastics, Fig. 2 Microplastic particles with different shapes

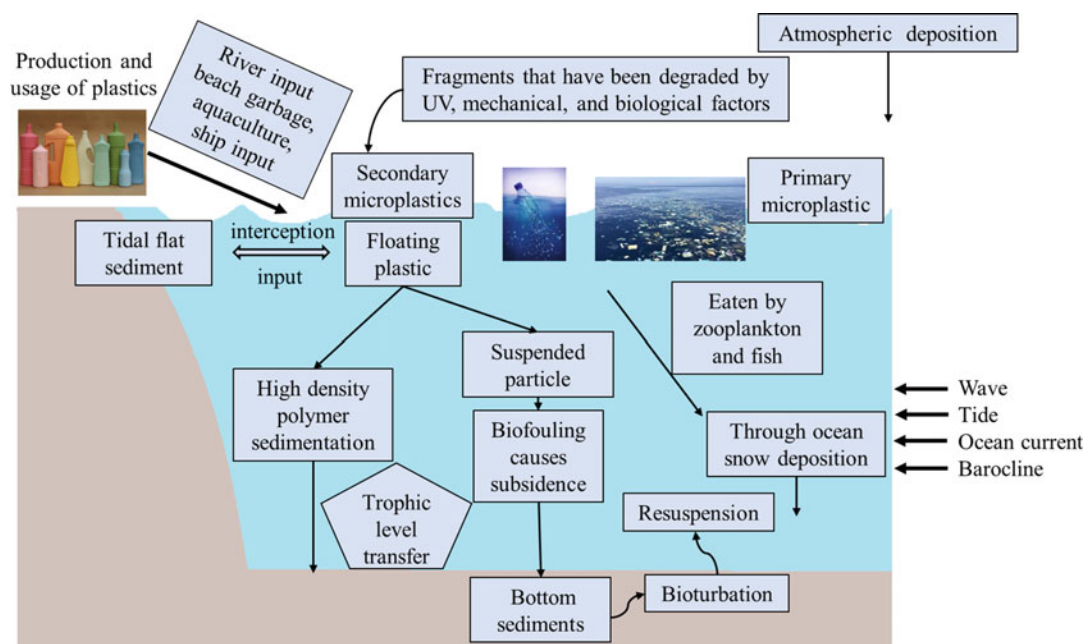
sediments (including abyssal and hadal deep-sea sediment), benthic zone of water mass, water columns, surface waters, and beaches. Numerous investigations have shown that microplastics have the highest abundance in sediment sample, and the values of abundances range from 1.3 to more than 3300 items kg^{-1} dry sediment. However, the abundances of microplastics in surface waters are some orders of magnitude smaller, and the values of abundances range from 10^{-5} to 10^5 item/ m^3 . Meanwhile, the distribution shows distinct geographical variations. The factors affecting the transportation and distribution include large-scale forces currents driven by wind, ocean currents, geostrophic circulation, and oceanographic effect. As pivotal factors, the inherent properties of microplastics (such as density, size, and shape) can also affect transportation and distribution modes (Fig. 3). During long-distance transportation via wind and turbulence, microplastics may serve as substrates for microflora in the ill-nourished open seas (Miao et al. 2019; Zettler et al. 2013). The microplastic surfaces are considered as novel habitats for microbial communities and termed as “plastispheres” (Zettler et al. 2013). Microbial life may facilitate the degradation of plastic waste (Yang et al. 2014) and

affect plastic debris density, buoyancy forces, and the sinking rate (Rummel et al. 2017). They may also influence the toxicity of microplastics (Imran et al. 2019).

Microplastics are small in size and may be mistakenly ingested by marine organisms, potentially resulting in physical or toxicological effects (Waller et al. 2017), and have a serial negative effect on the organisms throughout the food web (Lusher et al. 2013). Some studies have reported the existence of indigestible microplastics in deep-sea organisms in Western Pacific Ocean (Zhang et al. 2020) as well as in tiny shrimp living in the six deepest ocean trenches in the world (Peng et al. 2018). This suggests that the pollution by microplastics has already touched the deepest part of the world’s ocean.

Categories of Microplastics

Microplastics in the marine environment are composed of particles that are different in polymer type, size, shape, and specific density. They are widely divided into two categories: primary microplastics and secondary microplastics.



Microplastics, Fig. 3

Primary Microplastics

Primary microplastics are plastics produced for specific domestic or industrial applications with microscopic dimensions, including personal care consumer products (Fig. 4) and industrial or pharmaceutical products.

As for domestic utilization, microplastics are typically used in toothpaste, resin pellets, facial cleansers, and cosmetics like eye shadow (Zitko and Hanlon 1991). They were also applied to medicine as vectors for drugs (Patel et al. 2009). Virgin plastic production pellets, usually with diameter about 2–5 mm, can also be considered as primary microplastics, although their inclusion within this category has been criticized (Andrady 2011; Cole et al. 2011). Since microplastics have been approved for use in cosmetics in the 1980s, the use of exfoliating hand cleansers and facial scrubs containing microplastics has risen remarkably, and microplastic scrubbers have replaced customarily used natural ingredients (i.e., ground almonds, oatmeal, and pumice) (Derraik 2002; Fendall and Sewell 2009). The majority of facial cleansers now

contain polyethylene microplastics which are not captured by wastewater plants and will enter the oceans.

With regard to industrial applications of microplastic, the use of primary microplastics in consumer products constitutes millions of small sources, and the commercial or industrial use of similar products can be expected to constitute significant point sources of microplastics. From the microplastics-related reports, very few commercial-use products with primary microplastics are documented: abrasive blasting media for cleaning metal surfaces, abrasive hand cleaner soaps (Gregory 1996), and brief mentions of some unspecified use in petroleum industry.

Secondary Microplastics

Secondary microplastics, as the name suggests, are tiny particles of plastic derived from the breakdown of large pieces of plastics. The combined action of physical, chemical, and biological processes can weaken the structural completeness of plastic debris, leading to fragmentation

Microplastics, Fig. 4 The personal care consumer products containing microplastics



(Cole et al. 2011). Both the environmental factors (such as temperature, pH, salinity, and sunlight) and the properties of the microplastic (such as size, shape, and specific gravity) will act on the disintegration of large plastics. Macroplastics exposed to sunlight may suffer from light degradation, and ultraviolet (UV) radiation from the sunlight causes oxidation of the polymer matrix, resulting in bond cleavage (Andrady 2011; Moore 2008). Plastic debris on beaches is exposed to both high oxygen and sunlight, which will lead to rapid degradation. When the plastics losing the structural integrity were washed out to estuary and sea, they are cumulatively susceptible to fragmentation resulting from mechanical disruption (i.e., wave action, turbulence, and abrasion) (Barnes et al. 2009). In addition, the microbial activities and animal activities will also affect the fragmentation of macroplastics to microparticles.

Sources of Microplastics

The ocean-based sources of marine microplastics, resulting from commercial fishing, vessels, and other activities in marine environment, only contribute 20% of total plastic debris in marine environment (Andrady 2011). The discarded fishing net floating at sea may lead to

the continued trapping and killing of marine life, which is called “ghost fishing.” Products (such as clam netting, oyster bags, cages, and floats) applied in shellfish aquaculture can become marine debris when they are discarded or washed out during storms.

The microplastics from land sources contribute the remaining 80%. Land sources contain different origins that mainly are personal care products, air-blasting process, incorrectly disposed plastics, or leachates from landfill (Cole et al. 2011). A mass of marine microplastics are derived from land-based sources, such as floating garbage cans, litter tossed along roadsides, cigarette butts, food, agricultural plastic mulch, domestic sewage, beverage containers, and plastic straws; improper disposal; or illegal dumping. Once land microplastics are released into the natural water systems, most of them would be eventually transported to oceans by rivers, while the remaining would reside in freshwater environment, even including such isolated water systems as remote mountain lakes (Free et al. 2014). Light items, such as styrofoam, plastic grocery bags, and helium-filled balloons, can blow for miles in the air before landing in waterways.

At the land-water boundary, degenerating coastline structures (i.e., wharves, old piers, and bulkheads) will breakdown and finally become marine debris.

Sample Extraction

The small size (<5 mm) of microplastics makes it impossible to use direct visual selection methods for extraction. Concentrated collection methods or large sample collection methods are usually used to increase the possibility of microplastic detection, and it also makes it more difficult to separate and identify microplastics from collected samples, especially in sediments and organisms. Therefore, sample pretreatment and the method selection are essential for effectively extracting microplastics. The microplastic extracting process includes direct visual inspection, filtration, screening, and density flotation.

Separation of Microplastics from Sediments

Density flotation is commonly used to separate microplastics from sediments. It utilizes the density difference between the target components and impurities in the sample to achieve their separation and to float microplastics to the upper layer by using some intermediate media. The usual method is to add saturated salt solution to the sample to be extracted. The commonly used salt solutions include saturated NaCl solution, saturated NaI solution, saturated KI solution, saturated CaCl_2 solution, or saturated ZnCl_2 solution. Which salt solution to be used depends on the specific conditions of the experiment. Generally, the saturated salt solution is added to the sediment, stirred, and oscillated until it is fully mixed. The sediment-water mixture is allowed to be settled for 24 h until the stratification is obvious. The upper solution is transferred, and the microplastics are separated to the filter membrane by vacuum filtration. Due to the complex operation of artificial density flotation, if the number of samples is large, the experimental amount is large, time-consuming, and labor-intensive, and the loss rate is high. Therefore, researchers have designed a microplastic density flotation device for separating microplastics in sediments, which mainly relies on the principle of air flotation and continuous repetition to design, improve, or optimize the separation device to promote the performance of the device for extracting microplastics.

Separation of Microplastics from Organisms

The processing of marine organism samples is as follows: Collect samples, measure their size (using ruler) and weight (using electronic scale) after determining the species of samples, and then store them at $-80\text{ }^{\circ}\text{C}$. The samples are taken out and thawed before the analysis. In most cases, it takes nearly 12 h to thaw. After thawing, samples are rinsed with pure water for several times. Dissect the samples to obtain the desired biological tissue samples.

The key to separate microplastics from biological samples is to digest organic matter effectively without affecting the subsequent qualitative and quantitative analysis of microplastics. The biological sample treatment methods reported include acid digestion, alkaline digestion, and enzyme digestion. Acid digestion refers to the digestion of a sample with an acid solution. The commonly used acid solutions include HCl, HNO_3 , HClO_4 , and mixed acids (HNO_3 and HClO_4). Although the acid solution has a higher efficiency in digesting bioorganic matters, it will also digest some of the microplastics, resulting in microplastics, especially fibers, being underestimated. Similarly, alkaline digestion is the use of alkaline solutions to digest biological samples. The alkaline solutions NaOH and KOH were usually used, and the most commonly used mixed alkaline solution is NaOH and H_2O_2 . According to experimental data, the digestion rate of 1 mol/L NaOH on aquatic plankton can reach 90% at room temperature. The 10% KOH solution has high digestion efficiency and good digestion effect, and will not affect the microplastics recovered. Commonly used enzyme digestion types are trypsin and protease K. Because of the specificity of enzymes, enzyme digestion requires different types of enzymes for different biological tissues, so the requirements for enzymes are relatively high. In addition, the cost of enzyme experiments is also high, so few researchers use enzyme digestion methods. After digestion, saturated salt solution is also added for flotation treatment, and then the supernatant is transferred to the filter membrane.

Until now, there is not a uniform standard for the sampling and sample preparation for the microplastic analysis, which varies considerably

among researchers and would result in the variations of microplastic abundance for the same sampling location (Cai et al. 2018). Therefore, the standardization of sampling and microplastic extraction method is of vital importance for the comparison of microplastic abundance among different places.

Microplastic Identification and Quantification

Since the microplastics isolated from environmental samples are still a mixture of multiple substances, and the fragments of these mixtures have different characteristics in shape, size, color, density, chemical composition, etc., it increases the difficulty of microplastic separation and identification. Only by developing a specific methodological standard can comparable monitoring data be obtained and used to make reasonable assumptions about the distribution, quantity, and composition of microplastics. In the process of the study, the physical characteristics of microplastics are measured mainly by the visual method (i.e., the naked eye) combined with stereomicroscope to determine the shape, size, and color of microplastics. After visual selection, different instruments are used for detection. Due to the limitations of their own principles and characteristics, these instrument detection methods have certain advantages and disadvantages (Table 2).

The physical characteristics of microplastics mainly include its color, size, shape, and density. These characteristics will largely affect the migration and bioavailability of microplastics in the environment and will also determine the ecotoxicological effects of microplastics on marine organisms. According to researches, the shapes of microplastics mainly include fiber, pellet, particle, fragment, thin film, foam, etc., among which fiber is the most commonly detected shape. In most research reports, microplastics with a size less than 1 mm account for the largest proportion. The chemical characteristic index of microplastics mainly refers to the chemical composition of microplastics. At present, most of the research carried out generally uses Fourier transform

infrared spectroscopy (FTIR) and Raman spectroscopy (RS) to detect the polymer types of microplastics according to the comparison of their spectral results with the spectra library. Usually, only a polymer result with a matching degree greater than 70% can be considered as microplastics and recording its chemical composition. Using FTIR to identify, not only can we know the polymer composition, but also we can obtain information on the quantity of microplastics. Using this information, it is possible to speculate on the source, origin, and discharge path of the microplastics to a certain extent. The composition ratio of oxygen-containing bonds (such as hydroxyl, carbonyl, and other groups) in the infrared spectrum can also provide information about the weathering degree of microplastics. However, most of the microplastics in environmental samples have been weathered, and their composition is complex. They often need to be combined with multiple instruments to obtain a clear and accurate spectrum.

The Hazards of Microplastics

Microplastics can pose hazardous effects to the living organisms and environment due its chemical and physical properties, i.e., they are biochemically inert polymers, contain chemical additives, and could serve as vectors of pollutants and pathogens. The harmful effects of microplastics mainly include but not limited to the following:

Harmful Effects to Marine Organisms

Due to the small size of the microplastics in the ocean, it could be easily mistaken by aquatic animals and difficult to be eliminated from the body and eventually transferred to higher trophic organisms along the food chain. What's worse, microplastics of smaller size could easily enter the tissues and cells of vertebrates and accumulate in the body. Thus, microplastics in the body of living organisms could not only cause digestive system problems (intestine clogging, friction, penetration into the intestine wall, etc.) but also give rise to the problems of growth inhibition (reduction in feeding activity, malnutrition,

Microplastics, Table 2 Microplastic testing methods and analysis comparison

Detection method	Brief introduction	Sample preparation	Merit and demerit
Raman spectroscopy (RS)	Laser activation causes molecules to vibrate to measure molecular structure	Clean the sample	Advantages: no direct contact, no damage to the sample Disadvantages: The color of the sample will cause great interference to the detection, and sometimes the pigment map is obtained
Fourier transform infrared spectroscopy (FTIR)	Detect the vibration absorption of functional groups and chemical bonds, and compare with the library to get the polymer type	Clean the sample	Advantages: The interference of water vapor and carbon dioxide can be eliminated; the micro area can be scanned; the position of the sample can be visually seen and adjusted through the computer software; the sample is not damaged Disadvantages: If you want to obtain a higher degree of matching, the requirements on the sample are higher
Pyrolysis gas-liquid chromatography (Py-GC/LC)	The microplastics were decomposed by heat, and then the products were detected to infer the composition of the microplastics	The sampler shall be installed with a thermal desorption system	Advantage: It can analyze organic plastic additives as well as microplastic polymer types Disadvantages: damaged samples, complex equipment
Scanning electron microscope (SEM)	The surface morphology of the microplastic sample is obtained by electron beam interaction with the sample	Special coating protection, such as gold spraying, is required for samples under high vacuum conditions	Advantages: can get high-pixel images Disadvantages: Special coating is required; the cost is higher; damage to the sample
Environmental scanning electron microscope (ESEM)	The radiant energy of the microplastic surface is diffracted and reflected	/	Advantages: Obtain the image of the element composition and surface structure of the sample

M

starvation, etc.), enzymatic disruption, as well as the reproduction problems.

Harmful Effects to Human

The existence of microplastics in aquatic organisms, drinking water, table salt, sugar, and honey may become the sources of microplastics to human through their diets. Microplastics are able to pass through cell membrane and travel from gut cavity to circulatory system and subsequently induce digestive system problems, liver lipid disorder, inflammation effect, respiration problem, and even potential of cancer (De-la-Torre 2020; Yu et al. 2020).

Key Applications

Efforts for Controlling Microplastic Pollution

At present, the pollution problem of marine microplastics has become increasingly serious, and it has developed into a global pollution incident. Due to their enduring property, the effective treatment of microplastic pollution is lacking. The good news is the proposition and implementation of a “prevention first” strategy in lots of countries, starting from the management level to eliminate more pollution. Many countries have issued various plastic restrictions, for example, Chile has imposed a comprehensive plastic ban.

For reducing the use of plastic products, the Ministry of Environment of South Korea banned the use of disposable plastic cups in coffee shops from August 1, 2018. In the next year, the scope of “plastic limit” has expanded from coffee shops to supermarkets and roasting shops. Also, the “Law on Saving Resources and Promoting Resource Recycling” came into full force in South Korea in the same year. It stipulates that 2000 large supermarkets and 11,000 supermarkets with an area of more than 165 square meters completely stopped using the disposable plastic bags. In January 2020, the National Development and Reform Commission of China and the Ministry of Ecology and Environment issued the “Opinions on Further Strengthening the Treatment of Plastic Pollution,” which clearly sets the implementation goals into three stages in 2020, 2022, and 2025 to greatly reduce the use of plastic products. Plastic ban has been a global consensus and policy trend for environmental protection. In May 2019, the “EU version of the ban on plastics” came into full force, stipulating that the use of disposable plastic products, such as plastic straws, disposable tableware, and cotton swabs, will be prohibited by 2021. Moreover, the EU member states need to achieve the goal of recycling 90% of beverage bottles by 2029. The EU has also announced a ban on single-use cutlery, cotton buds, straws, and stirrers from 2021, as well as a reduction of other plastic items that are not included in the ban by at least 25% by 2025 in each member state. The “EU Plastics Act” also requires companies to take extended producer responsibility and economic responsibility for waste management and pollution control of disposable plastic products.

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Micro-polymer Particle

- [Microplastics](#)

Micro-synthetic Organic Particle

- [Microplastics](#)

Military Ships

- [Surface Warships](#)

MinDoc (Minimum Deepwater Operating Concept)

- [SPAR Platform](#)

Mineral Processing and Metallurgy

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Synonyms

[APAL](#) – Acid leaching process at atmospheric pressure; [HPAL](#) – Acid leaching at high pressure

Introduction

The seabed ore processing contains ore dressing and metallurgy.

The ore dressing is based on the physical properties of different minerals in the seabed ore and adopts the methods such as gravity separation, flotation, and magnetic separation to separate the useful minerals containing manganese, nickel, cobalt, and copper from the gangue minerals. Metallurgy is the extraction process of different valuable metals from seabed minerals by the chemical methods such as roasting, smelting, leaching, solvent extraction, and electrodeposition.

The seabed metal deposits which are discovered so far contain polymetallic nodules, cobalt-rich crusts, and polymetallic sulfides. The polymetallic nodules and cobalt-rich crusts are belonged to polymetallic oxide ore; their main mineral phases are iron-manganese oxide, and the valuable elements are nickel, cobalt, copper, and so on. In the polymetallic nodules and cobalt-rich crusts, there are no independent minerals of nickel, cobalt, and copper; most of them replace the elements Mn and Fe in iron-manganese oxides in the form of isomorphism and are distributed in different iron-manganese oxide phases. Part of them are adsorbed on the surface of iron-manganese oxide phase or form the colloidal precipitates with iron-manganese oxide phase. The elements nickel, cobalt, copper, and manganese cannot

be separated by cheap physical ore dressing methods, and they are always recovered by the direct metallurgy processes or by the combination process of ore dressing and metallurgy that could separate part of entrainment bedrock by ore dressing before metallurgy process. The metallurgical process for polymetallic sulfides is similar with that of the sulfide ore on land, namely, first, the concentrate of sulfide is obtained by ore dressing and then go into metallurgy process.

In addition, seabed minerals can be directly used or prepared for functional materials such as catalysis, adsorption, and energy storage.

Mineral Process

Overview

Because the cobalt-rich crusts are always attached on the surface of bedrock, it is an inevitable problem that the bedrock is mingled into cobalt-rich crust ore in mining process, which causes ore grade dilution. Before metallurgy process, the pretreatment by ore dressing is necessary to remove bedrock, which could improve the elements' grade such as nickel, cobalt, copper, and manganese, at the same time decrease the metallurgy economic cost. The common ore dressing methods contain gravity separation, magnetic separation, and flotation. Therein, flotation separation outcome is the best, and its removing rate of bedrock and recovery rate of metals are higher than that in gravity separation and magnetic separation. Comparatively speaking, gravity separation and magnetic separation are more suitable for removing bedrock offshore, while flotation is more suitable for cleaner flotation onshore.

Gravity Separation

Gravity separation is based on the difference in density, particle size, and the rate and direction of movement between the gangue mineral particles and the useful mineral particles containing manganese, nickel, cobalt, and copper in cobalt-rich crusts to carry out separation each other.

According to the density difference between cobalt-rich crusts and bedrock, a part of bedrock could be removed by gravity separation. If the raw ore of cobalt-rich crust contains Co 0.44%, Mn 16.24%, Cu 0.086%, Ni 0.38%, and CaO 11.81%, when the ore size is less than 1.0 mm, the concentrate obtained contains Co 0.47%, Mn 17.14%, Cu 0.089%, and Ni 0.40%, and their recovery rates are about 95%, 94%, 92%, and 94%, respectively. The removing rate of bedrock is more than 90%.

Magnetic Separation

Magnetic separation is based on the difference in magnetic permeability between gangue minerals and the useful mineral particles containing manganese, nickel, cobalt, and copper to carry out separation each other. In the same magnetic field, the iron-manganese oxide minerals containing nickel, cobalt, and copper with high magnetic permeability are sucked up by the disk as concentrates, but the gangue minerals with low magnetic permeability are not sucked up and remain in the raw material as tailings.

According to the magnetic difference between cobalt-rich crusts and bedrock, a part of bedrock could be removed by magnetic separation. If the raw ore of cobalt-rich crust contains Co 0.44%, Mn 16.24%, Cu 0.086%, Ni 0.38%, and CaO 11.81%, when the ore size is less than 1.25 mm and the magnetic intensity is 1592.3 kA/m, the recovery rates of cobalt, manganese, copper, and nickel are about 96%, 96%, 91%, and 96%, respectively, and the removing rate of bedrock is more than 95%.

Flotation

Flotation is based on the difference in the surface physicochemical properties between gangue minerals and the useful mineral particles containing manganese, nickel, cobalt, and copper to carry out separation each other. Under the alone or combination action of collector agent, foaming agent, regulating agent, and dispersing agent, the air is blasted into slurry and by means of bubble buoyancy; the useful minerals are separated from gangue.

If the raw ore of cobalt-rich crust contains Co 0.44%, Mn 16.24%, Cu 0.086%, Ni 0.38%, and CaO 11.81%, the closed flow flotation test was carried out at the optimal conditions: feed size -0.074 mm of 70%, regulating agent TH 2000 g/t, and collector agent BK485 7000 g/t; the concentrate contains Co 0.56%, Mn 20.41%, Cu 0.10%, and Ni 0.46%. The recovery rates of cobalt, manganese, copper, and nickel are about 94%, 94%, 83%, and 90%, respectively. For the sulfide ore containing more than Cu 0.8%, Pb 2.5%, and Zn 2.5%, the metal recovery rate for separated sulfide concentrate (copper concentrate, lead concentrate, zinc concentrate) is more than 87%, and the metal recovery rate for the mixed sulfide concentrate is more than 92%.

chemical reaction; at high temperature conditions, a series of physical and chemical changes occur in seabed minerals, and the valuable metals are separated from gangue or other impurities. The hydrometallurgy is the process that seabed minerals contact with a solution (acid, alkali, salt, etc.); the useful metals in the minerals are transferred into liquid phase by chemical reaction and then separated or enriched.

The main processes that are studied currently are shown in Fig. 1. Typical processes contain APAL process, Cuprion process, HPAL process, reducing calcining-ammonia leaching process, chlorination calcining-leaching process, sulfating calcining-leaching process, and smelting-leaching process.

Metallurgy Process for Iron-Manganese Polymetallic Oxide Ore

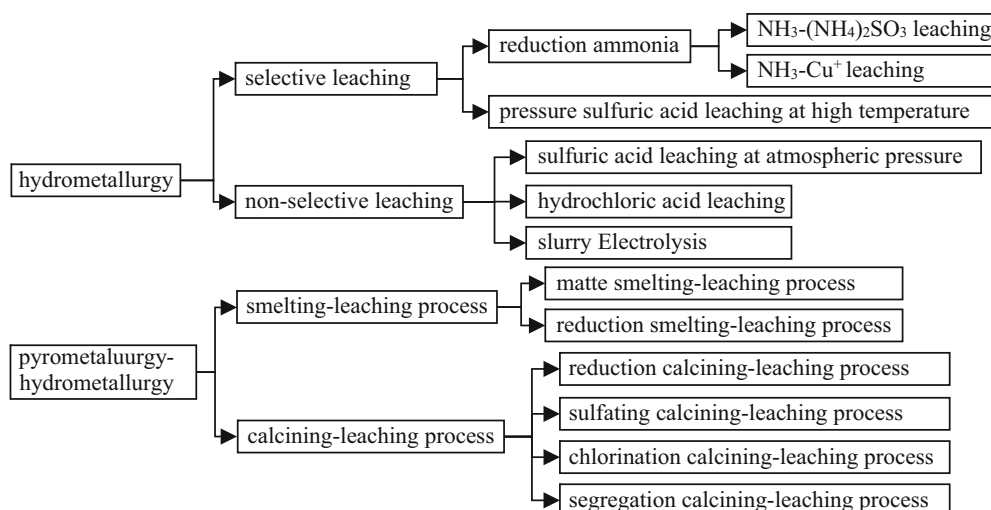
Overview

The metallurgy processes of iron-manganese polymetallic oxide ore are divided into two major types: pyrometallurgy and hydrometallurgy.

The pyrometallurgy is based on the heat utilization generated by fuel combustion heat or electrical energy heat or the heat released by a

APAL Process

In APAL process, by adding reducing agent in leaching process, the structure of iron-manganese minerals is destroyed, so the elements nickel, cobalt, and copper distributed in iron-manganese minerals are exposed to acid solution and then are leached. The common reducing agents contain metal sulfides (such as pyrite, hydrogen sulfide, copper matte, etc.), sulfurous acid and sulfite (such as H_2SO_3 , SO_2 , Na_2SO_3 , $(\text{NH}_4)_2\text{SO}_3$, etc.),

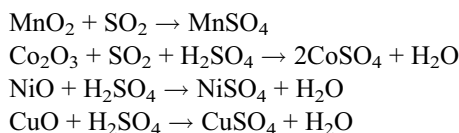


Mineral Processing and Metallurgy, Fig. 1 Main metallurgy processes for iron-manganese polymetallic oxide ore

H₂O₂, salts containing Fe²⁺, organic reducing agents, etc.

By controlling the potential and pH of leaching system, in theory, nickel, cobalt, and copper can be leached selectively, while the whole manganese and iron go in residue. Actually, due to the special structure of iron-manganese polymetallic ore, it is difficult to control all manganese in residue (Tan et al. 2004).

When gas SO₂ is taken as reducing agent, the main chemical reactions during leaching are as follows:



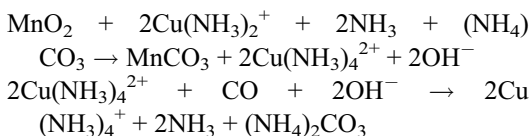
For the leachate, first, copper is recovered selectively by solvent extract with Lix84; the organic phase loaded copper is stripped with electrolytic waste liquid and then goes into electrodeposition operation; the cathode copper is obtained; for the raffinate of copper, after purification, the valuable element separation process is as follows: precipitation of Ni and Co with sulfide, sulfuric acid leaching of nickel-cobalt mixed sulfide at high pressure, solution purification, separation of nickel and cobalt with Cyanex272, and electrodeposition; finally the metallic nickel and cobalt are obtained. The solution from the operation unit of precipitation of Ni and Co with sulfide mainly contains manganese sulfate. By precipitation with ammonium bicarbonate and concentration crystallization, the products manganese carbonate and ammonium sulfate are obtained. The process diagram of sulfuric acid leaching at atmospheric pressure is shown in Fig. 2.

For the raw material cobalt-rich crusts, the metal recovery rates for the whole process are about Co 95%, Ni 96%, Cu 74%, Mn 98%, and Zn 82%. The consumption of SO₂ and H₂SO₄ are 0.26 kg/t ore and 0.19 kg/t ore, respectively. The final productions are metallic cobalt, cobalt oxide, cobalt oxalate, metallic nickel, metallic copper, and manganese carbonate (Yin et al. 1998) (Jiang et al. 1998).

Cuprion Process

For the Cuprion process, in the ammonium solution containing Cu⁺, the iron-manganese polymetallic ore is leached when the reducing agent CO is introduced. In leaching process, manganese enters the residue in the form of precipitation MnCO₃ that can be recovered as MnCO₃ concentrate by flotation or used directly as a metallurgical material for different manganese products. Nickel, cobalt, and copper are leached as ammonia complex. For the leachate, part of them return to the leaching operation unit for supplying copper and carrying out the self-catalytic; the others go into solvent extraction process for the separation of copper and nickel, and then by electrodeposition the cathode copper and cathode nickel are obtained. For the raffinate from solvent extraction, by precipitation with sulfide, the intermediate product cobalt sulfide is obtained, which is used as material for preparing of metallic cobalt (Agarwal et al. 1979). The main chemical reactions in leaching process are as follows:

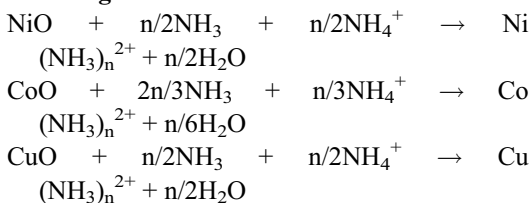
Reduction-Oxidation Reaction



Overall Reaction



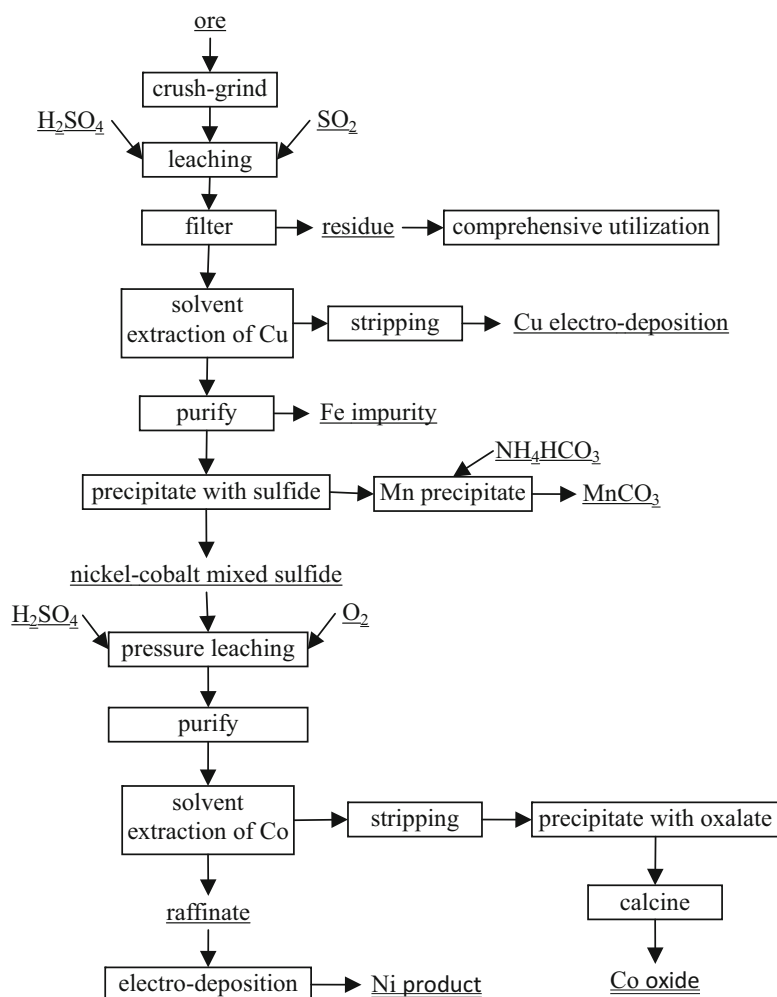
Leaching Reaction



For polymetallic nodules, the ammonium-ammonium sulfate system is used. At the conditions leaching temperature 45 °C and solution potential -400 mV to -450 mV, the leaching rates of nickel, copper, and cobalt are 98%, 97%,

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Fig. 2 Process diagram of APAL



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and 90%, respectively. The leachate contains copper ion 10–12 g/L, nickel ion 13–15 g/L, and cobalt ion 2–3 g/L. At the same time, molybdenum of 96% in ore is leached, which can be recovered comprehensively (Jiang et al. 2005) (Fig. 3).

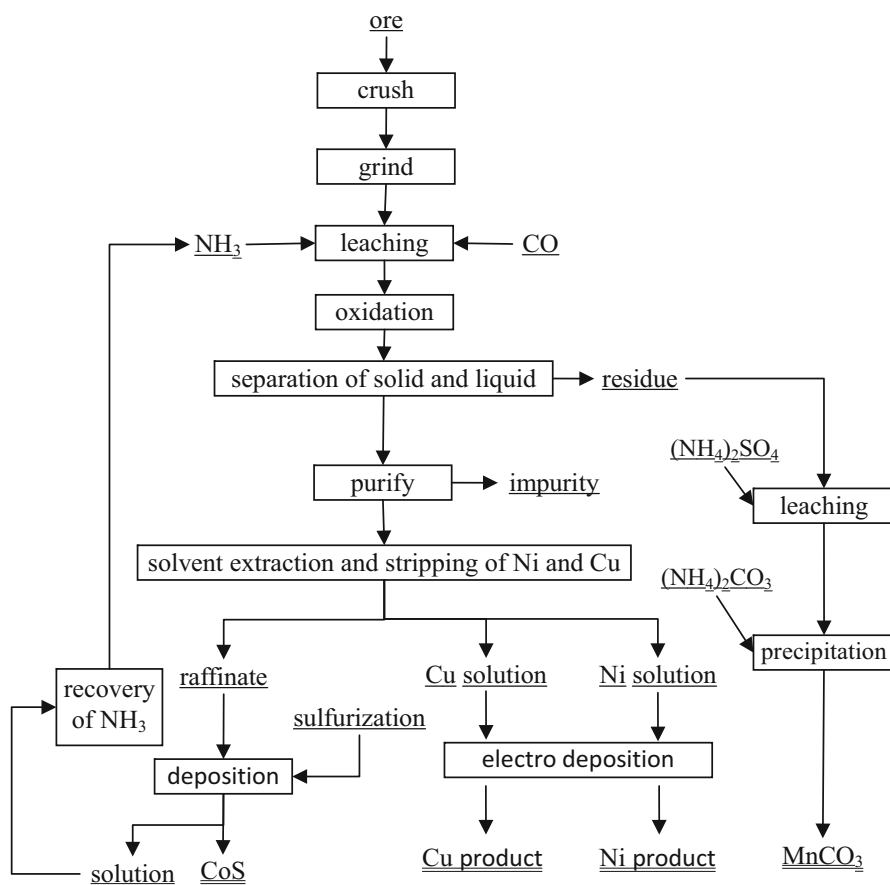
HPAL Process

The HPAL of iron-manganese polymetallic ore is similar to that of laterite ore. In leaching process, most of copper, nickel, and cobalt are dissolved, while iron and manganese rarely are leached. By solvent extraction and electrodeposition, metallic copper, cobalt, and nickel are obtained from leachate. At the conditions temperature 200 °C, pressure 3.1 MPa, partial pressure of oxygen

1 MPa, pH 1.63, and leaching time 1 h, the leaching rates of main metals in polymetallic nodules are Ni 80%, Cu 90%, Co 30%, Mn 5%, and Fe 2%, respectively. By solvent extraction with Lix64N, stripping with acid, and electrodeposition, metallic copper, cobalt, and nickel are obtained from leachate (Fig. 4).

Reducing Calcining-Ammonia Leaching Process

At 400–650 °C with different reducing agents, first, the high-valent iron oxides and manganese oxides in iron-manganese polymetallic ores are reduced into low-valent oxides; at the same time, the metal oxides of nickel, cobalt, and copper in minerals are reduced into metallic state and



Mineral Processing and Metallurgy, Fig. 3 Cuprion process

dissociated from iron-manganese polymetallic ores, and then the calcination is leached with the ammonia-ammonium carbonate solution which contains NH_3 50–100 g/L and CO_2 30–60 g/L; the leaching rates of metals are about Cu 85%, Ni 75%, Co 50%, and Fe less than 1%, respectively. For the leachate, by solvent extraction and electrodeposition, the metallic copper, nickel, and cobalt are obtained (Han et al. 1974) (Fig. 5).

Chlorination Calcining-Leaching Process

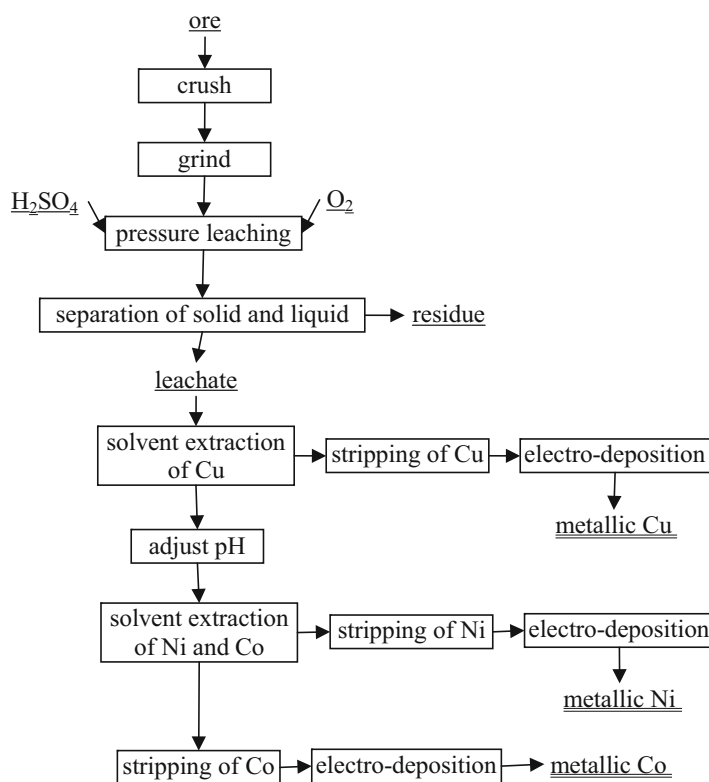
The chlorides of nickel, copper, cobalt, manganese, and iron have the properties such as low melting point, high volatility, and high water solubility; in chlorination calcining-leaching process, first, the iron-manganese polymetallic ore reacts with chlorinating agents at high

temperature, which makes the metal elements in ores convert into chlorides, and then they are recovered by hydrometallurgical processes. According to the type of chlorination agent, chlorination calcining is divided into the chlorination with hydrogen chloride and the reducing chlorination with chlorine salt (Cardwell 1973).

For chlorination with hydrogen chloride, iron-manganese polymetallic ore reacts with hydrogen chloride gas at 500–600 °C, and the elements manganese, nickel, copper, and cobalt form soluble chlorides, and then with water spraying at 300–400 °C, the ferric chlorides hydrolyze and convert into iron oxide; finally the chlorination product is leached with water and hydrochloric acid; the leaching rates of metals are about Ni 98%, Cu 99%, Co 97%, and Mn 99%,

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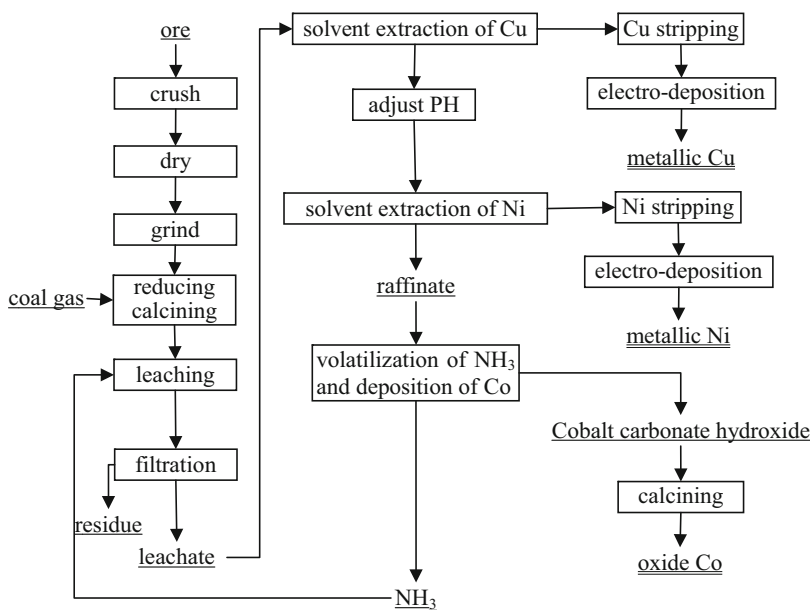
Fig. 4 Process diagram of HPAL



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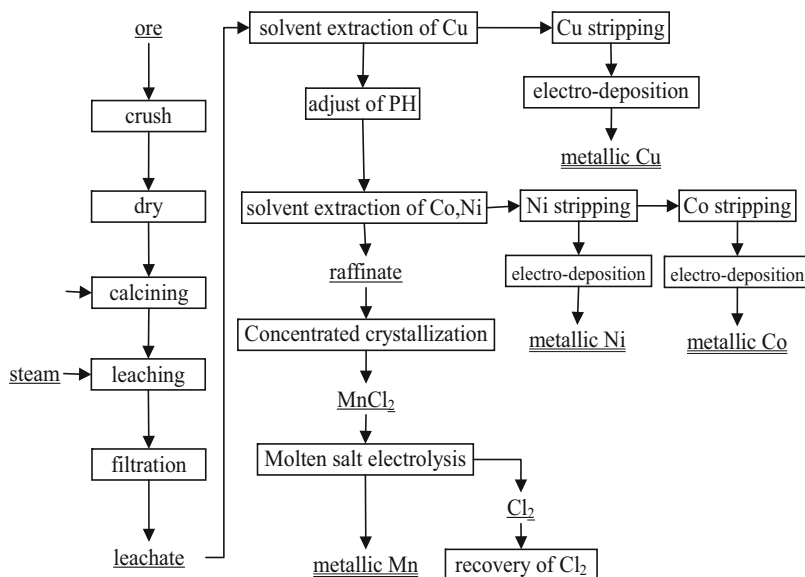
Mineral Processing and Metallurgy,

Fig. 5 Reducing calcining-ammonia leaching process



Mineral Processing and Metallurgy,

Fig. 6 Chlorination calcining-leaching process



respectively. In leaching operation, the iron oxide goes into leaching residue. The elements nickel, cobalt, and copper in leachate are separated by solvent extraction and electrodeposition; the products are metallic copper, nickel, and cobalt. For the raffinate containing manganese chloride from solvent extraction, by evaporation, crystallization, and molten salt electrolysis, the product metal manganese is obtained.

For reducing chlorination with chloride salts, the ferromanganese polymetallic ore, CaCl_2 , and coke are calcined at 900°C for 1 h. The metal chlorides are formed first and then reduced into metal by coke; after that by magnetic separation, the concentration containing nickel, cobalt, and copper is obtained from calcination. The metal recovery rates are Cu 46%–88%, Ni 23%–32%, Co 7%–22%, Mn 0–9%, and Fe 0–9% (Hoover et al. 1975) (Fig. 6).

Sulfating Calcining-Leaching Process

For sulfating calcining-leaching process, first, the ore is decomposed by sulfuric acid in calcining process, and the valuable elements such as manganese, nickel, cobalt, and copper are converted into independent salt sulfate and then separated further by hydrometallurgical process. When the iron-manganese polymetallic ore reacts with sulfuric acid or sulfur dioxide at 650°C , the

calcination contains the iron oxide and sulfate of nickel, copper, cobalt, and manganese. When the calcination is leached with water, the leaching rates of nickel, cobalt, copper, and manganese are 92%, 97%, 87%, and 99%, respectively. For the leachate, by replacement of copper with iron powder, precipitation of nickel and cobalt with sulfide, acid leaching of mixed Ni-Co sulfides at high pressure, and solvent extraction, the productions are obtained such as sponge copper, zinc sulfate, cobalt sulfate, and nickel sulfate (Meyer-Galow et al. 1973) (Fig. 7).

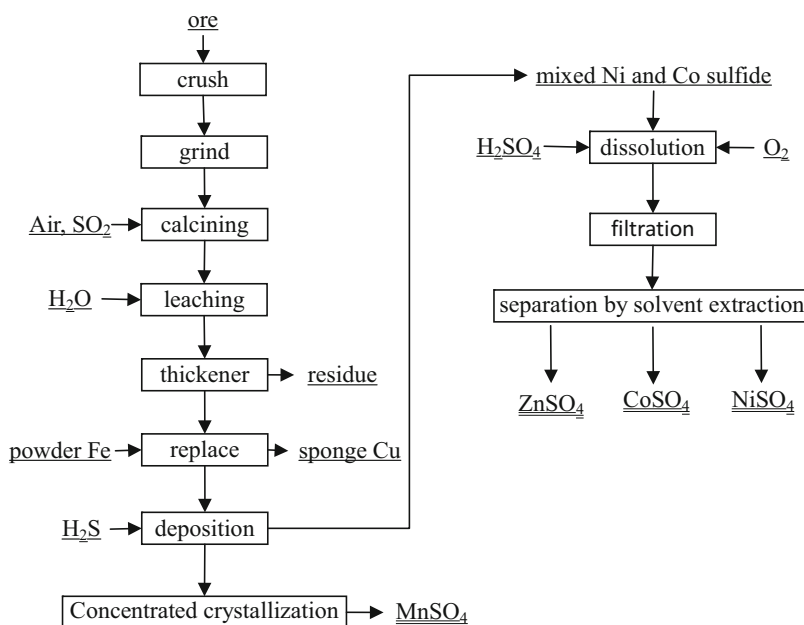
Smelting-Leaching Process

In smelting-leaching process, by controlling the smelting reaction temperature, amount of reducing agent or sulfite agent, the elements nickel, cobalt, and copper in iron-manganese polymetallic ore are reduced into the metallic state and then form the alloy or sulfide with iron and finally are separated by hydrometallurgy process. The manganese goes in the slag in smelting-leaching process that could be used as raw material for preparation of various manganese products.

Smelting-leaching process is divided into matte smelting-leaching process and reduction smelting-leaching process.

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Fig. 7 Sulfating calcining-leaching process



1. Matte Smelting-Leaching Process

In matte smelting-leaching process, the iron-manganese polymetallic ore is pre-reduced at 1100 °C with coal or fuel oil and then reduced into nickel-copper-cobalt-iron alloy at 1400 °C in electric furnace. The alloy ratio is 6%–10% which contains more than 90% of nickel and 75% of copper in iron-manganese polymetallic ore. The alloy is blown into high-grade matte with sulfide at 1350 °C in the converter. For the high-grade matte, by sulfuric acid leaching at high temperature and pressure, purification, solvent extraction, and electrodeposition, the metals such as cobalt, nickel, and copper are obtained. About manganese of 97% in iron-manganese polymetallic ore go into slag in smelting process and could be used as material for ferromanganese alloys (Sridhar et al. 1976).

2. Reduction Smelting-Leaching Process

In reduction smelting-leaching process, first, the iron-manganese polymetallic ore is pre-reduced with coke at 1100 °C and then smelted at 1400 °C with flux silica; the alloy is obtained which contains Ni 17.7%, Co 1.8%, Cu 9.6%, and Fe 67%. In smelting process, the recovery rates of nickel, cobalt, copper, and

iron are 93%, 68%, 87%, and 44%, respectively. For the alloy, by crushing, leaching, purification, solvent extraction, and electrodeposition, the products metallic nickel, cobalt, and copper are obtained (Richard Tinsley 1976) (Fig. 8).

Direct Application

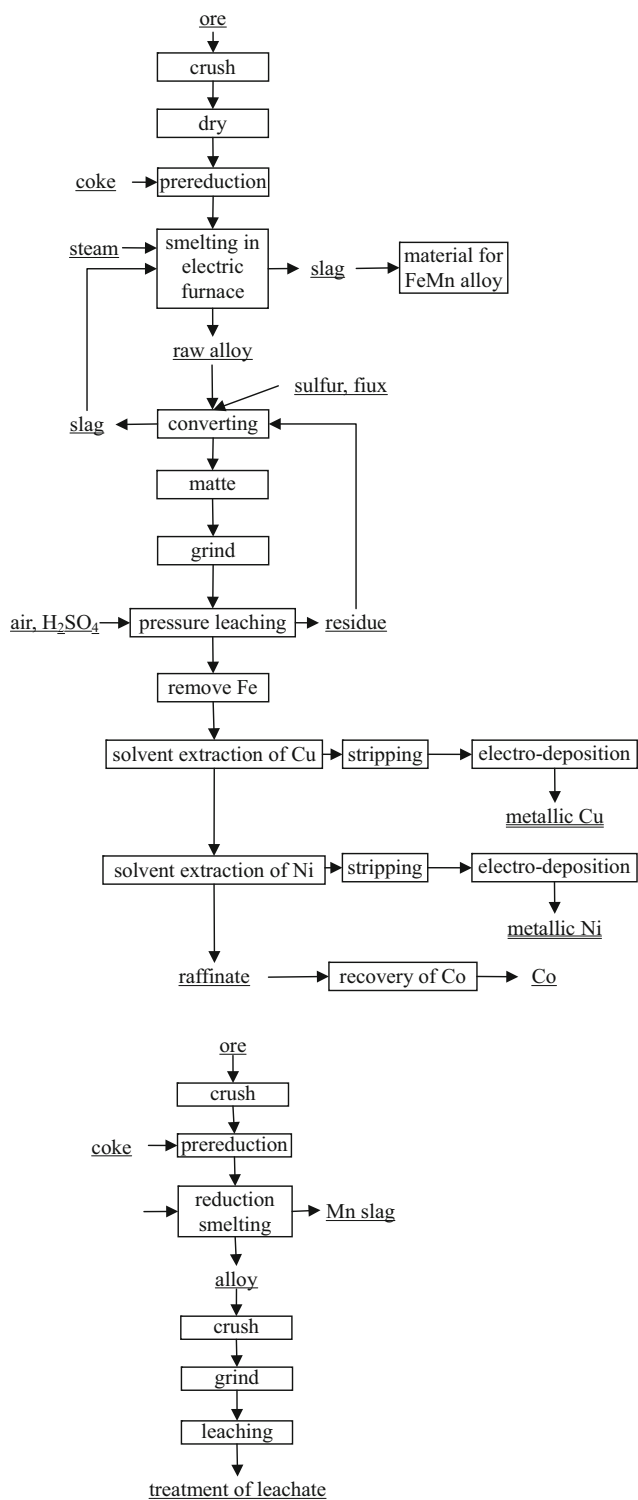
Direct application means that there is no mineral process and metallurgy process, but according to the iron-manganese polymetallic ore prospects such as layered structure, tunnel structure, much microscopic pores, high-grade transition elements, and active silicon, they are used or prepared directly for functional materials such as catalysis, adsorption, and energy storage.

Catalytic Active Material

The active elements Ni, Cu, and Co in iron-manganese polymetallic ore play a catalytic role. It has the catalytic function in the field of the hydromethanation of CO and CO₂; the oxidation of SO₂, CO, and some alkanes; and the decomposition of NO_x. In the preparation process of carbon nanotube by CVD, the iron-manganese

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Fig. 8 Smelting-leaching process



polymetallic ore is used as a natural catalyst to generate a large number of helical carbon nanotubes (Cheng et al. 2002).

Adsorption Material

The iron-manganese polymetallic ore has a selective adsorption effect on the elements in wastewater. The adsorption order for different elements is $Pb > Cd > Cu > Hg > Zn > Co > Mn > Ni > As$. The lead adsorption rate in wastewater is 90%~99% at pH 1.4~2.0, and the saturated adsorption capacity of lead can reach 200 mg/g ore. At pH more than 2.0, the adsorption rate of lead, zinc, cadmium, cobalt, copper, nickel, and manganese in wastewater can reach more than 99%, and the saturated adsorption capacity of zinc, cadmium, cobalt, copper, nickel, and manganese are all more than 40 mg/g ore (Tan et al. 2003).

Power Material

As a battery material, the iron-manganese polymetallic ores have good load characteristics, high discharge capacity, and good cycle stability characteristics. When the lithium-ion battery cathode material is prepared with iron-manganese polymetallic ore, the insertion potential of lithium ions is 2.6~3 V, and the actual reversible capacity of electrode reaches 110 mAh/g ore. There is still 70 mAh/g ore after 250 time cycle of charge and discharge (Yu et al. 2001).

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MIZ, Marginal Ice Zone

- [Numerical Simulation of Ice-Going Ships](#)

Mobile Offshore Base (MOB)

- [Mega-Float](#)
- [Station-Keeping System for VLFS](#)

Mobile Offshore Drilling Unit (MODU)

- [Polar Offshore Engineering](#)

Mobile Offshore Drilling Units (MODU)

- [Jack-Up Platforms](#)

Modal Superposition Method

- [Aquaculture Structures: Numerical Methods](#)
-

Mode Fibers

- [Fiber-Optic Cable](#)
-

Model Test

- [Experimental Investigation of Offshore Renewable Energy](#)
-

Modern Aquaculture Structures

Zhen Wang
DNV AS, Høvik, Norway

Synonyms

[Closed-containment systems](#); [Semi-closed containment system](#); [Submersible aquaculture structures](#)

Definition

Aquaculture production is growing and contributing in increasing volumes to global demand for marine foods. It is natural that fish farmers wish to expand and diversify their operations. Globally, the suitable land, fresh or coastal water, for fish farming has become more limited. Thus, fish farmers in many countries are considering expanding aquaculture production into locations that are further from the coast and where the conditions are harsher for fish, structures, and personnel.

Currently, the conventional gravity net cage is widely used for aquaculture production. The

gravity net cage system normally consists of four components: the float collar system, the net cage system, the sinker system, and the mooring system. The float collar provides buoyancy and the sinker system provides weight, maintaining a stable shape of the whole cage system. The net cage can significantly deform when subjected to ocean waves and currents. In moving the fish farm to exposed/offshore areas, environmental loads from wave, current and wind increase rapidly. Increased current loads will cause excess deformation of the net cage and might cause injury to the fish. Larger waves may cause fish to escape and challenges for operation and daily duties. In order to overcome these constraints on conventional fish farm, it is necessary to develop an offshore fish farm, which can be sited in high-energy environments.

Moreover, there are many problems with traditional fish farming including: accumulation of waste and feces on the seabed, disease, and fish escapes. Moving fish farms from the sheltered coastal areas to exposed sites leads to more space, stronger currents, and deeper waters. These can help to solve a number of these problems. Stronger ocean currents are particularly effective to dilute and disperse the fish waste and reduce the build-up wastes below fish cages. Sea lice settlement on fish is less successful under stronger currents and the impact of sea lice is therefore reduced. Greater distance to the shore also minimizes interactions with wild fish and results in negative impacts such as compromised reproductive health.

Given the legal and environmental restraints present in the aquaculture industry today, sea farmers have to look for new, innovative ways to increase volumes. The future will most likely be diversified – changing from one way of producing fish as seen today, to an industry approaching sea farming with several solutions.

Global Development of Aquaculture

The aquaculture industry has large regional differences. Regional variations are important in both the amount of fish farming activity and in

the diversity of farmed species. Norway, Chile, and Canada are leaders in the world for farming salmon, whereas China is the largest producer of farmed fish by volume. Some of the recent developments of offshore fish farm are listed in Tables 1 and 2.

Norway is a hub for offshore farming innovation due to its high value of its salmon farming. The Norwegian government has desired to have a significant growth in aquaculture. In 2015, Norwegian Ministry of Fisheries and Coastal Affairs introduced a free development concession for up to 15 years to promote new technologies that can solve environmental challenges and area issues facing the aquaculture sector. The aim is to support larger and more specific technology-oriented projects including prototypes, industrial designs, equipment in installations, and full-scale trial production. By the closing date 17 November 2017, the scheme had attracted 104 applications. Many applications are seeking to explore locations in more open waters than those presently used for salmon farming. The first project was accepted in 2016 and, as of today (May 2021), 22 projects have been granted for concessions. Two of these projects are described in greater detail below.

China is the largest player in the aquaculture industry. It is developing offshore salmon farms on an unprecedented scale with projects in the Yellow Sea, while the country's growing domestic appetite for other fish species will continue to drive nontraditional aquaculture practices. In 2016, the Ministry of Agriculture announced a "National Construction Plan for Marine Ranch Demonstration Zone (2017–2025)," which will build 178 pilot farms and increase China's aquaculture production by 30% by 2025. China's provincial governments are increasingly partnering with both privately owned and state-owned companies to build offshore finfish aquaculture farms.

New Technology in the Modern Aquaculture

Fish Health and Welfare

Farm-raised fish are susceptible to illness and disease. The welfare of fish is affected when

they are held in captivity as opposed to their natural habitat. Fish welfare is related to physical conditions at the aquaculture, such as fish density, temperature, water quality, disease, etc. Welfare issues may also be linked to other health problems that include dysfunctions related to poor feed or side effects of vaccines.

Good fish health is a precondition for fish welfare. In 2016 alone, 53 million salmon died in their pens in Norwegian fish farms, mostly due to sea lice (Veterinærinstituttet 2017). Improvement of the health status and welfare situation is crucial for future expansion of aquaculture industry.

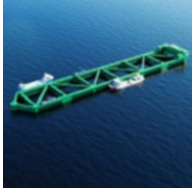
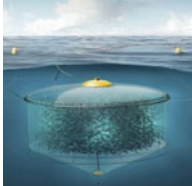
The aggregation of fish in cages increases the risk of infections and diseases. Aquaculture installations in open waters may help mitigate some risk factors. For example, greater distances between cages will reduce the risk of diseases spreading. However, fish farming under exposed conditions requires fish to cope with stronger waves and currents. Weather conditions should be monitored to ensure that they are within acceptable limits for the fish.

The stress from handling of fish for counting and delousing treatments of lice compromises fish welfare and increases mortality. Sound digital sea lice monitoring method, efficient delousing methods, and robust regulations will pave the way for fish farming under good welfare and sustainable conditions.

Going to Offshore

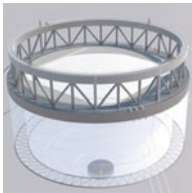
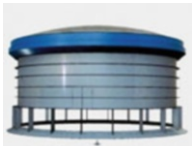
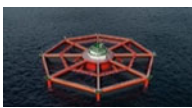
Within the fish farming community, a site-classification system is generally accepted. The site-classification system is based on wave energy of local wave climates, which is linked to distance from shore. In a report "Farming the deep blue" published at 2005 (Ryan 2004), four site classes were conceived: Class 1 sheltered inshore site, Class 2 Semi-Exposed inshore site, Class 3 Exposed offshore site and Class 4 Open ocean offshore site. Traditional surface-based aquacultures are mostly suitable for Class 1 and Class 2. Further development of aquaculture system suitable for Class 3 and Class 4 requires novel technology (Fig. 1).

Modern Aquaculture Structures, Table 1 Offshore development licenses awarded by the Norwegian Directorate of Fisheries as of 23/04/2021 (<https://www.fiskeridir.no/Akvakultur/Tildeling-og-tillatelse/Saertillatelse/Utviklingstillatelse/Kunnskap-fra-utviklingsprosjektene>)

No.	Company	Project Name	Concept	Status
1	Ocean Farming AS 	Ocean Farm 1	Independent unit Rigid steel structure Semi-submersible Permeable expanded nets Exposed locations	In operation at location 33757 Håbranden
2	Nordlaks Oppdrett AS 	Havfarm 1 & 3	Independent unit Ship-like open frame structure in steel Permeable nets Exposed locations #1: Permanently anchored #3: Dynamic positioning	Havfarm 1: In operation at Hadseløya
3	MNH Produksjon AS 	Aquatraz	Rigid steel structure Semi-closed facility Permeable bottom net Active power setting Existing locations	Cage #1–#3 in operation at Kyrøyene; Cage # 4 in operation at Kipholmen
4	AkvaDesign AS 	Semi-closed facility at sea	Floating collar in concrete Flexible closed net bag Semi-closed facility Sheltered sites Sludge collection	The concept is operated on the premises: 35,737 Seterrosen, 38,037 Andalsvågen, and 38,057 Hamnsundet
5	Mowi Norway AS, Hauge Aqua 	Egg	Composite material Closed facility Existing locations Sludge collection	Detail engineering in progress
6	Atlantis Subsea Farming AS 	Atlantis	Floating collar in plastic Continuously submerged production Permeable nets Partially exposed localities Underwater aeration	The first cycle at Gjerdinga and others at Skrubholmen

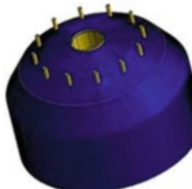
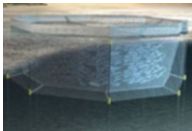
(continued)

Modern Aquaculture Structures, Table 1 (continued)

No.	Company	Project Name	Concept	Status
7	Norway Royal Salmon ASA 	Arctic Offshore Farming	Rigid steel structure Semi-submersible construction Continuously submerged operation Permeable nets Exposed locations Continuously submerged production Underwater aeration	Under construction
8	Hydra Salmon Company AS, SalMar ASA 	Production tank	Rigid steel construction, Semi-closed facility Permeable bottom nets Existing locations Passive power supply Existing locations	Detail engineering in progress
9	Cermaq Norway AS 	iFarm	Technology for increased resource utilization, fish welfare and lice control	Detail engineering in progress
10	Måsøval Fiskeoppdrett AS 	Aqua Semi	Semi-submersible semi-enclosed fish farm	Detail engineering in progress
11	Mowi Norway AS 	Marine Donut	Solid closed unit	Detail engineering in progress
12	Mariculture AS 	Smart fish farm	Comprehensive solution for the open sea	Detail engineering in progress
13	Nova Sea AS 	Spider cage		Detail engineering in progress
14	Stadion Laks AS 	Stadium pool	Closed floating pool	Detail engineering in progress

(continued)

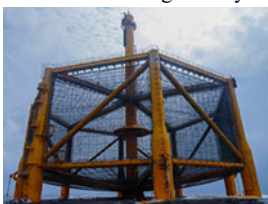
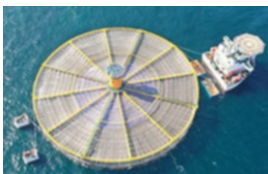
Modern Aquaculture Structures, Table 1 (continued)

No.	Company	Project Name	Concept	Status
15	Nekst AS 	Havliljen	“Sea platform” – submersible system	Within purpose
16	Salaks AS 	Fjordmax	Semi-closed integrated farming platform	Within purpose
17	Fishglobe AS 	Fish globe	Closed cage technology	Detail engineering in progress
18	Lerøy Seafood AS 	Pipe farm	Closed floating longitudinal current plant	Within purpose
19	Reset 	Reset	RAS facility at sea	Within purpose
20		Salmon Zero	RAS facility on land and production concrete tanks at sea	Within purpose
21		Blue Farm	Floating concrete cage collar with tension anchoring to the seabed	Within purpose
22		AquaBarge	A closed containment system based on RAS technology	Within purpose

Moving offshore aquaculture to more exposed water is an ongoing process, and the destination is to establish complete offshore installations in open rough waters. Farming in exposed areas requires new technical solutions

combined with operational concepts for maintaining security and ensuring production reliability. Knowledge and experience will be essential in the progression to full offshore aquaculture.

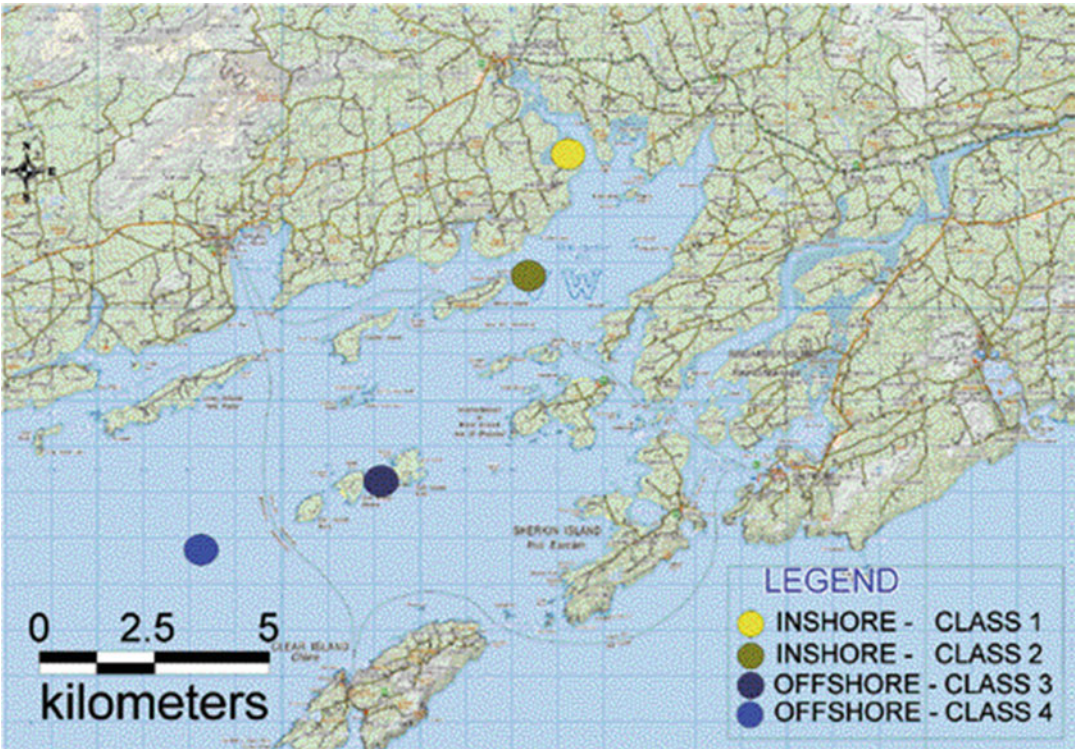
Modern Aquaculture Structures, Table 2 Recent development of offshore fish farm in China

No.	Company	Project Name	Concept	Status
1	Rizhao Wanzenfeng Fishery 	Deep Blue 1	Submersible floating truss fish farm (perimeter of 180 m and height of 35 m, culture volume 50,000 m ³)	In operation at 130 nautical miles to the Yellow Sea
2	De Maas SMC 	Hai Xia 1	Central column semi-submersible deep-sea fish farm (139 m in diameter and 12 m deep, culture volume 150,000 m ³)	In operation at East China Sea
3	Guangzhou Institute of Energy Conversion 	Peng Hu	Semi-submersible wave power fish farm (68 m long, 28 m wide and 16 m deep, culture volume 1000 m ³)	In operation at Zhuhai Wanshan
4	Zhuhai De-sail Marine Fishery Technology Co., Ltd. 	De Sail 1	Semi-submersible floating truss fish farm (91.3 m long, 30 m wide, and 7.5 m deep, culture volume 30,000 m ³)	In operation at Zhuhai Wanshan
5	Changdao HongXiang 	Chang Jing 1	Gravity-based fish farm (66 m in width and length, culture volume 60,000 m ³)	In operation at Daqin island of Bo Hai
6	Shandong Ocean Group 	Geng Hai No.1	Gravity-based marine ranch complex platform (3 net cages with diameter of 40 m, culture volume 27,000 m ³)	Installed at 1 km away from the coastline of fisherman's wharf in Laishan district of Yantai

(continued)

Modern Aquaculture Structures, Table 2 (continued)

No.	Company	Project Name	Concept	Status
7	Shanghai Zhenhua Heavy Industries 	Zhen Yu No. 1	Deep-sea automatic rotating marine fish farming platform with rugby shape (60 m long, 30 m wide and draft 1.2–1.5 m, culture volume 13,000 m ³)	In operation at Ding Hai Wan, Lianjiang, Fuzhou city



Modern Aquaculture Structures, Fig. 1 Four site classes (Ryan 2004)

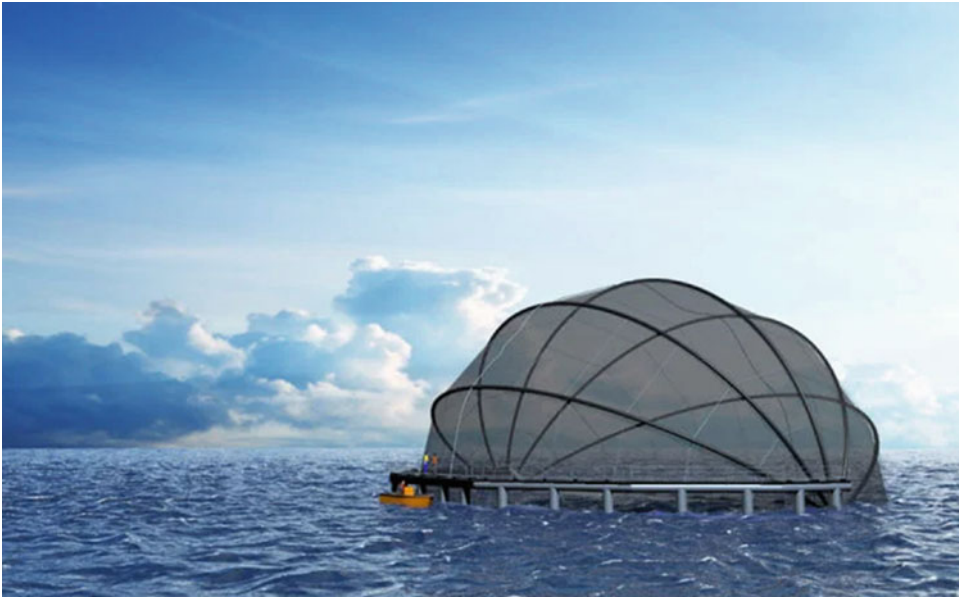
Submersible (Semi-Submersible) System

A submersible cage is a cage that can be lowered below sea level for a certain period, usually during storms. Some types are always partly under surface, where maintenance and normal operation are carried out at surface.

There are many advantages by lowering the facilities and keeping the fish further below sea

level. Wave forces on facilities are much smaller when below the sea level and risk from boat traffic and floating objects is reduced.

A new design for submersible cages was emerged in 2004, such as 40,000 m³ Ocean Globe (Fig. 2) from Byks of Norway (Fish Farming International 2004), but few of them have been built and tested in field at full scale. AKVA



Modern Aquaculture Structures, Fig. 2 Ocean Globe designed by Byks of Norway. (Source: <https://www.byks.no/>)

Group and Egersund Net have developed a concept for submersible cages called “Atlantis Subsea Farming.” The submersible cage must be submerged in the sea as far as possible and as little as possible in surface so that salmon lice will not become a problem.

Salmon farmers in Norway are also looking to the knowledge transfer from Norway’s offshore oil and gas industry to develop semi-submersible fish farm, which can move operations further offshore where environmental conditions are best suited for the growth of fish stocks.

Semi-submersible structures are designed to have small water-plan area, which is less affected by wave loadings. Larger buoyant tanks (pontoons) below the ocean surface can be ballasted/de-ballasted to match payload and free-board needs, as well as raise platforms from a deep to shallow draft. Salmar’s Ocean Farm 1 has now been operating for several years and utilizes such a concept to enclose a pen environment within its vertical “columns.” There are also a couple of projects that are under development, for example, “Havfarm 1,” “Aqua Semi,” “Arctic fish farm,” and “Smart fish farm”.

Closed and Semi-Closed System

Closed-containment systems (CCS) and semi-closed containment system (S-CCS) tackle some of the challenges with offshore farming. Placing closed or semi-closed system will enable to prevent escapes and sea lice as physical barriers.

As the facilities are closed, the feed will not be picked up and transferred out of facility by currents. Hence, the feed conversion ratio is decreased. Closed facilities need to process the water moving in and out of the facility. Water will be pumped up from 20–30 m below the sea level, where sea lice are not able to live. Moreover, closed facilities enable collection of sludge further reducing sea farming’s environmental footprint.

Many players such as Lerøy, Stadion Laks, and Mowi have chosen to focus on completely closed system, and the most famous of these projects is the “Egg” of Mowi. An egg-shaped farm is used to avoid lice and escape of fish, at the same time waste is collected at the bottom of the “Egg.” Mowi has also developed another closed concept, “Marine Donut,” which is a closed facility shaped like a donut.

Several companies such as Midt-Norsk Havbruk, Måvasøl Fiskeoppdrett, Nova Sea, AkvaDesign, and Hydra Salmon Company focus on partially closed facilities.

Recirculating Aquaculture Systems (RAS)

The use of recirculating aquaculture systems (RAS) will play an important role in the future aquaculture industry. Anadromous fish like salmon and trout are typically raised in fresh water until they are mature enough to migrate to salt water, where they are farmed in sea cages. With RAS technology these fish might spend their entire life on land by alternating fresh and saltwater environments through controlling the water chemistry.

Currently, sea farmers are looking into producing post-smolt of up to 500–1000 g or some even larger in land-based farming in order to decrease mortality and increase the number of production cycle (Ernst & Young 2019). RAS system can help to produce post-smolt with a good condition for better survival in sea when transferred to the open sea cage.

RAS technology can reduce water consumption and use lower energy for heated water during winter and spring seasons. Recirculation systems use 100 times less water per kilo of fish than traditional land-based systems. In addition, the water quality can be monitored continuously, which lessens the risk of disease and the need for antibiotics.

Denmark is a leader in recirculation system aquaculture. Hallundbæk Dambrug is using RAS technology to produce rainbow trout with recirculating over 96% of water (<https://stateofgreen.com/en/partners/akva-group-denmark/solutions/fish-farm-hallundbaek-dambrug/>). The discharged water is filtered, and the sludge is used for biogas or fertilizer. The discarded water is treated for removal of nitrate.

The carbon footprint is predicted to increase as fish farms move offshore due to increased energy use for transportation of materials, feed, and cultured fish. Facilities with RAS technology can be placed closed to the end market, reducing the transportation costs and carbon footprint.

Digitalization

Digitalization has the potential to simplify offshore operations and aquaculture is focusing extensively on digitally based technologies and solutions.

In Norway, a hub for offshore farming innovation, Fishtalk (AKVA Group) and Mercatus (ScaleAQ) are the main registration and reporting tools for production data. The aim of digital solutions is to reduce mortality rates, increase efficiency of farm inputs such as feed and vaccines, prevent disease and lice infection, and optimize farm management.

Even more interactive tools based on cloud services are anticipated for the future operation of large-scale offshore farm, providing real-time structural monitoring, complex analyses, and early warning system based on real-time processing of measurements. Cloud services can help to easily access relevant data and exchange the knowledge output and response process.

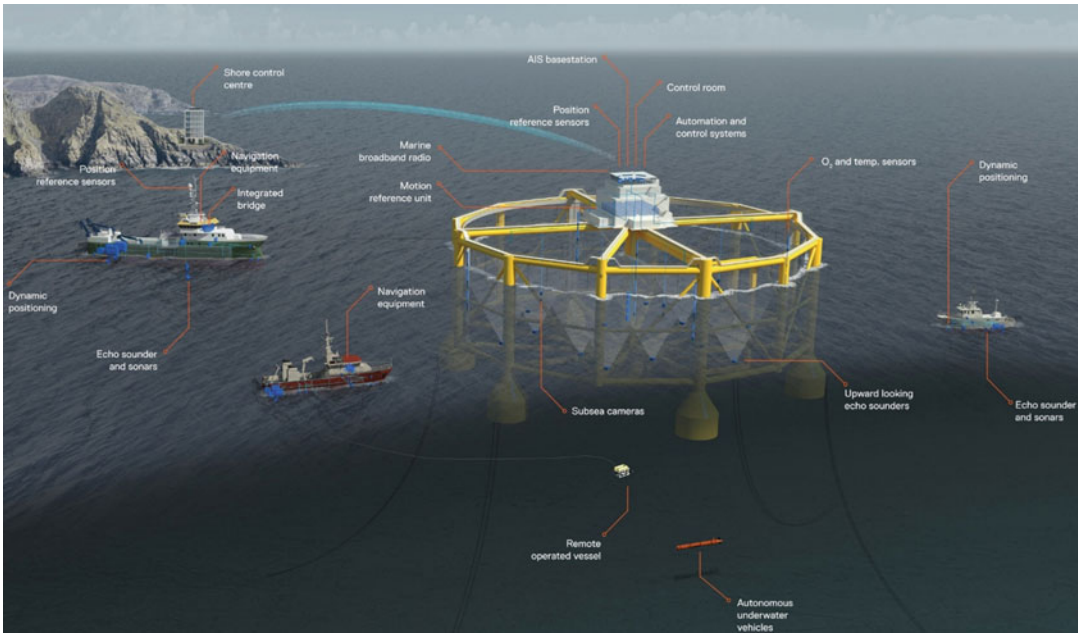
Digitalization has been promoted in application to a free development concession from Norwegian Ministry of Fisheries and Coastal Affairs. Some examples of different projects focusing on digital solutions are: Salmar's Ocean Farm 1 and Cermaq's iFarm.

Ocean Farm 1 is the first in the world to combine marine engineering, marine cybernetics, and marine biology via a "big data" approach fusing all the available underwater sensors and in this way offer decision support systems for the operators controlling and monitoring the feeding of the salmon and the overall physical environment of the sea (Fig. 3).

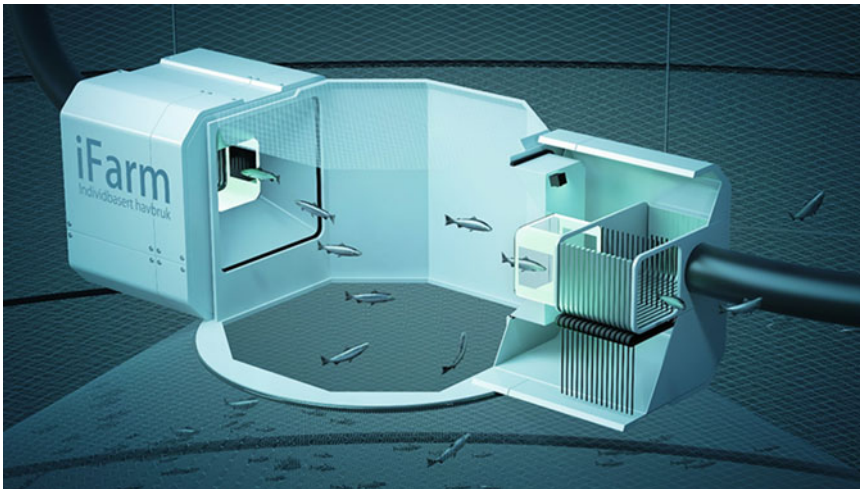
iFarm is a technology focusing on farmed fish health and welfare and recognizing fish as individuals – it thus has the potential to revolutionize fish farming. This allows us to monitor factors including growth, sea lice, disease, lesions, and other aspects that affect the health and welfare of the individual fish (Fig. 4).

Automation

Automation will be a strong contributor to efficient fish farming in the future. Automation can minimize the requirement for people in high-risk operations and can enable remote control.



Modern Aquaculture Structures, Fig. 3 Kongsberg digital farm. (Source: <https://www.kongsberg.com>)



Modern Aquaculture Structures, Fig. 4 iFarm from Cermaq. (Source: <https://www.cermaq.com>)

Based on technology development, new innovations and individual business goals, it is likely that the intensification of technology used in aquaculture will further strengthen the safety of the employees.

One application is the shift from on-site to a more off-site feeding control regime, where the

employee is located on a nearby safe feed barge or onshore. The feeding is carried out using a central feeding system involving video cameras and computers. This approach reduces the potential for incidents, during harsher weather conditions.

The other application is autonomous Remote Operated Vehicle (ROV). The equipment is being

developed with the intention of carrying out net integrity (check for holes) and net cleaning. Development and implementation of future autonomous vehicles for carrying out various operational aspects at cage site, is likely to lower the requirement for physical constraint and thereby enhance the safety of the employees.

Salmar's Ocean Farm 1 is one example of operation focusing highly on applying advanced technology for farming purposes. Ocean farm1 is fully automated and equipped with an integrated high-tech sensor and measuring system. The entire facility can be run by a crew of 2–4 people only and heavier manual operations are avoided. Therefore, all farming operations can be managed either onboard or remotely, minimizing the use of service vessels and outside equipment, making the entire facility more environmentally friendly.

Pioneer Projects for Offshore Fish Farm

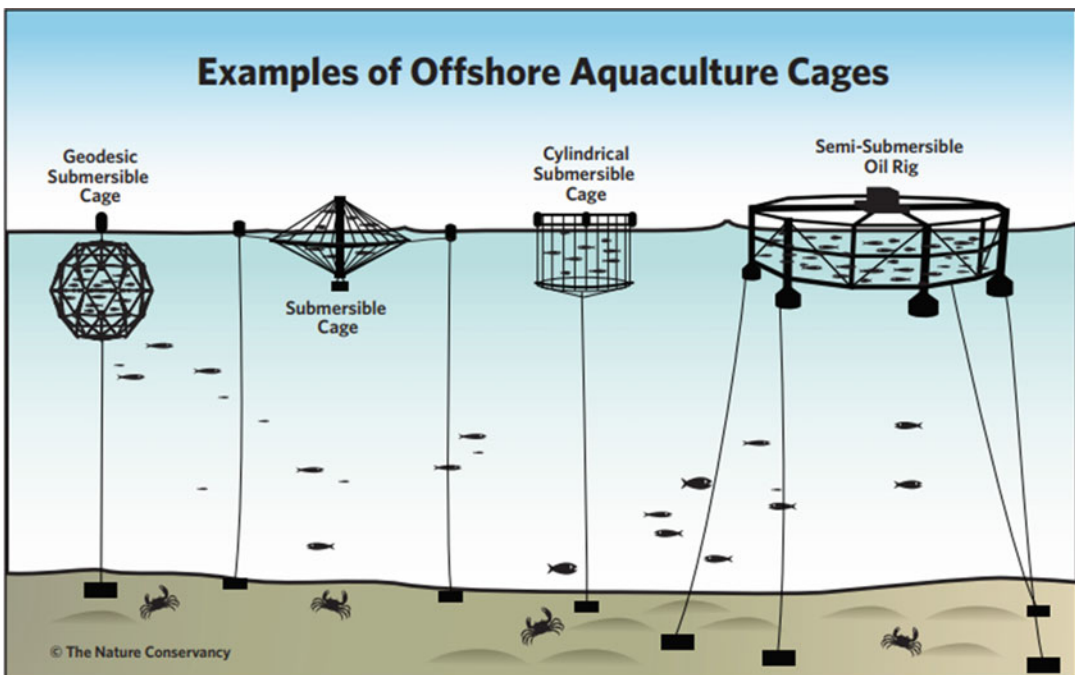
Offshore fish farms operate in the open ocean far from the coast. The offshore fish farm should be

robust enough to resist heavier environmental condition, especially during storm events. Off-shore fish farms in Norway have been designed with experiences from offshore oil rig technology.

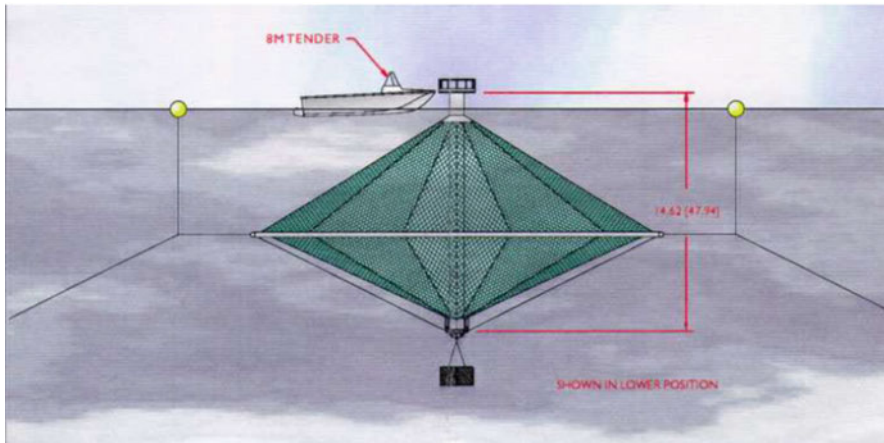
Fully enclosed cage is normally used for off-shore fish farm, which can be tethered or moored to the seabed. The offshore fish farm can be either partly or fully submerged in the sea in order to reduce the loadings from waves. Several designs have been developed in the world today, which is either for commercial production or experimental pilot unit with a small scale. Some recent pioneer projects in commercial production in Norway and China are introduced in the following sections (Fig. 5).

Sea Station

InnovaSea has developed a semi-rigid submerged net pen, "Sea Station (Fig. 6)." The system has a double cone form and includes a single central steel tube vertical spar inside a circular tube rim. These two elements are joined by radiating frame lines and the netting is fitted around this framework.



Modern Aquaculture Structures, Fig. 5 Offshore Aquaculture Cages (O'Shea et al. 2019)



Modern Aquaculture Structures, Fig. 6 Sketch for concept of Sea Station (Ryan 2004)

The central spar is used to provide buoyancy. The steel rim is to maintain net's shape and has ballasting capacities. In harsh weather, the cage can be fully submerged by varying the buoyancy in the central spar. A small platform on top of the central spar allows for feeding, access, and monitoring.

The system is moored at the central spar, allowing either for single point or fixed configurations.

The standard production model of Sea Station has a volume of 3000 m³ and the volume could be increased to 6000 m³ for larger version. 20 Sea Stations have been installed at Open Blue Farm, which is 11–12 km (7 miles) off the North coast of Panama. Currently 2000–3000 mt of cobia (*Rachycentron canadum*) can be produced each year (Fig. 7).

Aquapod

Marine biologists at Ocean Farm Technologies (Now InnovaSea) have developed a spherical fish cage, “Aquapod,” (Fig. 8) which can be carried on the high seas. The system can be fixed on deep sites or drift according to currents.

The cages consist of a large number of triangular galvanized steel frames. Their assembly makes it possible to obtain spheres measuring 8–28 m in diameter (capacity of 115–11,000 m³). Each triangle consists of a frame containing a polyethylene net, made of 80% from recycled products, covered with a

geometallic coating (<https://en.how10.com/1529763-the-aquapod-an-amazing-concept-of-marine-farms-on-the-high-seas>). The cages can be used on the surface or at depth, both on the coast and offshore. The triangles weigh between 40 and 50 kg in the air, but their weight vanishes once in the water. These spheres can be immersed with ease, allowing them to escape the action of waves, for example, in case of storm, or to protect them from any risk of collision with floating devices such as ships.

The Aquapod is firstly being trialed in Hawaii in a research project called Velella. Now 34 Aquapods are currently deployed in Puerto Rico, Indonesia, Panama, Florida, and Hawaii. They are mainly anchored, using long cables, in deep water with water depth between 200 and 300 m. In the future, Aquapods could potentially be equipped with propellers and a GPS system, and used to transport fish to arrive at their destination with the fish ready to harvest (Fig. 9).

Ocean Farm 1 and Smart Fish Farm

The world's first offshore fish farm with a semi-submersible design is already operating off Frohavet in Norway since September 2017. Ocean Farm 1 (Fig. 10) is owned by Norwegian owner, SalMar and was built in the China Ship-building Industry Corporation (CSIC) in Qingdao, China.



Modern Aquaculture Structures, Fig. 7 InnovaSea's Sea Station at Open Blue Farm. (Source: <https://www.openblue.com>)

SalMar has developed the world's first offshore fish farm from 2012 to 2015. In 28 February 2016, the Norwegian Ministry of Fisheries and Coastal affairs awarded first development licenses for aquaculture purposes to Ocean Farm 1 (Ocean Farming 2019). SalMar has invested approximately NOK 1 billion to design and develop Ocean Farm 1.

Ocean Farm 1 is an anchor fixed structure (diameter 110 m, height 68 m, and volume 250,000 m³) floating steadily in the ocean at water depths of between 100 and 300 m. It has a cylindrical net cage including two closable fixed bulkheads and a sliding bulkhead rotatable about the central column. The facility can be divided into three separate compartments if needed, enabling various fish operations. A bottom that can be elevated is provided between the two closable fixed bulkheads.

The net is fixed to the structure and has an incredible strength in order to prevent fish from

escaping. In addition, there is an extra net in the surface zone to protect against drifting matters.

Ocean farm1 is fully automated and equipped with an integrated high-tech sensor and measuring system. The entire facility can be run by a crew of 2–4 people only and heavier manual operations are avoided. Therefore, all farming operations can be managed either onboard or remotely, minimizing the use of service vessels and outside equipment, making the entire facility more environmentally friendly. This means that the fish can stay inside the net from stocking to harvestable fish.

After the facility was successfully installed, the company initiated a pilot phase with around one million 270–280 g salmon smolt in the cage. After 15 months of production in the sea, the fish has shown very good growth and that the quality is smooth and good. Few salmon lice have been observed and it has not been necessary to carry out a single lice treatment (Ocean Farming 2019).



Modern Aquaculture Structures, Fig. 8 Aquapod from InnovaSea. (Source: <https://www.innovasea.com/>)

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Modern Aquaculture Structures, Fig. 9 Aquapod fish farm. (Source: <https://atlasofthefuture.org/project/aquapod-fish-farm/>)



Modern Aquaculture Structures, Fig. 10 Ocean Farm 1. (Source: <https://salmar.no/>)

A second production cycle started in August 2019 and so far in the cycle good biological status has been observed, with good growth, low lice, and low mortality.

Salmar is eyeing farming in deeper, more exposed locations after the Norwegian Ministry of Fisheries and Coastal affairs awarded permission to “the Smart Fish Farm” concept Fig. 11.

The Smart Fish Farm will have twice the capacity of Ocean Farm 1 and the total aquaculture volume of the chambers is 510,000 m³ when the operating depth is 45 m. The Smart Fish Farm concept is based on a semi-submersible steel structure consisting of a wide center column and a surrounding framework mainly with circular cross-sections. The framework stretches supports netting panels that provide eight separate chambers. The Smart Fish Farm has an estimated price tag of NOK 1.5 billion.

Although the Smart Fish Farm looks similar to Ocean Farm 1, it differs significantly in some areas. It will withstand substantially more exposed areas and have twice the capacity. But the main difference is that the central closed column will be equipped for processing fish, control

and management of the unit, as well as an advanced system for transporting fish related to the eight surrounding production chambers.

Havfarm 1 & 2

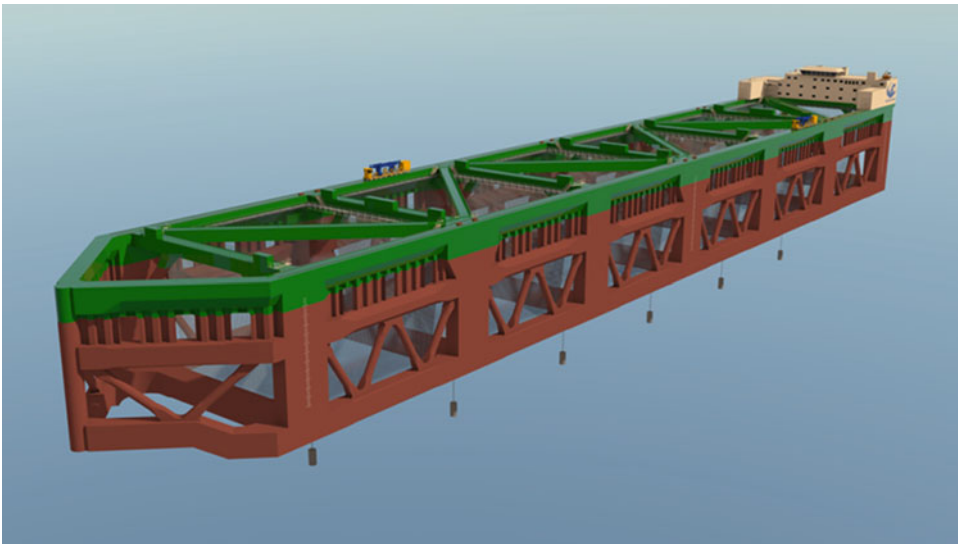
The Havfarm concept was developed by Nordlaks along with veteran ship creators at NSK Design in 2015 and awarded 21 licenses for a stationary Havfarm (Havfarm 1) (Figs. 12 and 13) and a dynamic Havfarm (Havfarm 2) (Fig. 14).

Havfarm 1 is a moored, long, and slender ship-like fish farm, 385 m long and 60 m wide. The unit consists of a bow section to reduce the wave and current loads in the cage system, a central open frame construction that holds six pyramid-shaped net pens, and a stern section with superstructure. The construction weight is around 33,000 tonnes and can hold up to 2 million salmon.

The Havfarm 1 is moored through a turret in the bow, and a total of 11 mooring lines and 11 anchors weighing 22 tonnes each. The unit will weathervane around this, thus making it capable of absorbing the feces and eventual feed spill without negative effects on the environment. This has resolved some of the challenges in the



Modern Aquaculture Structures, Fig. 11 Smart fish farm. (Source: <https://mariculture.no/>)



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Modern Aquaculture Structures, Fig. 12 3D model of Havfarm 1. (Source: <https://www.tu.no/artikler/det-forste-lakseskipet-har-12-ganger-sa-mye-stal-som-en-vanlig-bronnbat/450811>)

aquaculture industry, where environmental footprint in small geographical areas is often a consequence if the natural current does not transport any feed spill and feces away.

Havfarm 1 was built at CIMC Raffles Shipyard in China and has been anchored at the site at Ytre Hadseløya, Norway in July 2020. The smolt will be released in traditional facilities for growth up to



Modern Aquaculture Structures, Fig. 13 Havfarm 1 on site. (Source: <https://www.nordlaks.no/pressekt-havfarm>)



Modern Aquaculture Structures, Fig. 14 Concept of Havfarm 2. (Source: <http://www.nskshipdesign.com>)

1–1.5 kg before the fish is transferred to Havfarm 1. The first stock of fish is scheduled to be released into the farm in September 2020.

Havfarm 2 will not be permanently moored but will use a dynamic positioning (DP) system to maintain position. The dynamic positioning

system allows vessels to automatically change their position and heading using thrusters and propellers. The unit can move to a sheltered location in the event of a storm by its own propulsion.

Havfarm 2 is planned to operate in two different zones: an exposed zone, which can be divided into

several anchor points, and a safe zone. Havfarm 2 shall be located in an exposed location when the weather conditions allow and shall shelter in a location less exposed to waves in harsh weather by its own propulsion. This design allows the unit to operate in larger and more exposed areas.

Deep Blue 1

In 2018 China's first deep-sea fish farm, "Deep Blue 1," (Fig. 15) was successfully launched by Wuchang Shipbuilding Industry Group and towed to Cold Water Mass area in Yellow Sea, China.



Modern Aquaculture Structures, Fig. 15 Deep Blue 1 in dry dock. (Source: <http://www.wanzefeng.com>)

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Modern Aquaculture Structures, Fig. 16 Deep Blue 1 with new retrofitted tower. (Source: <http://m.rzw.com.cn/particle/485837>)



Modern Aquaculture Structures, Fig. 17 Hai Xia 1 in dry dock. (Source: <http://www.sasac.gov.cn/n2588025/n2588129/c14167773/content.html>)

The unit has a perimeter of 180 m and height of 35 m. The volume is 50,000 m³ and can produce about 1500 tonnes fish per season. In 2019 the “Deep Blue 1” was retrofitted with a tower of 35-m height (<https://ilaks.no/den-kinesiske-havfarmen-deep-blue-no-1-tilbake-i-drift-dette-er-nytt/>). The central tower (spar) houses machinery spaces, feed storage, and provides accommodation for operators (Fig. 16).

Its position underwater can be adjusted between 4 and 50 m to secure the best temperature for the salmon. By submerging under water, the pen will be able to withstand typhoon.

Hai Xia 1

The Dutch company De Maas SMC, a firm operating in the offshore oil and gas industry, has partnered with the local government in Fujian province to develop central column semi-submersible deep-sea fish farm, “Hai Xia 1 (Fig. 17).”

“Hai Xia 1” is 139-m in diameter and 12-m high, which is composed of a central cylinder

structure and the outer frame structure. The unit has a floater, which is below the surface and fused to the central spar. Water can be pumped in and out to sink/rise the unit. The spar houses the control room for the unit (<https://www.undercurrentnews.com/2018/10/01/expert-qa-designing-building-offshore-aquaculture-pens/>).

“Hai Xia 1” was constructed in a shipyard in the port city Mawei, Fujian and successfully deployed at East China Sea.

Cross-References

- Net Structures: Design
- Net Structures: Hydrodynamics
- Traditional Aquaculture Structures

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Modification for Reuse

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 Tianjin Key Laboratory of Port and Ocean
 Engineering, Tianjin University, Tianjin, China

Synonyms

[Change for reuse](#); [Conversion for reuse](#)

Definition

Some decommissioning vessels, offshore structures, or their components are in relative good conditions. After the condition assessment and a series of maintenance and transformation, such as adding some new facilities, deck reinforcement, changing the single-layer tank to double layer, the updated vessels or platforms can continue to work for other projects.

Scientific Fundamentals

Background and Introduction

Shipowners scrap their ships for various reasons, such as aging, technical obsolescence, low earnings, high scrap prices, and bad market expectations (Stopford 2009). Most shipowners base this decision primarily on the price offered by the ship

recycling yard to buy an end-of-life ship. The recycling yard offering the best price usually wins the contract (Jain et al. 2017). The recycling yards offering “green” recycling services generally quote lower prices than other yards due to the higher cost of dismantling a ship by following international ship recycling regulations and health, safety, and environmental (HSE) management systems, or the ship recycling regulations such as the Hong Kong Convention and EU Ship Recycling Regulation are considered innocuous to environment. According to an estimate by Abdullah et al. (2012), the annual global capacity of recycling was around 780,000 lightweight tonnes (LDT) in 2012. Such yards generally offer a lower price compared to other yards operating in the same region.

The ship recycling process recovers steel and other materials in good condition. These materials can be reused and sold to the steel industry. The recycling shipyards in South Asia have been using substandard and standardized methods. The substandard method is the most common and corresponds to the (beaching) modality utilized mainly in India, Bangladesh, and Pakistan. The standardized method uses the landing and dry docking methods in Turkey, China, Europe, and North America (Ocampo and Pereira 2019).

Various shipyards have experience in managing repair and conversion projects with different vessels in order to extend their life or have more functions. The modification work includes but does not limit to structural steel renewal and refitting, hull blasting and painting, large machine shop facility, tail shaft and propeller repairs, engine overhaul, turbo charger overhaul, electrical repairs, automation and control systems, tank cleaning and coating, aluminum hull repairs, yacht repairs and refurbishments, deck machinery overhauls, pipeline and valve overhauls/renewals, and so on. During the modification progress, some guidelines or rules proposed by the official societies, such as DNV (2013), ABS (2014), BV (2017), CCS (2016) etc., should be obeyed or referred.

Apart from vessels, some of the offshore platforms are approaching the end of their production life and will need a strategy for removal, reuse, and recycling in the following future. For each

existing, as well as new, platform, it will be necessary to document the removal and modification procedures in detail. The opportunity to reuse a platform would require a favorable circumstance for old removed, reinstalled, and refurbished structure to be considered in comparison to building an entire new and purpose-built facility (Ole Olsen 2001).

It is therefore important that available reusable platforms are advertised well in advance. The operators must be aware of the reuse option when planning for new field developments. Within some years, the operators may have several existing platforms to choose from, when assessing the best option for the actual new field development (FIB 2002).

Key Applications

Representative Modifications

Oil Tanker Modification for FPSO/FLNG

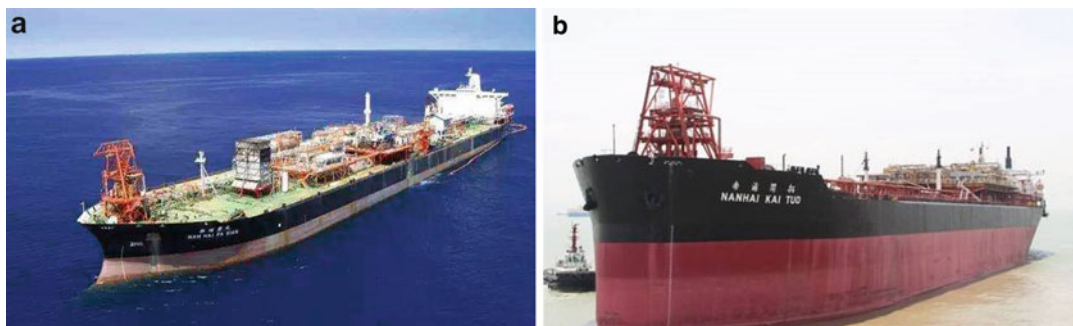
In recent offshore oil and gas markets, there are two major approaches to build FPSOs (Floating Production, Storage, and Offloading Facilities). One way is to construct a new facility, and the other way is to modify oil tankers. Comparing with the latter approach, building a complete new FPSO is more time-consuming and demands for relative large funds. Hence, more shipyards and shipowners would like to choose modifications, which will save almost 10-month construction periods as well as 20% of the total cost. In some regular sea states, the rental fee of a FPSO

converted by oil tanker is more than the sum of modification fee and rental fee of oil tanker. Hence, it attracts more and more shipowner to modify their aged oil tankers to FPSOs (Zhang 2017). Some converted FPSOs in China are shown in Fig. 1.

The base of FPSO modification project is to choose a suitable old oil tanker. First of all, based on the requirement, crude production, deck area, and transportation ability of shuttle tanker in operation offshore oil field, the main dimensions and tonnage of the target oil tanker are determined. Afterward, the next step is to seek the suitable oil tanker according to the information in every way. Based on a set of hull design parameters, different hull configurations are evaluated and the respective advantages and disadvantages assessed in order to identify the best candidates for conversion (Biasotto et al. 2005). Furthermore, based on investigations on the target vessels, the cost of the whole modification project could be predicted, taking all factors into consideration, including the age of vessel, status of vessel and its facilities, etc. During the project demonstration of the conversion, the key factor of the feasibility is the economic performance compared with the new building FPSO. Finally, a structural assessment is necessary to judge whether the oil tanker meets the requirement of the design life.

The tanker conversion activities include the following parts:

- Hull conversion
- Mooring structure conversion
- Risers



Modification for Reuse, Fig. 1 Converted FPSOs in China. (Source: CNOOC)

- Deck reinforcement
- Accommodation – living and safety
- Helideck and deck cranes
- Hull corrosion protection
- Tanker propulsion
- Tanker services (power and heat plant)

Reuse of Offshore Platform

After deciding to totally remove a platform from the seafloor, operators have several options (O'Connor 1998; Van Voorst 1999; Gibbs 2000).

- The platform can be taken to shore, where it is disassembled and the components either recycled, sold as scrap, or discarded in landfills or other depositories.
- The structure can be reconditioned and reused. As an example, in 1997 a platform was removed from the North Sea, taken to shore and cleaned, refurbished, and shortened by 10 m (33 ft) and installed in another North Sea location. A few small platforms have also been reused in the Gulf of Mexico (Schroeder and Love 2004).
- Platform structure can be towed to a site remote from the intact platform (in either shallow or deep water) and reefed. Remote, shallow-water reefing has occurred a number of times in the Gulf of Mexico, with the most extreme example towing structures of two Tenneco platforms over 1480 km (920 mile) from offshore Louisiana to a site 2.4 km (1.5 mile) off Dade County, FL (Wilson et al. 1987).

A platform could be either reused in situ or removed, reinstalled, and reused at its new location. The design purpose, to produce oil and gas, could be changed as well. Finding another oil field that will match all the platform properties and production capacity is perhaps impossible but also not necessary. Some modifications and refurbishment must be performed anyway. Successful reuse of offshore platforms is dependent on water depth, seabed conditions, and other environmental loads.

An advantage for reused structures could be a shorter timescale from planning to production start. Another advantage is the good durability of

offshore structures, when well designed and built. There are also a lot of other proposed uses for old platforms: bridge piers, coastal defense, wind farms, and artificial reefs in addition to some more exotic examples as prison, casino, and fish farms.

However, there are also several issues existing:

- Skirts and bottom slabs are designed for particular soil conditions. If the conditions at the new location are totally different, this may make relocation impractical.
- Estimated life remaining (particularly with regard to fatigue) must be equal or longer than the estimated production life for the new field. Assessment of structure life remaining is important in this process.
- Using the platform for new activities requires due consideration of personal safety, transportation issues, and maintenance of the platform.

There will always be a day for end use and decommissioning for a platform; thus, the dismantling process will always be applicable. However, it should be noted that steel jackets will have an imaginable thickness loss due to the corrosion after its lifetime service. It is a possible choice to evaluate and certain the high-value elements which will be used in the following projects, in order to judge whether it is fit and economic for reuse. On the other hand, concrete gravity-based structures are usually not removable, and they are commonly modified and reuse in its “former” operation area.

Reuse for Subsea Components

It is widely accepted that subsea structures, such as wellheads, production manifolds, etc., are designed as a reusable facility. It will not only reduce the development cost of the offshore oil fields but also be beneficial to protect the offshore environment and maritime safety. As one of the intricate machined hardware sized for the flow rates, well shut-in pressures, installation vessel interfaces, and intra-field connections, it is normally designed to a high specification with corrosion resistance (Gorman and Neilson 2012). For these deepwater subsea structures, they are

utilized for production in a short duration. Besides, they hardly experienced external corrosion, so they are pretty suitable for a second-round operation in another oil field.

However, it should be noted that there exist many issues or obstacles before to make the decision to reuse this facility in another place. On the one hand, the factors, including valve size, material class, and its remaining useful design life, should be carefully evaluated and examined. On the other hand, the designers should check the tree's flowloop configuration as well as the valve number. According to the working area water depth, the subsea trees could be categorized into four types, which are mudline, diver assist, diverless, and guidelineless. The different type has its individual features, such as thickness, metal loss protection, and structure complexity. Hereby, this is another factor to be considered during the modification. Moreover, the wellheads, foundations, control systems, and flowline connection techniques may also affect successful reuse (Kawase et al. 1997).

Reuse for Mooring and Riser System

For the moored offshore platforms, there are mainly two mooring types, one is permanent and the other is disconnectable. Taking FPSO as an example, this system usually contains turret, mooring cables, and risers. Based on the evaluation from different aspects, each part of them could be reused in other oil fields.

In general, both internal and external turret design standardizations further facilitate reuse in that plans for additional, or future, risers may be well thought out avoiding the complications due to structure or piping interference in a future modification. The turret system piping, manifold, swivels, and safety features must be evaluated for compatibility with the new field requirements following a thorough inspection to determine the existing condition (Kawase et al. 1997). Furthermore, if the modification is adopted, it is best to perform this progress in a shipyard, and in some particular cases, dry-docking may be needed due to the project plan.

The mooring cables can also be reused with the proper consideration of corrosion, wear,

fatigue, and handling, so can the anchor components. However, a careful inspection work on field life, strength, design temperature, and installation equipment should be adopted in order to check the feasibility to the new field requirements. Similarly, the risers are also able to hold the reusable feature, although they are custom-designed for their corresponding area. Specifically, the lazy wave and free hanging catenary risers are easier to re-utilize as the riser forms part of the flowline after touchdown. However, the inspection before the second-time adoption contains much more terms, such as service life and type, design temperatures and pressures, site particulars, riser configuration, and installation equipment. Recertification of the riser by the riser manufacturer or the certifying authority may also be required. Moreover, a detailed examination of the external sheath and the end fittings will be required. Buoyancy modules will need to be removed, refurbished, and replaced during reinstallation (Kawase et al. 1997).

Future Technics

Combined Offshore Energy Harvesters

Renewable energy resources, called sustainable or alternative energy, are energies generated from natural resources such as wind, sunlight, tide, hydro, biomass, and geothermal which are naturally replenished (Díaz-de-Baldasano et al. 2014). The main challenge of replacing legacy systems with more environmentally friendly alternatives is how to capture the maximum energy and deliver the energy at a minimum cost. A combination of two or more types of energy sources might provide a good chance of optimizing power generation system (Da and Khaligh 2009). Hence, a series of novel offshore structures for renewable energy power generations, such as wind/tidal, wind/wave, wind/solar, and so on, are developed. However, most of present offshore energy harvesters only have single function. In order to establish the hybrid power system, a modification is necessary.

CO₂ Emissions of Oil and Gas Production Platforms

Oil and gas platforms are energy-intensive systems, and each facility uses from a few to several hundred MW of energy, depending on the petroleum properties, export specifications, and field lifetime. For instance, the Norwegian oil and gas offshore sector has contributed for about 20–30% of the total Norwegian CO₂ emissions in the last decade, and this number is expected to stay in the same magnitude in the coming years. These emissions are caused in a large share by the combustion of natural gas in gas turbines to produce the power required to drive the compression and pumping operations, and the remaining is associated with gas flaring and diesel combustion. Several technologies for increasing the energy efficiency of these plants are investigated. Among them, it is an effective way to make some modifications or add some new facilities on the old platforms in order to lessen CO₂ emissions. They aim at reducing the electrical or thermal energy use, by redesigning some sections of the processing plant (production manifolds), re-dimensioning the compressors (gas recompression and treatment), promoting energy and process integration (heat exchanger network), and implementing expanders and waste heat recovery cycles. The savings potentials differ significantly from one platform to another (Nguyen et al. 2016).

Fish Farms

Nowadays, more and more researchers are considering setting up fish cage structures in large offshore structures such as platforms or jackets to create such new structures for offshore ranching. However, there still exist several issues to be taken into account. On the one hand, there will be many difficulties for the cage structure, such as motion response according to the change in draft, lowering and lifting, and so on. On the other hand, storm damage due to snagging and fretting of nets or cages is likely to be frequent and not easily repaired. Furthermore, the development and maintenance cost is likely to exceed the profits of inshore fish farming.

Modification for Entertainment, Tourism, or Living

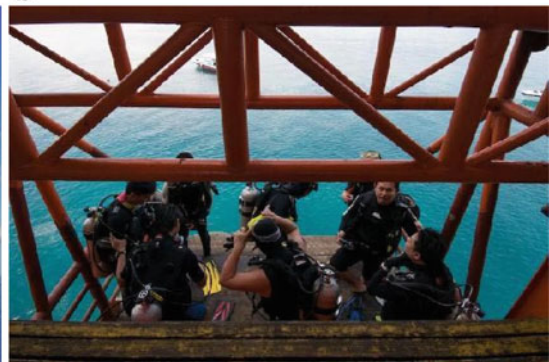
Apart from the above solutions, there is an interesting and creative way to reuse the retired offshore platforms. This answer is to modify the platform as a sea-view hotel or a diving resort. The Seaventures (see Fig. 2) is a dive resort made from a converted oil rig offering a perfect location for nonstop diving. Besides, the dive resort also provides accommodation and entertainment facilities. Another similar idea is performed on the abandoned oil platform “Albuskjell 2/4-F” by Norwegian architect Sverre Max Stenersen. The platform is modified to add some living or business features, and it is planned to be located in the port of Trondheim.

M

a



b



Modification for Reuse, Fig. 2 Seaventures dive resort. (Source: sipandan.com)

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Monopile Foundations in Offshore Wind Farm

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Synonyms

[Single pile foundation](#)

Definition

Monopile foundations refer to the commonly used steel pipe piles with large diameter and thick wall, which are usually driven (hammered) into the seabed.

Scientific Fundamentals

Scope for Development of Monopile Foundation

The energy crisis and air pollution caused by traditional energy resources around the world have led to an urgent need for clean and renewable resources in the twenty-first century. Under this background, offshore wind farms have become an important source of renewable energy in recent years. The first large-scale offshore wind farm was successfully installed in 2002 in Denmark, which is the

milestone of offshore wind farm construction. Since then, in Europe, there have been booming developments in offshore wind farm benefitted from strong political and financial support. Moreover, political strategies in Europe, and in the United States, aim to enlarge to the offshore wind energy sector to limit the dependence on oil, gas, and coal (Malhotra 2011). When it comes to China, the first large-scale offshore wind farm (offshore wind farm of Donghai Bridge) is built in Shanghai by 2009, indicating the new epoch of offshore wind energy. By 2015, the capacity of offshore wind farms in China has reached over 1.01 GW, and several projects that made this happen are shown in Table 1. The relative statistical data of several countries of annual offshore wind capacity from 2012 to 2015 and 2017 are shown in Table 2 and Fig. 1, respectively.

There are several types of foundations to support offshore wind turbine structures, which include monopile foundations, gravity foundations, tripod foundations, suction foundations, and floating foundations. Each type of foundation has their own structural advantages and limitations. Among these foundations, monopiles are most commonly used foundations in offshore wind farm as they can withstand harsh loading condition transferred from the supported superstructures in deep waters with the advantage of lower cost, more structural simplicity, and smaller foundation settlement. In Europe, the monopiles constitute 81.7% of all installed turbine substructures, which are shown in Fig. 2. The same phenomenon can also be found in China's offshore wind farms.

Monopile Foundations

Monopile foundations are usually steel pipe piles with large diameters and thick walls that are driven (hammered) into the seabed. Carlo (Ortolani 2017) and Byrne (2015) reported commonly used monopile foundations with large diameter (D) of 4–7 m (Fig. 3), sometimes even up to 10 m in diameter with their lengths of 40–50 m (while in China, the length of several monopiles is as long as 70 m, i.e., monopiles in Xiangshui offshore wind farm). The

Monopile Foundations in Offshore Wind Farm, Table 1 Offshore wind farms in China from 2007 to 2015 (<http://www.cwea.org.cn>)

year	location	Capacity (MW)	
2007	The Bohai Sea, Liaoning	1.5	1.5
2009	Nantong, Jiangsu	8.0	16.0
	Yancheng, Jiangsu	2.0	
	Weihai, Shandong	6.0	
2010	Shanghai	102.0	135.5
	Nantong, Jiangsu	24.5	
	Nantong, Jiangsu	2.5	
	Yancheng, Jiangsu	6.5	
2011	Nantong, Jiangsu	101.0	109.6
	Shanghai	8.6	
2012	Nantong, Jiangsu	110.0	127.0
	Nantong, Jiangsu	3.0	
	Weifang, Shandong	9.0	
	Fuqing, Fujian	5.0	
2013	Nantong, Jiangsu	5.0	8.0
	Yancheng, Jiangsu	3.0	
2013	Nantong, Jiangsu	4.0	31.0
	Binghai, Tianjing	27.0	
2014	Rudong, Jiangsu	20.0	227.2
	Shanghai	102.2	
	Rudong, Jiangsu	56.0	
	Rudong, Jiangsu	49.0	
2015	Putian, Fujian	12.0	360.5
	Putian, Fujian	50.0	
	Haimen, Guangdong	1.5	
	Dafeng, Jiangsu	9.0	
	Rudong, Jiangsu	100.0	
	Xiangshui, Jiangsu	32.0	
	Binghai, Jiangsu	20.0	
	Rudong, Jiangsu	56.0	
	Rudong, Jiangsu	80.0	
Total		1016.3	

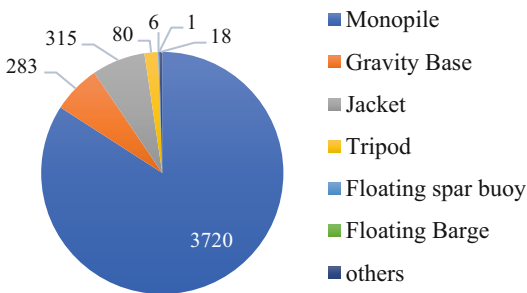
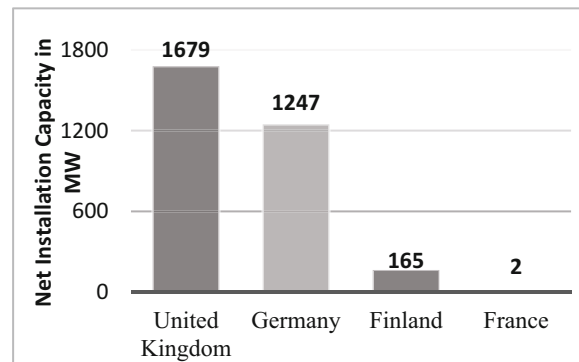
monopiles with large diameters and long embedment would allow the supporting of larger wind turbines – from 3–4 to 5–8 MW of power capacity and with the installation possible in deep waters of 30 m. “Offshore wind turbines with 6–8 MW capacity have already existed in the energy market, with superstructure where the height of the hub can reach up to 100 m and blade diameters up to 150 m, e.g., DONG's Westernmost Rough and Burbo Bank Extension wind farms” (Ortolani 2017).

Monopile Foundations in Offshore Wind Farm, Table 2 Capacity of offshore wind farm in several countries (in GW) (<https://windeurope.org>)

Year	2012		2013		2014		2015	
Country	New	in summary	new	in summary	new	in summary	new	in summary
Britain	854.2	2948.0	733.0	3681.0	813.0	4494.0	566.0	5060.0
Germany	80.0	280.0	240.0	520.0	529.0	1049.0	2282.0	3331.0
Denmark	46.8	921.0	350.0	1271.0	0.0	1271.0	0.0	1271.0
China	127.0	389.6	39.0	428.6	229.3	658.0	3610.0	1091.0
Belgium	185.0	380.0	192.0	571.5	140.0	711.0	0.0	711.0
Netherland	0.0	247.0	0.0	247.0	0.0	247.0	180.0	427.0
Sweden	0.0	164.0	48.0	212.0	0.0	212.0	0.0	212.0
Japan	0.1	25.3	24.4	49.7	0.0	49.7	3.0	52.7
Total	1293.0	5355.0	1626.0	6981.0	1711.0	8692.0	3392.0	12084.0
Global Total	1298.0	5415.0	1631.0	7046.0	1725.0	8771.0	3392.0	12105.0

Monopile Foundations in Offshore Wind Farm,

Fig. 1 Annual offshore wind capacity installations per country in 2017 (MW) (<https://windeurope.org>)



Monopile Foundations in Offshore Wind Farm, Fig. 2 Share of substructure types for grid-connected wind turbines (units) (<https://windeurope.org>)

Components of Monopile-Supported Offshore Wind Turbine

As illustrated by Velarde (2016), a typical monopile-supported offshore wind turbine consists of a steel pipe pile with a large diameter which is driven into the seabed by a steam or

hydraulic-powered hammer. The support structure consists of three main parts: foundation pile, transition piece (or without), and tower as shown in Fig. 4. These components of the support structure hold the rotor-nacelle assembly in place. The nacelle houses electronic and mechanical parts of the turbine such as gearbox and generator. The function of monopile is to transmit loads such as wind loads (on the turbine and tower), wave and current loads, lateral load, and overturning moments (due to cantilever structure of turbine) and axial loads (due to the weight of the superstructure) to the underlying soil without any failure over the design period. The monopile is connected to the tower by the transition piece which is grouted. This transition piece can be used for anchoring of boats, as a ladder, as platform, and can also be used for correction of vertical misalignments during pile driving. The

Monopile Foundations in Offshore Wind Farm,

Fig. 3 A 7.5-m diameter monopile in an offshore wind



monopile and transition piece assembly are the most important parts to connect the rotor-nacelle assembly to the tower.

Installation of Monopile Foundation

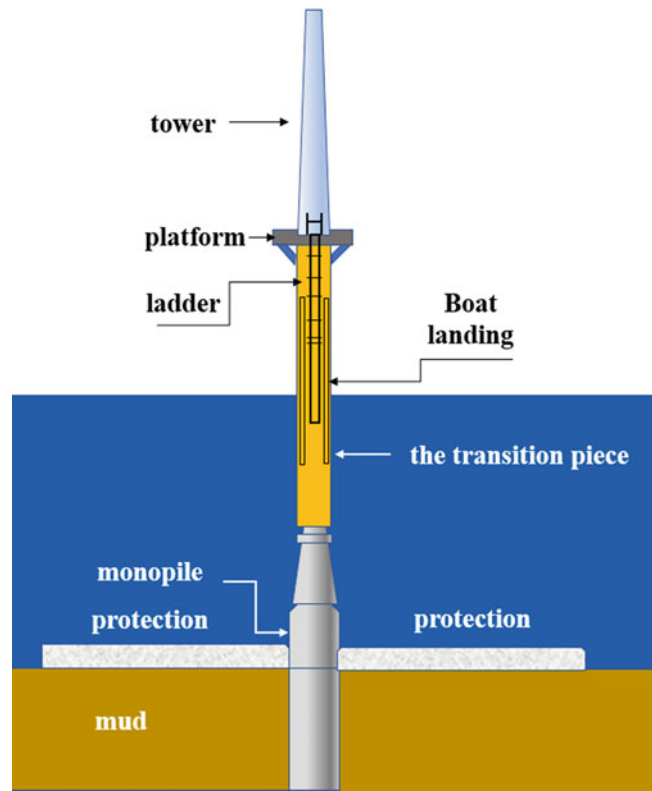
The installation procedure for monopile includes: (1) Transportation to site by feeder barge, (2) transportation to installation vessel or floating vessel, (3) upending and lifting into position, (4) alignment to position, and (5) driving or drilling into the seabed. The installation of monopiles can mainly be done in three methods (Fig. 5) depending on the subsurface conditions of the seabed: (1) Impact method, (2) vibratory method or grouted into sockets, and (3) drilled into rock. The drilling method is more efficient and less cost-effective of installation than the driving method. Most offshore piles are installed with impact-driven piles, which produce noise, and the noise must be reduced using expensive techniques to limit the damage on sea mammals. Vibratory driven piles are commonly used in sandy soils; the pile is driven at the head by

applying quick sequences of downward and upward motions to the pile in order to reach the required embedded length without noise mitigation system and fatigue induced by impact driving (Matlock 2015). Drilled method is used when the soil is too hard for installation. This installation method has large effect on the bearing capacity of piles. Monopiles used for offshore wind turbines can be considered as low-displacement piles as Pedram (2015).

There are many researches focusing on the effect of installation methods on the pile behavior; this has been studied by various methods such as field tests, laboratory study, mechanical modeling, or numerical studies. During design of monopiles, installation methods should be taken into consideration while determining soil strength and stiffness around pile body (Phanikanth et al. 2012). A typical installation physical modeling was proposed by Hokmabadi et al. (2015), a numerical modeling by McCabe and Sheil et al. (2015), and some field testing/observations was done by Zhuang and Cui ((2015); (Kog 2016; Eslami and

Monopile Foundations in Offshore Wind Farm,

Fig. 4 The monopile support structure



Fellenius 1997; Eslami et al. 2012). Zarrabi and Eslami (2016) has conducted more than 30 axial compressive and tensile load tests which were performed on various monopiles installed with six different installation methods (jacking, drilling and grouting, driving, drilling and placing, screwing, and postgrouting). From the results, Zarrabi and Eslami (2016) found that jacked and precast-in-place piles had the highest and lowest capacities, respectively. Also driven piles had the next highest capacity among other piles.

Field Tests Example on Monopile with Large Diameters

In offshore wind projects, field test on monopiles is an essential part in construction; the main purpose of the field test is to make sure of the bearing capacity before construction and to prove that the existing pile is secure under static or dynamic loading conditions.

Depending on the force direction, there are two kinds of loading methods: vertical loading tests under axial-compressive and tensile loads, and lateral load test. Depending on the test methods, the commonly used tests are conventional loading test, bidirectional test (Osterberg cell or O-Cell test), low strain impact integrity test, and high strain impact test.

The conventional axial compressive load test is one of the most commonly used methods to determine the vertical bearing capacity of monopile in offshore area. Anchor piles as a load test reaction system and a jack as a force-applied system are used to offer the vertical load on the pile head, which can be seen in Fig. 6; the conventional static load test is a reliable method to determine the bearing capacity of piles (Russo 2013). But for monopile with very large diameter, it is unattainable for this loading test to determine the capacity behavior of these piles (where the load test reaction systems often do not have the capacity to take the pile all the way to plunging).



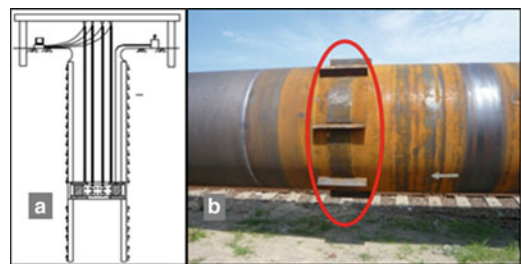
Monopile Foundations in Offshore Wind Farm, Fig. 5 Pile installation by (a) impact method (Yushan Super Bridge project), (b) vibratory method) (De Neef et al. 2013), and (c) drilling method (Mai 2012)

Monopile Foundations in Offshore Wind Farm, Fig. 6 Conventional load test of an offshore wind project in China (offshore wind farm of Donghai Bridge)



As for the *bidirectional* test, it can solve the above problem, and this method has been successfully used in some offshore wind projects. In bidirectional load test, a very important equipment is load cell, which is shown in Fig. 7. During the loading process, the top and bottom planes of load cell were apart and pushed their corresponding pile sections to move contrarily until the ultimate resistance of top pile is reached. The displacement versus applied load of top and bottom planes was recorded at each loading step. However, there is a need for several hypotheses in bidirectional test to transform experimental results into those of an equivalent conventional load test.

As for the low-strain impact integrity test and high-strain test, the low-strain test can be



Monopile Foundations in Offshore Wind Farm, Fig. 7 (a) Bidirectional test of pipe pile and (b) Load cell of pipe pile

used to find out the integrity of pile body, while high-strain impact test can be used to determine the vertical bearing capacity of pile only if the result can be compared with that of static load test.

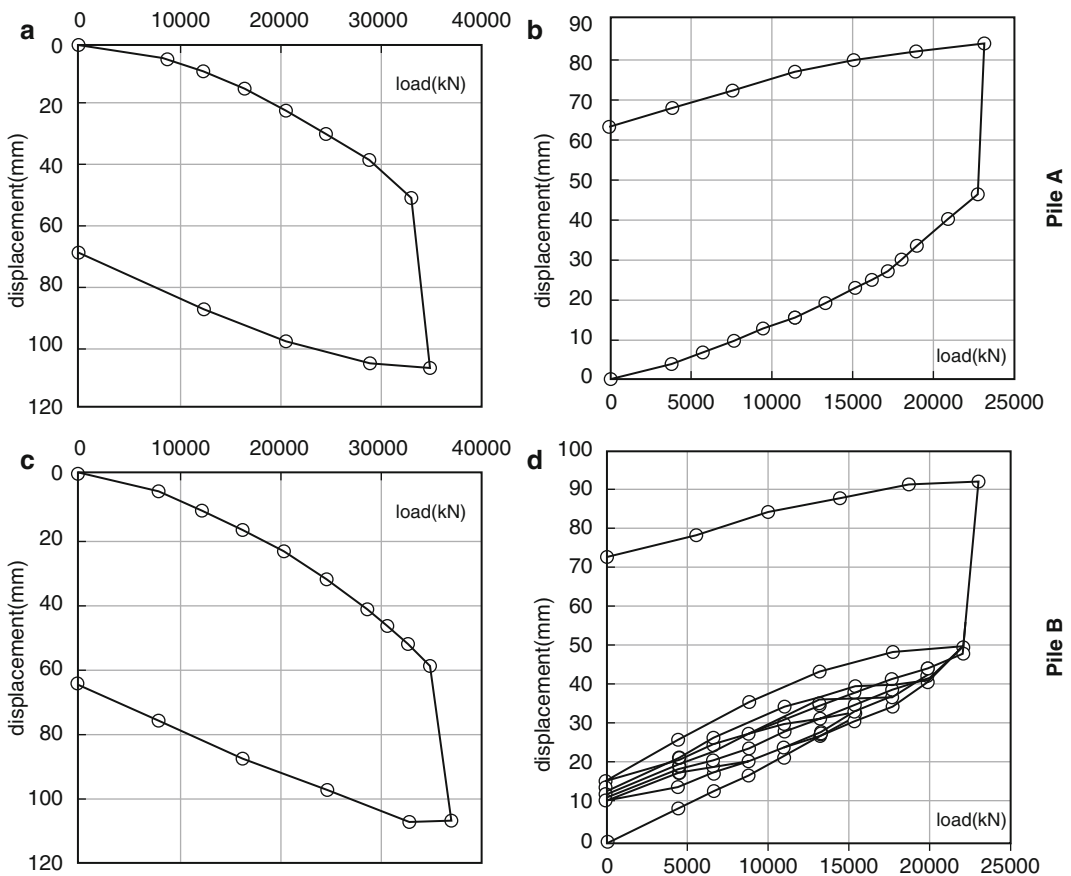
Sample Field Tests on Monopile

In this part, the field static loading tests results of an offshore wind farms in China are introduced. These loading test methods include axial-compressive and tensile load test, as well as lateral load test. The wind farm project is located at the

coast area of Jiangsu province; the soil profile here consists of marine clay and sand. The relative parameters of piles (A and B) are shown in Table 3, the corresponding vertical capacity test results are shown in Figs. 8 and 9, and the shaft resistance of each layer of piles A and B is shown in Table 4.

Monopile Foundations in Offshore Wind Farm, Table 3 Test pile parameters in field test (Li et al. 2019)

Project name:	Project 1	
Pile name	A	B
Pile length (m)	71.5	77.5
Pile diameter (m)	2.0	2.0
L/D	35.8	38.8
Test method	Axial compression & tensile load test, lateral load test	Axial compression & tensile load test

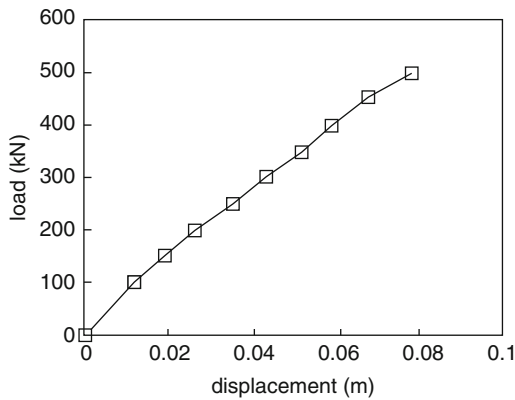


Monopile Foundations in Offshore Wind Farm, Fig. 8 (a & c) are downward load-displacement of pile top from test of pile A and B, respectively; (b & d) are

uplift load-displacement of pile top from test pile A and B, respectively. (Source: Li et al. 2019)

From Fig. 8a and b, the vertical compressing capacity of A is 32,800 kN, the total shaft resistance is 31,076 kN, and the uplifting capacity of B is 22,800 kN; in Fig. 8c and d, the vertical compressing capacity of A is 34,850 kN, the total shaft resistance is 33,260 kN, and the

uplifting capacity of B is 22,000 kN. From Fig. 9, the displacement of the pile at load-applied point of 500 kN is 0.0779 m. The ultimate shaft friction resistance at different soil layers is shown in Table 4. From these results, we can find that the friction resistance from compressive load test is much larger than that from tensile load test.



Monopile Foundations in Offshore Wind Farm, Fig. 9 Lateral load-displacement at mud line of A

The Problems with the Use of Monopiles

Monopiles are widely used in offshore wind projects, while large diameter single piles with diameters of 5–10 m (Byrne 2015) are considered for future design purpose in deep offshore areas. The most difficult problem in the use of monopile foundation is the current design methods, originally used for small diameter piles. There are many problems when using the traditional design method on the large diameter monopiles, some of which are mentioned below:

Monopile Foundations in Offshore Wind Farm, Table 4 Shaft resistance of A and B from load tests

Pile name	Soil type	Thickness of each layer (m)	Shaft resistance (kPa) from compression load	Shaft resistance (kPa) from tensile load
ZK01	Mud	2.62	23	11
	Silt clay with mud	7.90	54	20
	Silt sand	3.00	67	36
	Silt sand	8.20	107	69
	Silt clay with mud	9.80	105	81
	Silt clay	16.30	94	77
	Silt clay	1.70	95	79
	Silt sand	4.08	128	101
ZK28	Silt sand	1.37	32	9
	Silt clay with mud	7.90	40	12
	Silt sand	3.30	63	31
	Silt sand	7.00	77	44
	Silt clay with mud	10.10	80	53
	Silt sand	12.00	103	66
	Silt clay	4.60	111	81
	Silt sand	7.40	135	93
	Silt clay	1.90	97	64
	Silt	3.53	114	77

- (a) The mechanical model of pile-soil interaction of monopiles with large diameter in sand and clay
- (b) The definition of initial tangent modulus in p-y curves considering embedment conditions
- (c) The calculation method of natural frequency of monopile considering embedment conditions
- (d) The influence of installation method on soil response around the large diameter monopiles
- (e) The cyclic load effect on the lateral capacity of large diameter monopiles

Cross-References

► Pile Capacity

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Moored Ship in Ice

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Synonyms

ATOT, Arctic Tandem Offloading Terminal; FPSO, Floating Production, Storage and Offloading Unit; FPU, Floating Production Unit; HSVA, Hamburg Ship Model Basin; RMRS, Russian Maritime Register of Shipping; SPS, Subsea Production System; STL, Submerged Turret Loading Concept; UFR, Umbilical, flow line, and riser

Definition

Moored ship means a ship to be secured or fixed to some bodies by using cables, lines, or anchors.

The ship is often used to assist underwater activities such as diving, seeking, and retrieval. It could also be used to perform oil and gas exploration or exploitation at seas.

Introduction

Interests in the Arctic and sub-Arctic waters have been growing since several large big hydrocarbon fields in the Barents and Kara Seas were discovered. However, it is still challenging to perform hydrocarbon exploration and exploitation in these locations, which always involves risks from low temperature, sea ice, wind, waves, currents, darkness, icing, etc. One of the biggest challenges in the Arctic and sub-Arctic areas is the sea ice. It exists in many features, such as level ice, broken ice, ice ridge, iceberg, and so on. Resulting ice loads are dominant to open water load, wind load, current load, and wave load on most occasions for polar ships as well as offshore structures. To resist ice action and keep the station, different marine structures are used for drilling, production, and offloading operations in ice-covered waters. Fixed structure is a good solution to shallow water operation, such as gravity-based structure, jacket platform, artificial island, etc. As water depth increases, moored floating vessels are shown to be attractive. This chapter will mainly focus on the types of Arctic moored ship, industrial operations,

numerical simulation, model tests, and full-scale measurements.

Moored Vessel Types

Generally speaking, there are three kinds of marine structures. The first is bottom-fixed. The second is moored with mooring lines. And the third is the dynamic-positioned. Mooring systems are often used for polar ship station keeping in severe ice with medium waters. As we know, ice crushing against vertical structure often leads to large ice force which may exceed capability of the mooring system. Thus, moored vessel is required to be designed with downward slope which could bend pack ice easily. Two typical concepts are included herein. One is traditional ship-shaped structure and the other one is the conical structure.

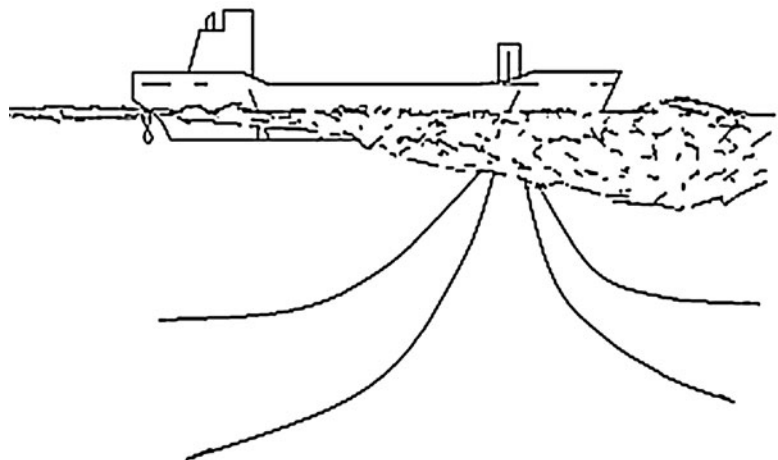
Ship-Shaped Structure

A ship-shaped structure includes relative long parallel mid-body and icebreaking bow. Some have also icebreaking capability with stern. In ice-covered waters, ice drift direction often changes from time to time. This makes ice vaning capability of moored ship very important. The moored ship (as shown in Fig. 1) should have a quick response to change heading under ice action and keep heading toward the ice drift. In general, a ship-shaped structure preforms well in surge, pitch, and heave but may experience large

M

Moored Ship in Ice,

Fig. 1 Moored ship ice in ice



sway and roll motion when ice intrudes from sideways.

Relevant experience with moored structures in ice is obtained from drilling operations in the Beaufort Sea. Starting in the mid-1970s to the late 1980s, Dome Petroleum deployed floating drillships named Canmar during the summer months. These were moored on site during the summer (open water) months, where relatively light ice conditions were encountered (Wright 1999). With the support of icebreakers, these drillships developed the capability of station keeping in a variety of ice conditions. This extended their open water operating season, although they did not work extensively in heavy ice. Unfortunately, although Canmar gained a great deal of experience with their Beaufort Sea drillships, there is very little documentation about their operations, particularly with regard to the load levels experienced by these moored vessels in ice.

For the development of the Grand Banks oil fields, the Terra Nova scheme was implemented (Wright 1998). It involves a floating ship-shape production vessel with integrated storage and offloading systems (FPSO), which is designed to continue operations in most environmental conditions. The FPSO could disconnect with the mooring lines and risers system and move off the site when avoiding severe ice actions. This approach is attractive for operations. The vessel has been operating year-around with light ice conditions encountered. The advantage is that capital cost is low and response to the ice conditions is quick, but the disadvantage is the potential for much downtime due to ice and the associated production delays. This experience has shown that moored vessels, with adequate levels of ice strengthening and ice management support, can be very effective in a wide range of moving pack ice conditions.

It is expected that the gas-condensate field Shtokmanovskoye, located in the central part of the Barents, will be developed in the near future. The water depth is 340 m. At this field, both occurrences of sea ice and icebergs have to be expected during operations. The general offshore facilities development scheme that has been

selected is based on the Subsea Production System (SPS) tied-back through a system of umbilical, flow line, and riser (UFR) to the ice-resistant ship-shaped Floating Production Unit (FPU) hosting gas processing, gas compression, living quarter, power generation, and all other utilities required to operate.

Some concept studies for different kinds of marine structures have been tried in model scale with level ice, broken ice, and ice ridge. Løset et al. (1998) performed a series of model-scale tests with the Submerged Turret Loading Concept (STL) for loading oil offshore in the Hamburg Ship Model Basin (HSVA) ice tank in Hamburg. The modelled ship had a conventional icebreaking bow at a scale of 1:36. The purpose of the tests was to study the feasibility of the STL concept in level ice, broken ice, and pressure ridges. It was found that pressure ridges produced for the model tests may cause forces over the capacity of the STL system. Later, a number of model tests with another concept were conducted in the HSVA ice tank (Løset et al. 2003). The concept comprises a single anchored moored ship located in the wake behind a moored buoy floating freely on the surface. The buoy with smooth surface was expected to break the ice in upward bending and also ridges. However, the results show that it is probably not practically feasible to apply a buoy with enough buoyancy to break the design ridges since ridges may exceed the breaking capacity of the buoy unless disconnection of the ship can be done. Ettema and Nixon (2005) reported model test data regarding ice-rubble loads acted on a moored conical platform. How ice-rubble loads were influenced by horizontal stiffness of the platform's mooring system were investigated with great interest. Aksnes (2009) used a simplified moored ship to perform model test in level ice. The ship model was fixed except surge degree of freedom with a linear spring. Different dynamic properties of the ship model as well as drifting ice were studied and analyzed.

Bonnemaire et al. (2008) proposed a new concept for offshore offloading operations in ice. The concept named Arctic Tandem Offloading Terminal (ATOT) is composed of a turret moored offloading icebreaker and an

offloading tanker in tandem. It was tested in the HSVA ice tank with focus on the measurement of mooring loads during interactions with ice ridges in head on drift.

Aker Solutions and Aker Arctic proposed an Arctic drillship concept which enables to extend operation season in waters with sea ice (Bruun et al. 2015). The concept was developed to perform drilling, coil tubing, wireline operations, and running and pulling x-mas tree through a forward turret for low air temperature.

The design work was performed regarding sizing of the hull, drilling, turret, and station keeping system. General arrangement drawings were developed for each deck level, and the technical work was well documented through development of an outline specification. The drillship was designed to operate in late ice season, summer season, and early ice season. The corresponding mooring system could resist a maximum load equivalent to interaction with level ice cover of 1.5 m thickness during ice drift change. This includes design of hull lines for effective ice breaking, a forward turret position to reduce the ice load during change of ice drift direction, and station keeping system (mooring system) with a high restoring capacity. A dual-class design complying with both RMRS (Russian Maritime Register of Shipping) ARC 6 and PC 5 (Polar Class) was performed. The thrusters can be used for both transit and station keeping. There are three open podded thrusters at stern and two nozzle retractable thrusters at bow used for both transit and station keeping. For drilling in open water, the five thrusters provide DP3 capability, while for drilling in ice, the turret mooring system will be used for station keeping assisted by aft thrusters during rotation in ice (forward thrusters will always be retracted when operating in ice).

Conical Structure

Considering the limited ice vaning capability of ship-shaped structure, conical structure with an axis symmetrical hull was developed. The waterline is circular. The structure does not need to change heading. Therefore, ice vaning is no longer required. The cost is that it may result in weak performance in surge, heave, and pitch.

A typical conical drilling unit, the Kulluk platform, was designed with a variety of special features to improve the performance under ice conditions in the shallow water (Wright 1999). It was deployed in water depths ranging from 20 m to 60 m. The system has a downward sloping circular hull near the waterline that breaks the oncoming ice mainly in flexure and an outward flare near the bottom that clears the broken ice cusps away from the moon pool and mooring lines. The strong mooring system could resist ice forces up to 450 t. It worked successfully with the downtime in operations less than 10%, but at a cost of extensive ice management by three icebreakers.

Considering the possible change in ice drift direction which may lead to a significant increase in ice load on the hull, some buoy-shaped structures based on prototype of the Kulluk were tested. The structure of this sort is attractive in that the need to keep heading toward the ice drift is no longer required. The conical structure FPU-Ice (by SEVAN Marine) was tested in the spring 2008 at HSVA (Løset and Aarsnes 2009). The purpose was to study the ice load level on the structure and its response in extreme first-year ice including the interaction with unmanaged ice ridges exceeding 20 m draft.

A Joint Industry Project was reported by Bruun et al. (2011) on development of conical floaters in ice, where a typical SPAR platform was developed during the project. Two large ice model test campaigns were performed in the period 2007–2010. The objectives were to investigate different floater geometries and ice model test setups (model fixed to a carriage and pushed through the ice vs. ice pushed toward a floating model moored to the basin bottom) and their influence on the ice failure mode and structure responses in the various tested ice conditions.

Main Considerations of Moored Ship Design

Three key elements should be well considered when designing a moored ship operating ice-covered waters. The hull lines of the vessel should

be optimized to reduce ice impact as much as possible and minimize the response of hull. The mooring and riser system are required to resist global ice action and keep natural position within an allowable watch circle. To estimate ice-moored ship interaction, some analytical methods and ice model tests are applied to calculate ice loads, mooring loads, and the responses of the ship, based on which both hull lines and mooring systems could be designed safely.

Vessel Systems

The specific hull lines of a moored vessel are important in that they have a strong influence on the vessel's ability of icebreaking and ice clearing process and the resultant ice loads that it is exposed to. In particular, the ice forces are significantly affected by the width of the vessel. The ice forces increase with beam size. The icebreaking load level at the waterline is often affected by the shape of bow to a large extent. How to optimize the bow shape to reduce ice load is a key consideration for moored vessels as well.

The overall hull form and draft are key factors with respect to the vessel's ability to clear ice and the potential for ice interference with its mooring and riser systems. Clearly, well-designed thruster systems are helpful to enhance ice clearance around a moored FPSO vessel or tanker. Especially for an azimuth thruster, it could assist both vessel's vanning response and ability to resist ice and other environmental forces. The configuration of the thrusters needs to be taken into consideration, which is expected to reduce ice resistance and ice impact on propellers. Good configuration will improve overall thrust efficiency of the vessel and save power. Moreover, any FPSO vessel or tanker that is used for ice operations should be structurally capable of sustaining the relatively high local ice pressures that may be experienced, particularly in heave ice, mixed ice and wave conditions, and small glacial ice.

Mooring and Riser Systems

Generally speaking, a vessel's mooring system should be strong enough to withstand relatively large ice forces, making sure that the vessel could maintain its position within an acceptable offset.

The overall mooring loads and individual mooring line tensions are of obvious importance and should not be exceeded. In the Arctic area, the drifting ice is changing direction frequently. Therefore, the mooring system is required to afford sound vanning capability to the vessel, which makes the vessel head toward to the new ice drift direction smoothly. During this process, the ability for riser system to track vessel motions with allowable offsets and stress levels is also important and should be taken into consideration.

When ice attacks the vessel from sideways, it is possible that ice floes move down the hull and become entangled with its turret, mooring, and riser systems. To avoid this potential risk, increasing the draft of the vessel preferably with some protection skirt around their submerged mooring and riser system is a reasonable solution. Ice ridge is often embedded into a level ice field. When it interacts with the moored ship, mooring lines are possible to be impacted by the keel part of the ice features. The resulting ice load and effect on the overall response of the system are also important, which endanger the safety of the mooring system. Under the condition of large ice features, the vessel has to disconnect from riser and mooring systems quickly and safely and move off location. This is one of the major challenges for most station keeping operations in pack ice.

Ice Loads and Vessel Responses

When designing a moored vessel in ice-covered waters, the ice conditions have to be determined as a basis. The ice loads and related ice-vessel interaction cases together with vessel responses to the ice loads are primary considerations. The ice management support should also be included to reduce ice load level if the ice condition is severe. Researchers have carried out a considerable amount of model testing and analytic work to give estimation on ice loads, responses, and capabilities of moored vessels in drift level or pack ice.

Ice forces, vessel's responses, and the resulting mooring forces that occur during ice interaction process with moored vessels are three key factors to be studied. Toyama and Yashima (1985) applied a numerical model and a model test to conclude that the surge response was greater at

low ice drift speeds. Sayed and Barker (2011) applied the particle-in-cell method based on a hybrid Lagrangian-Eulerian formulation to simulate the interaction between broken pack ice and a moored Kulluk platform. Aksnes (2010) presented a semi-empirical method incorporating probabilistic models based on the model test results. However, this model is limited to one-dimensional simulations in the surge direction only. Moored structures in ice conditions may be subjected to ice drift from different directions depending on variations in currents and wind. The contact geometry and thus ice failure are influenced by the different incident angles between the hull and ice motion. However, this model is limited to one-dimensional simulations in the surge direction only. Zhou et al. (2011, 2012) presented a 2D method in the horizontal plane for simulating level ice-hull interaction process, where the ice load during submersion, sliding, and accumulation were simplified based on empirical and analytical methods. The simplified numerical model was validated through full-scale measurements of a conical platform called the Kulluk.

Some researchers are interested in model tests in ice basins. A number of model tests were conducted by Comfort et al. (1982), Evers et al. (1983), and Nixon and Ettema (1988). Comfort et al. (1999) assembled an extensive set of ice model test data for floating and moored structures and presented the data in a common format to identify overall trends, and the Kulluk is also included as a typical structure. Recently, Aksnes et al. (2008) and Bonnemaire et al. (2008) carried out ice model tests of an Arctic Tandem Offloading Terminal with a focus on mooring forces in level and ridged ice. Later, Aksnes et al. (2010) and Bonnemaire et al. (2010) conducted ice basin tests on a moored offloading icebreaker in variable ice drifting directions. Zhou et al. (2013a) performed a series of ice model tests to investigate the ice load experienced by an icebreaking tanker. The tanker was towed through the unbroken ice sheet to simulate the interaction process. Then the ice loads were measured. Zhou et al. (2013b) applied a numerical model to simulate the dynamic ice loads acting on the

icebreaking tanker Uikku in level ice, considering the action of ice in the vicinity of the waterline caused by breaking of intact ice and the effect of submersion of broken ice floes. The numerical simulations were also compared with the measured data as given in Zhou et al. (2013a). Good agreement was achieved though there are some deviations between predicted and measured results for some cases due to uncertainties from input factors.

Conclusion

To sum up, moored ships have been widely used in ice-infested waters for many kinds of polar activities. The interaction between moored structures and ice is a complex process which is difficult to predict theoretically. Due to severe ice conditions, the structures may experience much downtime which is not desirable. How to ensure the overall safety and extend ice operation season for moored ship remains a challenge to industrial and academic worlds of polar engineering. Some main aspects for designing a moored vessel operating in icy waters could be taken as a reference in this chapter.

Cross-References

- [Design of Mooring System](#)
- [Numerical Simulation of Ice-Going Ships](#)
- [Offshore Structure Design Under Ice Loads](#)

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Mooring

► Thruster-Assisted Mooring

Mooring Anchor

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Synonyms

[Drag anchor](#), [Offshore mooring system](#), [Pile anchor](#), [Station keeping system](#), [Suction anchor](#)

Definition

The passive mooring is the most commonly adapted station keeping method for offshore

floating facilities. An offshore mooring system usually consists of one or multiple mooring lines that connect the floater to the seabed to retain floater's position and limit offset to enable designated operations in various environments and water depths.

A mooring system is made up of mooring lines, anchors, and connectors, and the mooring line connects the floater to the seabed with its lower end terminate at the anchor at sea floor. The anchor is a physical device that is connected with the mooring line and penetrates into the seabed soil to hold the mooring line into position (Ma et al. 2019).

The mooring system is designed to withstand the environmental loads and operation loads in an offshore environment. Thus, the anchor holding power is a critical link of mooring system strength.

Historical Development

The history of the anchor dates back millennia. The most ancient anchors were probably made of rocks to hold ship at sea. However, using pure mass to resist the forces of a storm only works well in certain situations. Originated from anchor of rocks, advances in craftship and metallurgy encouraged development of improved shapes for more compact, durable, and efficient anchors. The modern navigation uses metal device with efficiently designed shape to connect a vessel to the seabed to prevent ship from drifting due to wind, waves, and current.

Over the years, the anchor technology has been continuously developed. The anchor design takes into account the soil resistance data and the requirements of horizontal and vertical holding capacities for safe mooring application.

For certain marine application, continuous deploying and retrieving anchors is an operation requirement, and anchors that were designed for such function have been developed.

Nowadays there are the following main types of mooring anchors (Vryhof 2015):

1. Drag embedment anchor
2. Vertical Loaded Anchor

3. Gravity installed deep penetration anchor
4. Pile anchor
5. Suction anchor
6. Gravity anchor

Working Principles of Anchors

The working principle of anchor is through its own weight or penetration into the seabed to interact with the soil to generate holding capacity. Therefore, there are two primary factors for anchor design.

1. Structural factor that is the material and shape of the anchor should have adequate strength and life expectancy
2. Geotechnical factor that is the soil characteristics, anchor size and depth of penetration should generate sufficient holding capacity

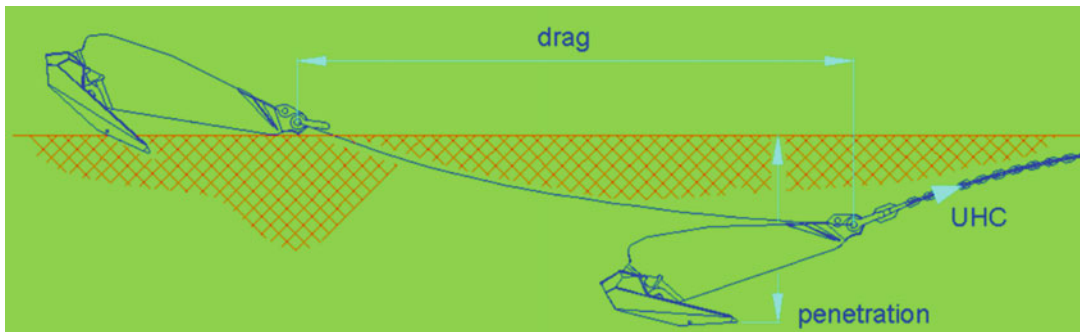
Drag Anchor

A drag embedment anchor (DEA) is the most utilized anchor for mooring floating MODUs. The drag anchor is dragged along the seabed until it penetrates the required depth. As it penetrates the seabed, it uses soil resistance to hold the anchor in place. The drag embedment anchor is mainly used for catenary moorings, where the mooring line arrives on the seabed horizontally. It does not perform well under vertical forces (Ozmutlu 2012) (Figs. 1 and 2).

The main advantage of the drag embedment anchor is it is highly mobile and can be deployed and retrieved without using heavy marine equipment. The main drawback is that the anchor's final position is less predictable which would be a problem for permanent mooring systems.

Additional features of drag anchors include:

- Standard off the shelf equipment
- Wide range of anchor types and sizes
- High capacity (greater than 100,000 lb)
- Performs poorly in rock seafloors
- Lower resistance to uplift loading
- Holding power highly directional

Mooring Anchor,**Fig. 1** Drag embedment anchor (Courtesy of Vryhof)**Mooring Anchor, Fig. 2** Drag anchor during installation. (Courtesy of Vryhof)

There are many designs and brands of drag embedment anchors. Following is a brief summary on a few well-known anchor types and their specific uses.

- The Stevpris New Generation accommodates the widest range of soils possible and its holding power is more than 30% higher than any other existing drag anchor.
- Stevpris Mk 5 is easy to handle, install, and retrieve and is used to secure semi-submersible rigs and permanent installations.
- The Stevshark Mk 5 does well in hard soils such as limestone, calcarenite, very dense sands, and coral.
- The Stevin Mk 3 was the original anchor design for Vryhof and outperformed all existing anchors at the time of its inception.
- The Bruce Mk 4 Twin Shank offers a choice of fluke angles for superior holding power in all

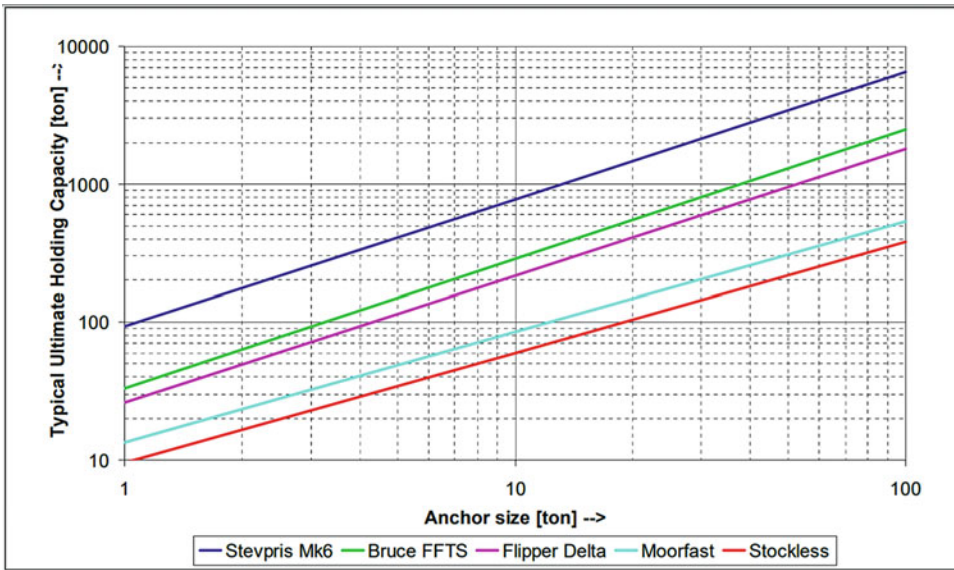
types of seabeds. It also has self-righting ability, even when it lands upside down.

- The Flipper Delta has an open construction for smooth penetration in different kinds of soil and no rotation, which means no decrease of holding capacity and no dragging of the anchor.
- Danforth is used for boats up to 27 ft. The I-beam fluke construction provides exceptional holding power.

The anchor holding capacity of drag embedment anchor is function of soil conditions, anchor type, anchor size, and fluke/shank angle setting. Designers often reply upon anchor capacity chart to select the suitable anchor (Fig. 3).

Vertical Loaded Anchor

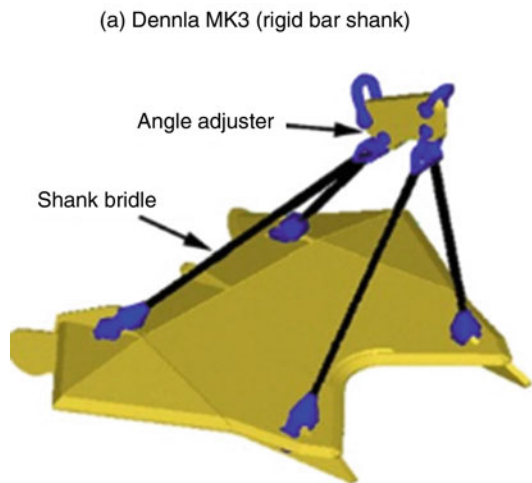
The conventional drag embedment anchor does not perform well with vertical mooring load.



Mooring Anchor, Fig. 3 Anchor holding capacity. (Courtesy of Vryhof)

However, development has been made of Vertical Loaded Anchor (VLA) which would allow certain amount of mooring vertical toad. Vertical load anchors are similar to drag anchors as they are installed in the same way. However, the vertical load anchor can withstand both horizontal and vertical mooring forces. It is used primarily in taut leg mooring systems, where the mooring line arrives at an angle the seabed.

For example, the Stevmanta Vertical Loaded Anchor (VLA) allows uplift at the anchor point, which is a requirement in semi-taut and taut leg mooring systems. Its main features include: deep penetrating anchor in soft soils, suitable for supporting large vertical and horizontal loads, easy installation like a conventional drag embedment anchor and loading in all directions possible (Fig. 4).



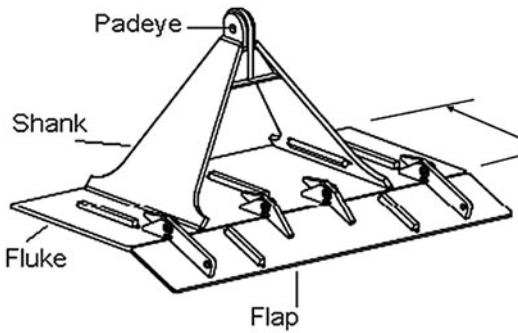
Mooring Anchor, Fig. 4 Stevmanta vertical loaded anchor. (Courtesy of Vryhof Anchor)

SEPLA Anchors

The patented SEPLA was introduced in 1997. It was a revolutionary concept for deepwater moorings that combines two proven anchoring concepts (i.e., suction installed piles and plate anchors capable of resisting vertical loads) to increase the accuracy of anchor positioning and burial depth for preset moorings. It gave the

industry a way of realizing the holding capacity of a suction installed anchor pile without the higher cost.

SEPLA anchors are believed to be more geo-technical efficient than the suction piles that are commonly used to withstand vertical loads. The SEPLA plate anchor is fully embedded in deeper, higher strength soil layers with all components



Mooring Anchor, Fig. 5 SEPLA anchor. (Courtesy of OnePetro)

actively resisting the mooring loads. On the other hand, a large part of a suction pile's structure is in shallower, weaker soils and does not contribute much to the holding capacity (Fig. 5).

Torpedo Anchors

Torpedo piles, essentially gravity-embedded cylindrically shaped projectiles, are used to anchor deepwater flowlines and facilities offshore. Torpedo piles typically range in size from 24 to 98 tons, and the largest torpedo pile can provide an anchor-holding capacity of up to 1000 tons (Fig. 6).

Pile Anchor

For mooring situations that the accurate positioning of the anchor is important, there is presence of vertical mooring load, the seabed surface soil consists of soft soil, and the drag embedment anchor becomes unsuitable. One possible alternative is to use pile anchor made of tubular steel pipe which is driven by underwater hammer to the desired penetration below the seabed surface. The mooring line is connected to the pile anchor through a bracket prewelded before pile installation (Fig. 7).

The behavior of a pile under lateral and axial loading interacting with the seabed soil is analyzed to determine the optimal length and diameter of the anchor pile, the position of the bracket along the pile, and the depth below the mudline to which the pile should be driven. Anchor piles generate lateral capacity through passive resistance of the soil bed and axial



Mooring Anchor, Fig. 6 Torpedo anchor. (Courtesy Deep Sea Anchors)

capacity by friction or adhesion along the pile shaft. In most cases, the mooring line connection is located below the top of the pile and it can vary from $\frac{1}{2}$ to $\frac{1}{3}$ of pile penetration.

The most common piles are long slender tubular piles (L/D ratio $> \sim 10$), which are typically fabricated from rolled steel sections. Diameters are in the range of 2 – 8 feet for the large mooring systems.

During installation, the pile is first lowered on the seafloor using a crane and gradually penetrates under the self-weight through a guide frame. The driving head is installed on top of the pile and underwater hammer is used to drive the pile to its design depth. When the seafloor is rock or composed of thin sediment over rock then piles cannot

Mooring Anchor,
Fig. 7 Pile anchor
 deployment. (Courtesy of
 InterMoor)



be installed by driving. In this case, an oversize socket must be predrilled for a pile to be inserted and grouted in place.

The key features of pile anchor include:

- High lateral capacity achievable
- Resists high uplift load
- Performs well on substantial slopes
- Suitable hard seafloors (rock and coral) by drilling and grouting
- Need quality site soil data
- Installation cost high

Suction Anchor

The suction pile is a relatively new type of anchor system. It is made of tubular steel pipe with sealed top and is driven into the seabed using pump to suck out the water from the top of the tubular, which creates negative pressure inside the pipe. The application has been growing steadily in the offshore industry particularly for soft soil in deep water. Suction piles can be installed from a single workboat making installation quick, relatively inexpensive, and less weather dependent than for driven or drilled piles.

Suction piles can be more expensive to fabricate since they are welded construction but they are easier and cheaper to transport. Suction piles

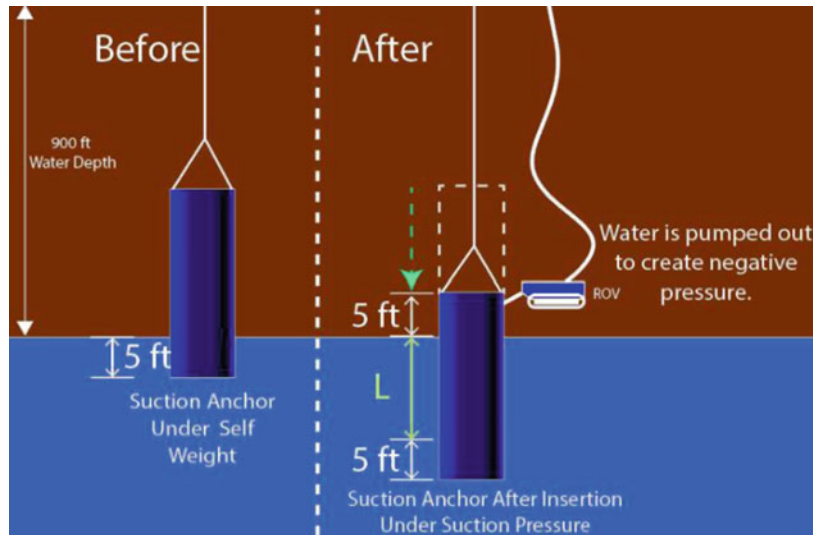
employ a lower slenderness ratio (length/diameter) than tubular piles. They are shorter and with larger diameter, ranging up to 10 m for soft soil. There are a number of suction pumps available including some that are operable by a remotely operated vehicle (ROV) a capability that provides flexibility in installation. An important feature of suction piles is their ability to be extracted and recovered by reversing the pump to apply pressure inside the pile (Fig. 8).

Suction piles are the predominant mooring and foundation system used for deepwater development projects worldwide. Suction piles can be used in sand, clay, and mud soils, but not gravel, as water can flow through the ground during installation, making suction difficult. They are also effective in normal sand seafloors but are not appropriate for hard bottom conditions. Once the pile is in position, the friction between the pile and the soil holds it in place. It can resist both vertical and horizontal forces.

The key features of suction pile anchors include:

- Requires less specialized installation equipment
- High lateral and vertical capacity
- Needs extensive and better quality soil data

Mooring Anchor,
Fig. 8 Suction pile anchor.
 (Courtesy of Drill
 Formulas)



Mooring Anchor,
Fig. 9 Gravity anchor.
 (Courtesy of Subcon)



- Removable by reversing installation pump
- Cost competitive installation

might provide the only reasonable anchoring option (Fig. 9).

The main features of gravity anchors include:

Gravity Anchor

A gravity anchor is made of heavy object placed on the seafloor to resist vertical and/or lateral loading. It is typically fabricated from concrete and steel and configured to enhance lateral capacity.

Gravity anchors are often used because they are inexpensive and readily sized for most seafloor and loading conditions. Especially there is less requirement of anchor position accuracy and lack of specialized installation equipment. In general, they are not very efficient (ratio of holding capacity to weight) compared to the other anchor types. They may require heavy lift capabilities for installation and they are a poor choice on sloping seafloors. However, on very hard bottoms they

- Resist large vertical load
- Reliable holding capacity
- Simple on-site construction
- Material readily available and economical
- Reliable on thin sediment over rock
- Not suitable in seafloor slope
- Size limited by load handling equipment
- Lateral load to weight ratio is low

Key Applications

Since the anchor connects the mooring line to the seabed, it is a critical component for the integrity of mooring system. The various anchor types find

their applications in different types of mooring systems.

For permanently mooring systems, the suction pile and driven pile are most commonly used for its reliability and location accuracy. The high-efficiency drag anchor and VLA have also been used for small floating units under mild environment.

For drilling operations with MODUs, the drag anchors are most commonly used for easy deployment and retrieval without specialized facilities. However, they have limited capability to withstand vertical loads. VLAs are the alternative where high vertical loads at anchor are present. Driven pile or suction pile are only used under special conditions such as very high vertical load or anchor movement strictly prohibited. Because of the temporary nature of the operation, a thorough soil investigation is normally not conducted, which makes the anchor design more challenging.

Torpedo pile and gravity-installed anchor are relatively new anchor concepts, which have been used in MODU and permanent moorings. The anchor design requires the full range of geotechnical analysis.

In situations where penetration into the seabed is not possible, for example, rock seabed, dead weight anchors made of reinforced concrete or scrap steel may be used. Vertical capacity is simply the submerged weight of the anchor.

Cross-References

- [Drag anchor](#)
- [Mooring system](#)
- [Suction anchor](#)

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Mooring Connector

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Synonyms

[Anchor shackle](#); [Chain connector](#); [D-shackle](#); [Offshore mooring system](#); [Station keeping system](#)

Definition

The passive mooring is the most commonly adapted station keeping method for offshore floating facilities. An offshore mooring system usually consists of one or multiple mooring lines that connect the floater to the seabed to retain floater's position and limit offset to enable designated operations in various environments and water depths.

A mooring line consists of line segments of different materials, such as chain, wire rope, and polyester. The connector is a physical device that connects different line segments and anchors.

The mooring system is designed to withstand the environmental loads and operation loads in an offshore environment. As part of the strength-carrying element, the connectors shall have compatible breaking load as other mooring components (API 1981).

Historical Development

The traditional mooring line for station keeping of ships in harbor was typically made with chain of uniform size. As the offshore moorings evolved, mooring systems become more complex with mooring lines made of different segments to achieve the desired performance characteristics. As such, a mooring line usually consists of a number of segments with different materials such as chain, wire rope, polyester, etc. Even if a mooring line is made of a single type of material, the line may be segmented for the purpose of transportation and installation. Therefore,

connectors are used to connect the various mooring line segments.

There are many types of connectors for connecting different types of mooring components. The commonly used ones include D-shackles for connecting the chain segments, D-shackle in association of wire rope termination for connecting the chain to wire rope, H-link for connecting the chain to polyester segment, etc. (Ma et al. 2019).

Working Principle

Since the connector is a load-carrying link within a mooring line, the strength and design life of the link must be compatible with those of mooring line segments to maintain the basic integrity and reliability of the complete mooring line. The design of the mooring link should take the following factors into consideration:

- Adequate strength
- Adequate fatigue life
- Physically compatible with segment terminations
- Ease of connection and disconnection

For permanent mooring systems, D-shackle and H-link are the two most commonly used



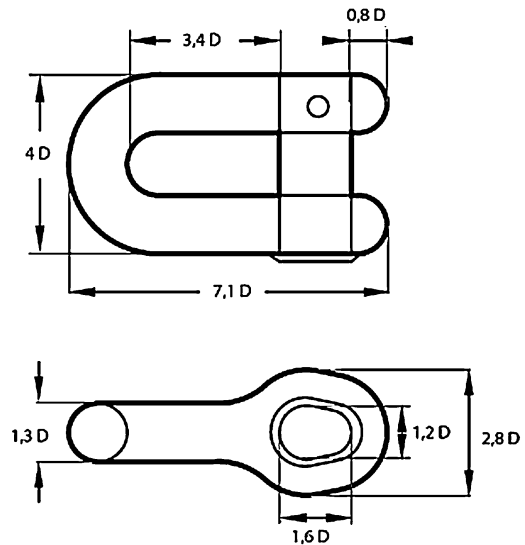
Mooring Connector, Fig. 1 Common D-shackle. (Courtesy of Zhengmao)

types for connection among the chain, wire rope, and polyester. Because inspection and replacement of connectors in a permanent mooring are difficult and costly, their designs need to be robust with adequate fracture toughness, fatigue life, and corrosion resistance. Manufacturing of connectors should be subject to an appropriate level of quality assurance corresponding to the same level as those of mooring chain.

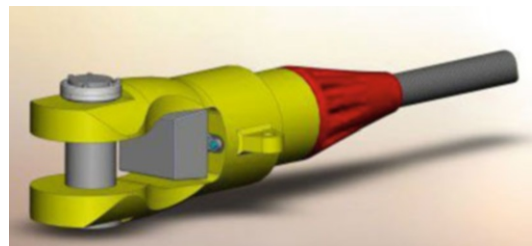
Applications

D-Shackle

The D-shackle is most commonly used for joining the adjacent chain segments. It consists of a bow,



Mooring Connector, Fig. 2 Common D-shackle measurement. (Courtesy of Sotra Marine)



Mooring Connector, Fig. 3 Wire rope termination. (Courtesy of Lebeon)

which is closed by a pin with single or double locking nut. The pin can be of rounded shape or oval shape depending on the purpose of application (Fig. 1).

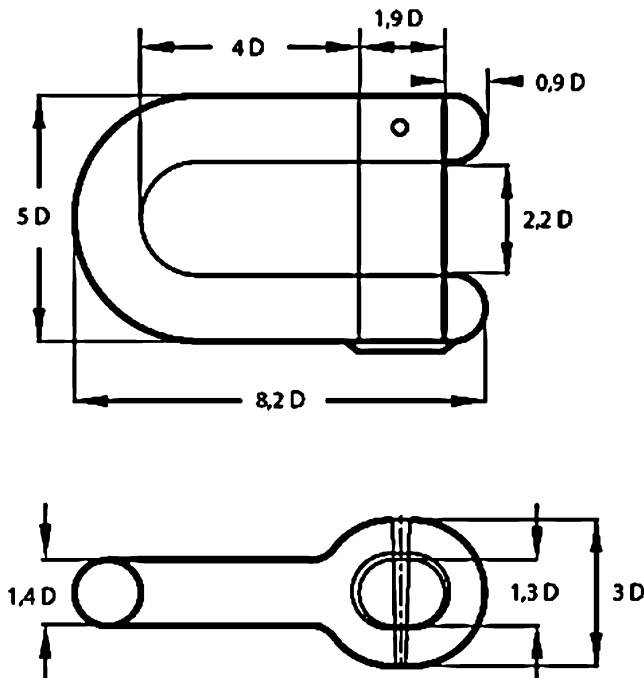
The shackle is made by forging and with corresponding grade as per chain links. D-shackle can be used in both temporary and

permanent moorings. The dimensions of the D-shackle are standardized as illustrated by the sketches below (Fig. 2).

D-shackle can also be used to connect chain segment to wire rope in association of the closed or open socket termination of the wire rope (Fig. 3).

Mooring Connector,

Fig. 4 Anchor shackle measurement. (Courtesy of Sotra Marine)



M



Mooring Connector, Fig. 5 H-link. (Courtesy of Vicinay)



Mooring Connector, Fig. 6 H-link in polyester connection. (Courtesy of Oceanside Equipment)

Anchor Shackle

Anchor shackle is another type of D-shackle that connects the chain segment to the mooring anchor. Like a common D-shackle, it consists of a bow, which is closed by a pin with single or double locking nut. The pin is of rounded shape for free rotation.

This type of D-shackle is made by forging and with corresponding grade as per chain links. It can be used in both temporary and permanent moorings. The dimensions of the anchor shackle

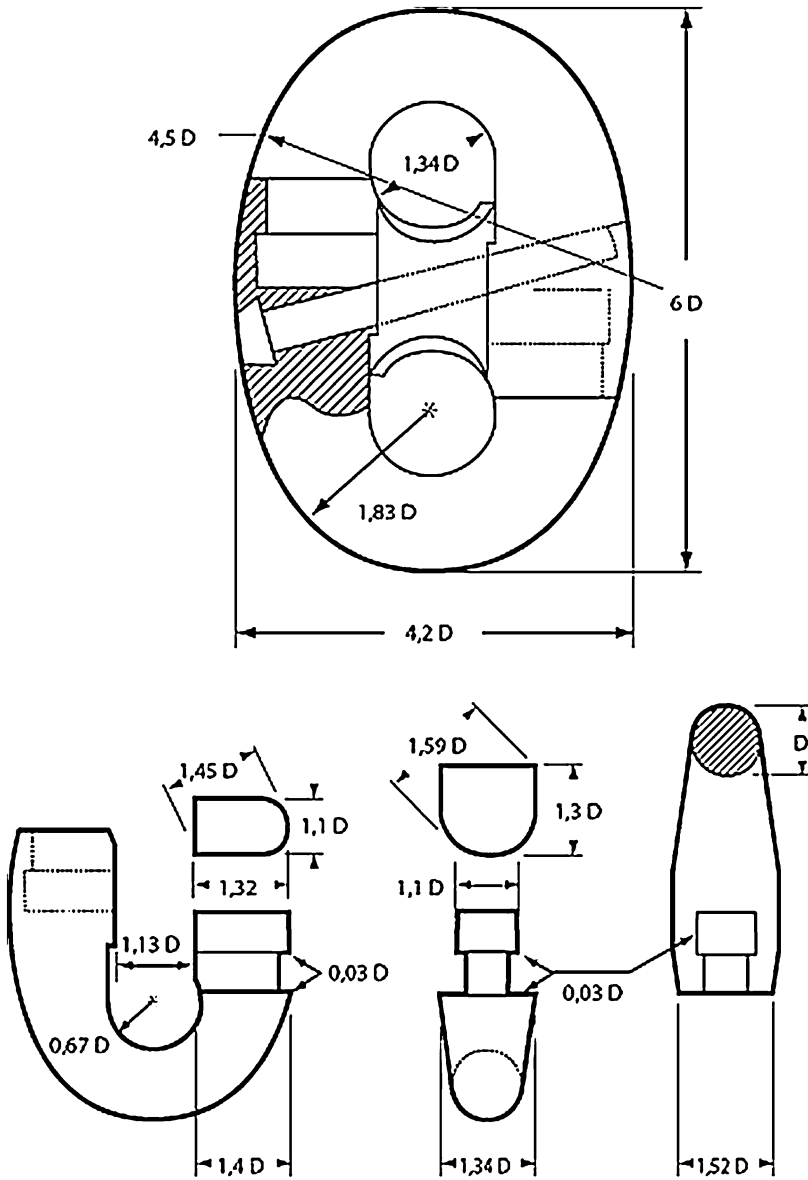
are also standardized as illustrated below (Fig. 4).

H-Link

The H-link is commonly used in mooring lines for connecting segments of the line. This type of connector has been traditionally used for connecting chain segments. Lately the H-link is introduced to connect polyester rope with chain segment. The main feature is to avoid time-consuming handling of bulky thimbles for the

Mooring Connector,

Fig. 7 Kenter link and application. (Courtesy of Sotra)



eyes of polyester ropes. They can also be used to connect two lengths of polyester ropes in deepwater mooring lines (Figs. 5 and 6).

Kenter Link

The connecting kenter-type link is most commonly used for the connection of two pieces of chain mooring line, where the terminations of the two pieces have the same dimensions. It has the same outside dimension as a chain link of the same diameter. It therefore allows the connected chain piece going through mooring devices such as fairlead, chainstopper, windlass, etc. The kenter link is commonly used to connect chain of mobile offshore drilling units which require frequent handling of mooring lines. Generally speaking, the kenter link is not used in permanent mooring systems, first because the frequent line handling is not a requirement and second, the kenter link has a shorter fatigue life than the chain of compatible strength.

The kenter-type joining link contains three parts along with taper pin and lead plug. The two main halves have numbers to be matched and arrows to be lined up for ease of assembly with the third piece (stud). The two main parts are attached to the ends of the chain in a vertical position and then fitted together, and the stud is then slid into place, which locks the link. The stud is secured by hammering a tapered pin into the hole drilled diagonally through all three parts of the joining link (Fig. 7).

Pear Link

The pear-shaped connecting link is, in essence, similar to kenter link and C-link, except that it is used for the connection of two pieces of mooring chain with different dimensions. Like kenter link and C-link, the pear-shaped connecting links are not used in permanent mooring systems.

Note there is a product called trident link that combines the features of kenter and pear links. Trident link is mainly used to connect an anchor shackle directly to a chain, thus reducing the amount of required connections. In this way, it is very similar to a pear Link with the only notable differences being the method of assembly. This type of shackle can also be used in a variety of

other applications where the crossover from one size to another is required.

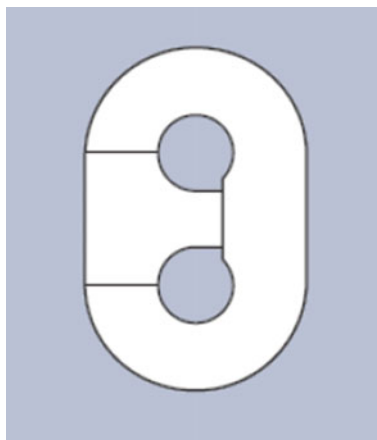
The pear link is often used during offshore installation, for example, where a smaller piece of line is connected to pull in the pre-laid mooring lines (Fig. 8).

C-Link

Like the kenter link, the connecting link C-type is used for the connection of two pieces of mooring chain with terminations that have the same dimensions. The major difference between the kenter type and the C-type is the way that the connector is opened and closed (Fig. 9).



Mooring Connector, Fig. 8 Pear link. (Courtesy of Vryhof)



Mooring Connector, Fig. 9 C-Link. (Courtesy of Vryhof)



Mooring Connector, Fig. 10 Swivel link. (Courtesy of Zhengmao)

twist and torque that builds up in the mooring line (Fig. 10).

The swivel can be placed at various locations along the mooring line. There are many different types of swivels available. One of the main design concerns is that when loaded, the swivel link may lock up due to the high friction inside the turning mechanism. Some newly designed swivels are fitted with bearing to enable swiveling under load.

Triplate

Mooring triplate can also be used in association with D-shackles to connect the mooring line segment. It is often used for handling during mooring installation (Fig. 11).

Cross-References

- ▶ [Anchor Shackle](#)
- ▶ [D-Shackle](#)
- ▶ [Mooring System](#)

References

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Mooring Connector, Fig. 11 Mooring triplate. (Courtesy of Le Beon)

Swivel Shackle

A swivel shackle can be used in a mooring line, generally of a temporary nature, to relieve the

Mooring Lines

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Introduction

Mooring lines provide the connections between floating facilities and anchoring foundations embedded in the seabed and are used for station keeping of floating facilities. Mooring line is

normally made up of steel chain links, wire ropes, and synthetic fiber ropes or a combination of the three.

In shallow water zone including rivers and lakes, mooring lines are used for limiting the shift of boats and barges, as shown in Fig. 1a. The most common choice for mooring line component is usually chain. In most practices, single line mooring is the most popular mooring pattern, where two-line mooring is also widely used. In offshore oil and gas industry, mooring lines are used for limiting the movement of offshore vessels and platforms to control the offset of risers and drill pipes connected to these floating facilities. The mooring patterns in deep water are usually using groups of mooring lines (see Fig. 1b), which can adopt a spread mooring, single point mooring, or turret mooring. Steel wire rope is a better choice in water depths between 300 and 2000 m due to its lighter weight and higher elasticity than chain. In ultra-deep water greater than 2000 m, multicomponent mooring line is

generally used, which is a combination of chain, wire rope, and synthetic fiber rope.

Mooring lines, functioning as a station-keeping component of an anchoring system, play the crucial role in safe operation of the moored floating facilities. The floating facilities would be out of control in case of failure of the mooring lines, which may cause serious casualties, property losses, and damage to marine environment. It is significant to understand the behaviors and failure mechanisms of mooring lines.

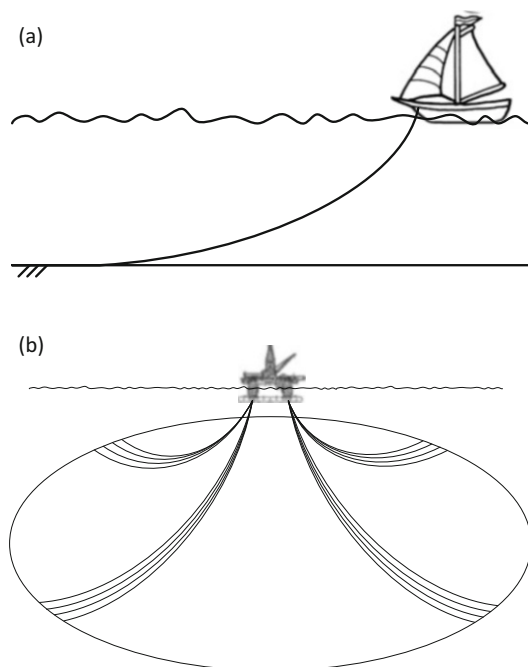
This entry is to provide readers with a general introduction of mooring lines, especially for offshore practice. The content includes historical development of mooring lines, materials, configurations and layouts in industry practice, and analytical solutions of mooring line behaviors.

Performance of Mooring Lines

Historical Development of Mooring Lines

Natural fiber rope was probably human's first invention used for mooring boats and ships. Tales of early anchor chain usage come from China. Around 2200 B.C., the Emperor Yu is reported to have used "iron chains, two fore and aft, which were thrown overboard to steady and stop the vessel." However, the iron chain was hardly used to moor vessels before the seventeenth century. Natural fiber rope was still used for mooring until the nineteenth century before steel chain and wire rope eventually replaced it. Steel chain was successfully produced by electric welding after World War I, which contributed to the rapid and wide spreading of chains. Synthetic fiber materials, including nylon, polypropylene, polyester, aramid, and polyethylene, were introduced into marine mooring around the mid-twentieth century (Flory et al. 2015). Nowadays, synthetic fiber ropes are becoming a promising alternative to wire ropes in many mooring applications, especially for oil production platforms in deep water.

As the offshore oil and gas industry is moving toward deep water, mooring patterns including spread moorings and single point moorings were used to withstand ocean environmental load.



Mooring Lines, Fig. 1 Mooring lines in shallow and deep water. (a) Mooring line for boat, (b) mooring lines for semisubmersible

Spread moorings are mainly used for semisubmersibles and spars, while single point moorings tend to be used more for ship-shaped vessels (API 2005).

Mooring Lines Components

A mooring system comprises mooring lines, connectors, anchors, and winches or windlasses. The mooring line is usually made up of chains, wire ropes, synthetic fiber ropes, or their combination.

Chain is the most common product type used for mooring lines. There are mainly two different designs of chain, studlink and studless chain, as shown in Fig. 2a. The studlink chain is commonly used for mobile moorings where the mooring line can be reset numerous times during their lifetime, such as semisubmersibles and mobile offshore drilling units (MODUs). The studless chain is often used for permanent moorings (such as a floating production storage and offloading (FPSO)) with design lives over 10 years. The studless chain has lower weight per unit of strength and less manufacture expense than the studlink chain.

Wire rope is made up of individual wires wound in a helical pattern to form a “strand” (Brown 2005). The types of wire rope used as mooring line include multistrand and single-strand rope. Multistrand wire rope, also called stranded rope, consists of a number of strands wound in the same rotational direction around a center core to form the wire rope (DNV 2013). The center core may be either a fiber or a metallic core to support the outer wires. As shown in Fig. 2b, six-strand rope is the most common type of multistrand rope and often used for temporary mooring. Single-strand rope, also called spiral rope, uses bundles of wires wound in opposing

directions to obtain the torque-balanced characteristics. Single-strand rope has longer fatigue life than the multistrand rope and thus is more commonly used in permanent mooring.

Typical materials of synthetic fiber rope that are currently used for mooring are polyester and high modulus polyethylene (HMPE), as shown in Fig. 2c. High-performance fiber rope materials, including aramid, HMPE, liquid crystal aromatic polyester (LCAP) fiber, and so on, have been introduced into the mooring since the mid-1960s. The major advantages of synthetic fiber ropes are their light weight and elasticity of the material (Vryhof Anchors 2015). The synthetic fiber ropes are usually employed with combination of chain and wire rope in deep water. During deployment, it is normally not permissible for fiber rope to contact with the seabed unless specifically qualified, because the ingress of foreign particles such as sand will affect the rope’s yarn-on-yarn abrasion resistance and hence will adversely affect the rope’s fatigue life (DNV 2008).

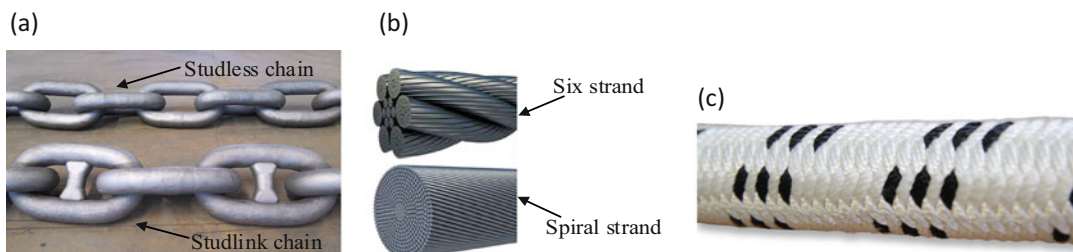
Connectors

Connectors are the components to be used to connect two different segments of a mooring line or a mooring line to vessels or to padeyes on anchors. Typical connectors include Kenter shackle, D-shackle, C-link, pear link, and swivel, as shown in Fig. 3.

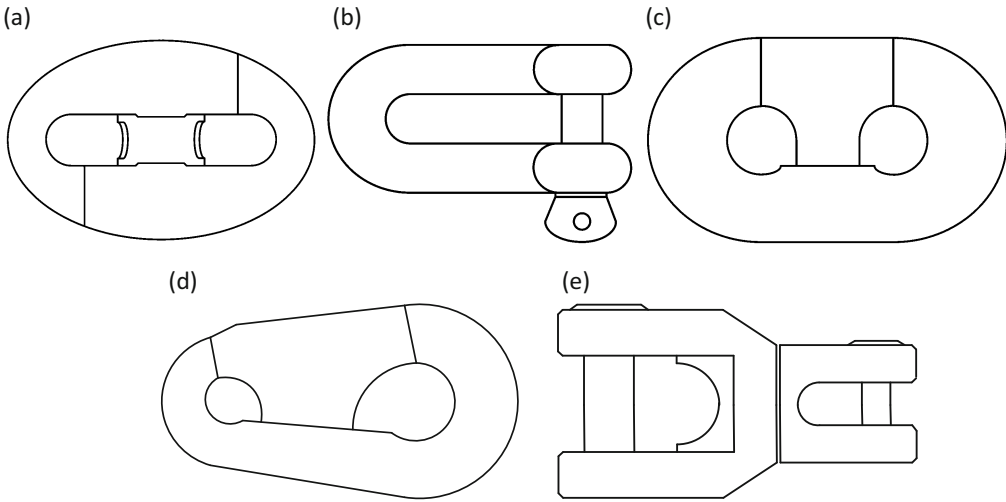
Mooring Configurations in Water Column

Mooring configurations can be catenary, taut, or semi-taut line or vertical moorings, as shown in Fig. 4.

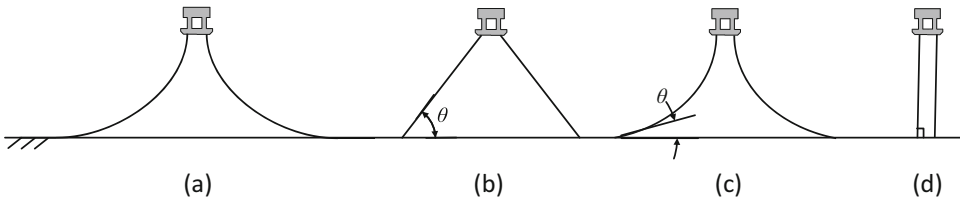
A catenary mooring configuration presents a mathematical catenary curve between the



Mooring Lines, Fig. 2 Mooring lines components. (a) Chain, (b) wire rope, (c) HMPE rope



Mooring Lines, Fig. 3 Typical connectors. (a) Kenter shackle, (b) D-shackle, (c) C-link, (d) pear link, (e) swivel



Mooring Lines, Fig. 4 Mooring configurations. (a) Catenary mooring, (b) taut line mooring, (c) semi-taut line mooring, (d) vertical mooring

floating facility and the seabed. The mooring line is suspended from the floating facility and touches down on the seabed. Therefore, an obvious characteristic of catenary mooring line is part of the mooring line lays on the seabed. During operation, most of the restoring forces to the floating facility are generated by the weight of the mooring line (Randolph and Gourvenec 2011).

Compared with the catenary mooring that arrives at the seabed horizontally, a taut or semi-taut line mooring arrives at the seabed at an angle, resulting in both horizontal force and vertical force to the anchor. The restoring forces of the taut line are mainly generated by the elasticity of the mooring line. Synthetic fiber ropes that mainly include polyester ropes and polyethylene ropes are often used in taut line moorings. An advantage of a taut

line mooring superior to the catenary mooring is its shorter line length and smaller seabed footprint, which means the spread radius of taut line mooring is smaller than that of catenary mooring for a similar application. Semi-taut line moorings provide a greater seabed footprint than taut line moorings but reach similar maximum angles under critical design conditions (Randolph and Gourvenec 2011). Typical horizontal distance from the touch-down point normalized by the water depth ranges from 1 for taut line mooring, to 1.5 for semi-taut line mooring, and to 2 for catenary mooring.

Vertical moorings or tendon/tension leg moorings are used to anchor large production units, such as tension-leg platforms, in very deep water. Recently, vertical moorings are also used in small floating units.

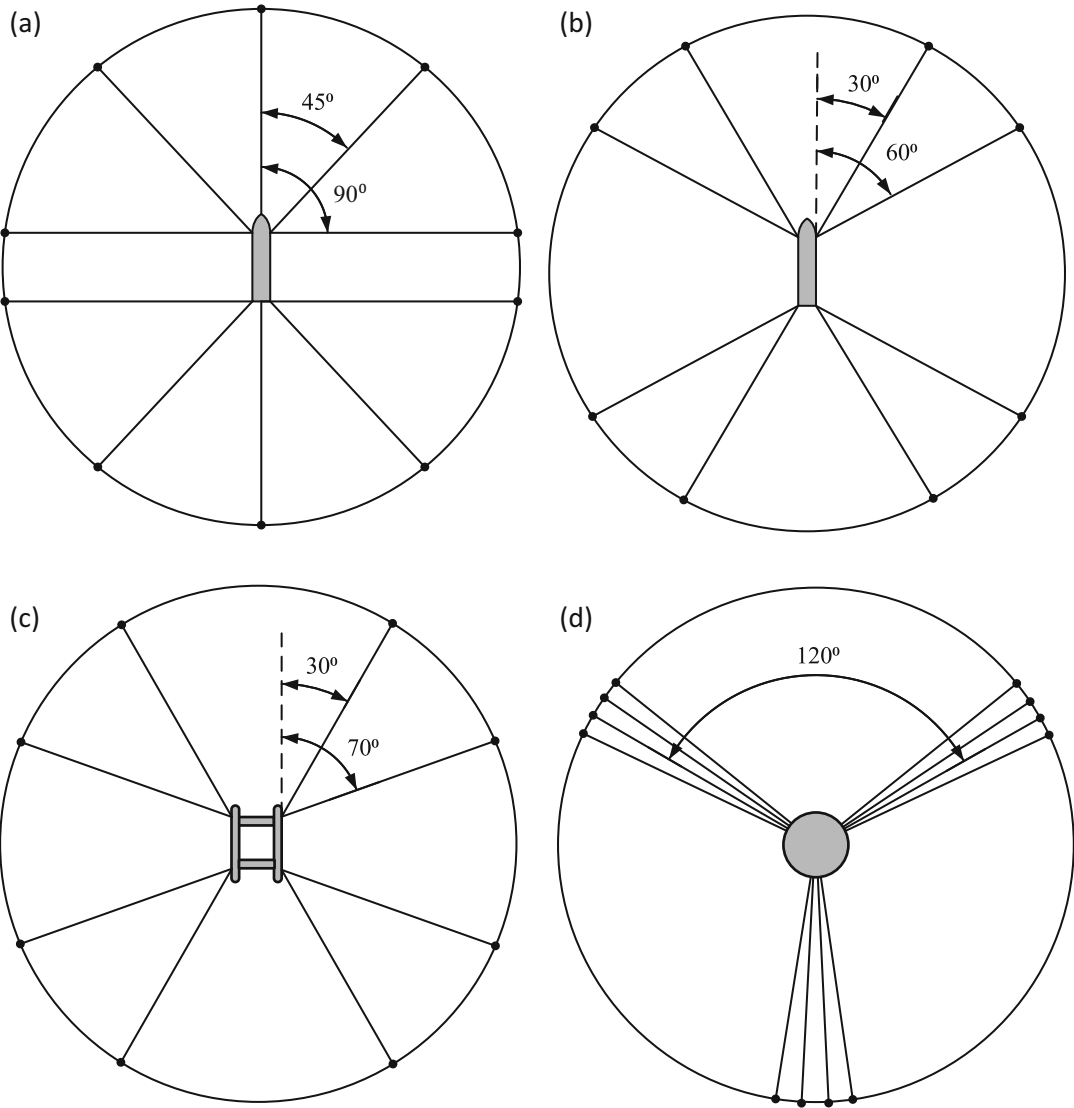
Offshore Mooring Systems Layout Patterns

Offshore mooring systems can be classified into spread mooring, single point mooring, and dynamic positioning.

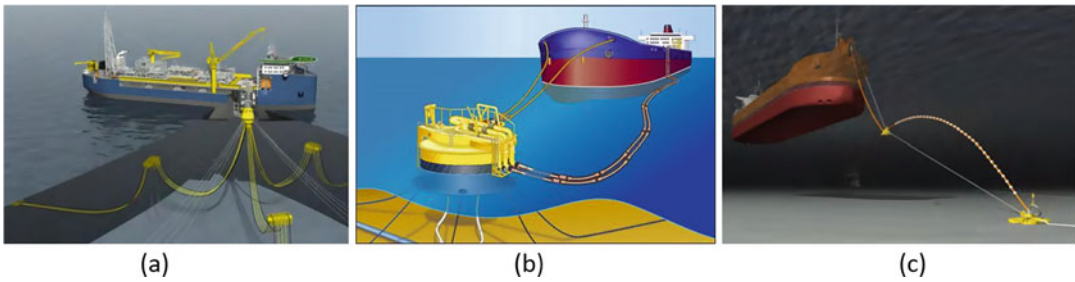
A spread mooring system consists of a group of mooring lines symmetrically distributed to the longitudinal center line of the platforms or vessels. The symmetrical arrangement of anchors keeps the heading of floating facilities being essentially fixed and prevent the rotation in the horizontal plane due to wind, waves, or current.

Figure 5 shows some typical offshore mooring patterns: Fig. 5a, b shows the patterns often used by vessels like drillship and FPSO, Fig. 5c shows the pattern used by the semisubmersible, and Fig. 5d shows the pattern used by a spar platform. Spread mooring is versatile because it can be applied to most vessels in varying water depth. However, large loads can be caused on the mooring line in ocean environment.

Single point moorings connect all the mooring lines to a single connection point that can be a



Mooring Lines, Fig. 5 Typical spread mooring (plan view). (a) Vessel mooring (ten lines), (b) vessel mooring (eight lines), (c) semisubmersible mooring, (d) spar mooring



Mooring Lines, Fig. 6 Single point moorings. (a) Turret mooring, (b) CALM, (c) SALM

buoy body or turret or tower. The vessel can rotate freely (i.e., weathervane) around the single point with the environment loads. There is a wide variety in the design of single point moorings, which typically include turret mooring, catenary anchor leg mooring (CALM), and single anchor leg mooring (SALM), as shown in Fig. 6.

A turret mooring system connects a turret assembly of a vessel, generally FPSO or floating storage and offloading (FSO), to the seabed by means of the mooring lines. The turret system that is fully passive contains a bearing system allowing the vessel to rotate around the fixed geostatic part of the turret. The turret mooring system is considered to be more economic and reliable and is widely used today.

The turret can be located externally or internally with respect to the vessel hull structure. For the external turret mooring system, the turret is located at the extreme end of an outrigger structure attached to the bow of the vessel. For the internal turret mooring system, the turret is integrated into the hull structure at the bow of the vessel, as shown in Fig. 6a.

Disconnectable turret mooring system has also been developed to further reduce the environmental loading on the mooring system in extreme conditions. The disconnectable turret uses a buoy that is located at the lower end of the turret. In a harsh ocean environment, the buoy can break away from the vessel and can still provide support to the risers and mooring.

The catenary anchor leg mooring (CALM) (sometimes called a single buoy mooring) consists of a large buoy held in place by a number of catenary mooring lines anchored to sea floor

(Olsen 2007). The vessel is fastened to a turntable or platform on the deck of the buoy through a hawser that is typically synthetic rope, or rigid arm or soft yoke system, so that the vessel can rotate around the buoy. The vessel can be easily evacuated from a harsh environment by unfastening the connections between the vessel and buoy.

For the single anchor leg mooring (SALM), a buoy under or on the water surface is anchored to an anchor point or base by a mooring line (taut wire rope or vertical chain), which is tensioned to pull the buoy against its buoyancy. The restoring force of the system comes from the buoyancy force of the buoy and the elasticity of the mooring line. The buoy is connected to the vessel through a hawser or yoke. In Fig. 6c, an underwater hose rises from the fluid swivel at the base or at a point above the base to the sea surface at a point away from the buoy for some distance.

Basic Mechanics of Mooring Lines

The mooring line of a vertical mooring is attached to the top of an anchoring foundation, and the one-dimensional responses of the mooring line in water are relatively easy to be analyzed. For catenary and taut line moorings, the behavior of the mooring line in water can be analyzed by solving the differential equations governing the segment of the line in water column. For catenary and taut line moorings using anchor piles, suction caissons, or drag anchors, the optimal attachment point of the mooring line is below the mudline. Therefore, it is essential to consider the performance of the anchor line in soil, which forms an inverse catenary shape due to soil resistance acting on the line. The behavior of embedded anchor line can be described by

solving the differential equations governing the segment of the line in soil.

Behavior of Mooring Line in Water

A mooring line element is shown in Fig. 7. The differential equations governing the segment of the line are:

$$(T + dT) \cos(\theta + d\theta) - T \cos \theta = 0 \quad (1)$$

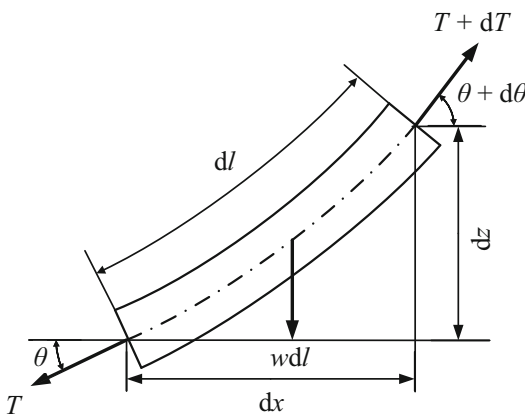
$$(T + dT) \sin(\theta + d\theta) - T \sin \theta - wdl = 0 \quad (2)$$

where T is the line tension, θ is the angle between the tension and horizontal direction, w is the submerged line weight per unit length, and l is the distance along the line. Provided that a mooring line is single-component (the submerged line weight per unit length is constant) and the elastic stretch is ignored, as shown in Fig. 8, the equations below can be obtained:

$$T_x = T_0 \cos \theta_0 = T_p \cos \theta_p \quad (3)$$

$$T_p = T_0 + wh \quad (4)$$

$$\begin{aligned} h &= \left(\frac{T_0}{w}\right) \\ &\times \left[\sqrt{\tan^2 \theta_p + 1} - \sqrt{\tan^2 \theta_0 + 1} \right] \\ &= \left(\frac{T_0}{w}\right) \left(\frac{1}{\cos \theta_p} - \frac{1}{\cos \theta_0} \right) \end{aligned} \quad (5)$$



Mooring Lines, Fig. 7 Force equilibrium of an inelastic mooring line element in water

$$S = \left(\frac{T_0}{w}\right) [\sinh^{-1}(\tan \theta_p) - \sinh^{-1}(\tan \theta_0)] \quad (6)$$

$$l = \left(\frac{T_0}{w}\right) (\tan \theta_p - \tan \theta_0) \quad (7)$$

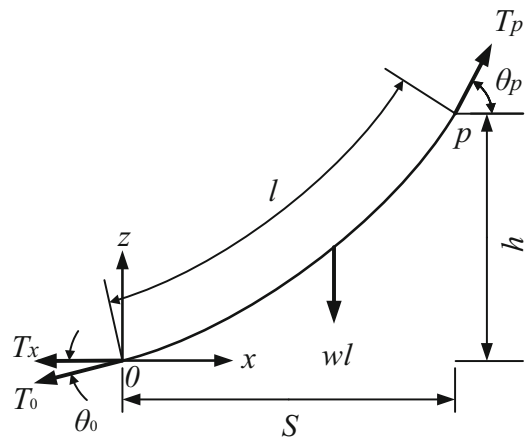
where the T_0 and T_p are the tensions at the touch-down point and top of the mooring line, respectively; T_x is the horizontal component of T_0 ; θ_0 and θ_p are the angles between the horizontal direction and the tension T_0 and T_p , respectively; h is the vertical distance from the touch-down point; and S is the horizontal distance from the touch-down point.

When $\theta_0 = 0$, the mooring state becomes the catenary mooring, and the following equations can be obtained:

$$\tan \theta_p = \frac{wl}{T_0} \quad (8)$$

$$T_p = T_0 + wh = T_0 + \sqrt{(wl)^2 + T_0^2} \quad (9)$$

$$\begin{aligned} h &= \left(\frac{T_0}{w}\right) \left[\sqrt{\tan^2 \theta_p + 1} - 1 \right] \\ &= \left(\frac{T_0}{w}\right) \left[\cosh\left(\frac{wS}{T_0}\right) - 1 \right] \end{aligned} \quad (10)$$



Mooring Lines, Fig. 8 Forces acting on inelastic mooring line in water

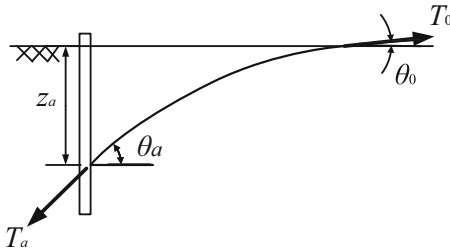
$$S = \left(\frac{T_0}{w} \right) \sinh^{-1} (\tan \theta_p) \quad (11)$$

A typical mooring line is usually multi-component line made up of connectors, wire rope, polyester rope, and chain. For the multi-component line, the tangential slopes of two components at the connectors are treated to be equal in order to solve the equations.

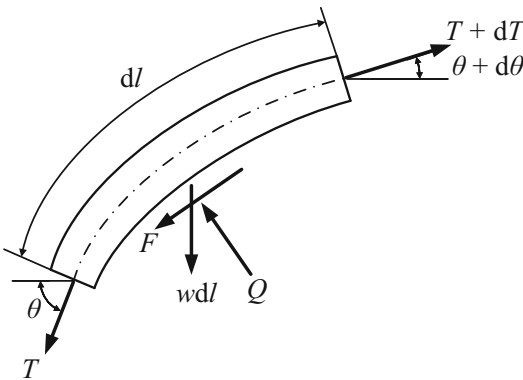
Behavior of Embedded Anchor Line in Soil

As shown in Fig. 9, the anchor line inclination at the attachment point of anchor or pile is critical in the design, because it will determine the mode of failure for anchor pile or suction caisson and the embedment performance of drag anchors.

Figure 10 shows the force equilibrium of a segment of anchor line. The differential equations governing the embedded segment of the anchor line are (Randolph and Gourvenec 2011)



Mooring Lines, Fig. 9 General configuration of anchor pile and embedded anchor line



Mooring Lines, Fig. 10 Force equilibrium of an embedded anchor line element

$$\frac{dT}{dl} = F + w \sin \theta \quad (12)$$

$$T \frac{d\theta}{dl} = -Q + w \cos \theta \quad (13)$$

where F is the soil friction (per unit length, parallel to the line) acting on the segment of the line, Q is the soil resistance (per unit length, normal to the line) acting on the segment of the line.

The soil friction F and normal soil resistance Q can be expressed as

$$F = A_s \alpha s_u \quad (14)$$

$$Q = A_b N_c s_u \quad (15)$$

where A_s is the effective surface area of the anchor line per unit length, α is the interface friction coefficient (typical values of α in clay range from 0.2 to 0.6 depending on whether wire rope or chain is used), s_u is the undrained shear strength, A_b is the effective bearing area of the anchor line per unit length, and N_c is bearing capacity factor. For wire rope, A_b is equal to the diameter of the rope d and $A_s = \pi d$. For a standard link chain, $A_b = 2.5d$ and $A_s = 8-11d$.

Neubecker and Randolph (1995) proposed a simplified analytical solution of an embedded anchor line, in which the self-weight of the anchor line is ignored. The analytical solutions are expressed as

$$\frac{T_a}{2} (\theta_a^2 - \theta_0^2) = \int_0^{z_a} Q dz = z_a Q_{av} \quad (16)$$

$$\frac{T_0}{T_a} = e^{\mu(\theta_a - \theta_0)} \quad (17)$$

where T_a is the embedded anchor line tension at the attachment point, θ_a is the embedded anchor line inclination at the attachment point, z_a is the depth to the padeye, Q_{av} is the average bearing resistance over the depth range of 0 to z_a , and μ is the friction coefficient and calculated by F/Q .

Cross-References

- [Catenary Mooring](#)
- [Diving of Anchors](#)

- ▶ External Turret Single Point Mooring System
- ▶ Fatigue of Mooring Lines
- ▶ Internal Turret Single-Point Mooring (SPM) System
- ▶ Mooring System
- ▶ Single Point Mooring
- ▶ Spread Mooring
- ▶ Taut Mooring

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Mooring System

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Synonyms

Dynamic positioning system; Single point mooring; Spread mooring

Definition

A mooring refers to any permanent structure to which a floating structure may be secured. Examples include quays, wharfs, jetties, piers, anchor buoys, and mooring buoys. A floating platform is secured to a mooring to forestall free movement of the structure on the water. An anchor mooring fixes a floating structure's position relative to a point on the bottom of a waterway without connecting the structure to shore. The mooring systems are usually composed of anchors, mooring lines, buoy, and connectors.

Scientific Fundamentals

Mooring system and dynamic positioning system are the two main methods to keep a floating structure positioning. They can be used with either ship-shaped or offshore floating structures.

Since ancient times, mooring system has been a traditional positioning way. Mooring system for ships is generally used to meet the needs of temporary berthing and prevents the ship from drifting freely on the water surface. It is mostly used in storm-free shallow water areas. Mooring system is also the most commonly used passive positioning form for offshore platforms. Considering severe ocean environment and the positioning requirements of deepwater drilling and production platform, the research, design, and manufacture of deepwater structure mooring system is so challenging that once become a technical bottleneck for deep-sea oil and gas exploitation. The research and development of mooring forms and mooring materials have been progressing steadily. Mooring ways change from traditional catenary mooring to tension mooring used in drilling and mining platform, such as SPAR and TLP platforms. Traditional high-quality mooring chains and wires are increasingly unable to meet the needs of the development of deep-sea drilling and mining, and the research and development of new mooring materials have been conducted (Luo 2015).

Usually there are three kinds of mooring systems according to the different positioning time. The mobile mooring system is used for pipe laying ship, lifting ship, and so on. Temporary mooring system is used for drilling ship or drilling platform. Permanent mooring system is used for the production of variety of floating platforms and FPSOs.

Mooring system can also be divided into catenary mooring system and tension mooring system according to the shape of mooring line. The former is mostly used for moderate depth and shallow water, and the latter has smaller mooring radius, length, and load than the former. Catenary mooring system is a traditional deployable mooring system. It has a long history and can adapt to harsh marine environment. It still plays an important role in current positioning technology of deepwater offshore oil and gas platforms. Tension mooring system is gradually developed with the application of fiber materials in deep sea. It was first applied to the tension leg platform and Spar platform in the 1980s.

According to the arrangement of lines, there are single point and spread mooring system. The former is mostly used for ship-shaped platform. In 1959, the first set of CALM (Catenary Anchor Leg Mooring) system designed by American IMODCO Company was successfully made in Sweden. It unveiled the prelude to the application of single mooring technology in offshore oil exploitation and offloading. From 1976 to 1985, single point mooring for FPSO experienced the initial mooring development phase of offshore oil and gas production. Single mooring technology welcomed its growth stage from 1986 to 1994, and extension stage from 1995 to 1998. Since 1999, the number of FPSO has increased rapidly. Major breakthroughs of mooring technology have been developed, especially for limits of water depth. Spread mooring is a natural extension of traditional ship mooring. It is suitable for Spar, Semi-submersible platforms, FPSO, and ship-shaped structures. And the environment conditions for spread mooring need to be moderate (Flory 2001).

In terms of mooring lines' materials, the typical materials have changed from steel chains and

wires to synthetic fibers. Steel chain is the most traditional mooring material. It has been widely used in single point and multi-point mooring of drilling and production ships and platforms since the early stage (Corbetta and Sloan 2001). Wire mooring line is made of steel wires, and its quality is lighter than steel chains, which is suitable for deeper-water mooring (Costa et al. 2001). With the development of tension mooring, synthetic mooring has become one of the key technologies of system research (Dove et al. 2000). Synthetic fiber line is usually made of polyester fiber (PET), high strength polyethylene (HMPE), aromatic nylon (Aramid), and so on, among which, polyester line is the most popular one. In the 1970s, foreign scholars began to study the traditional polyester fiber and nylon mooring lines as mooring materials. In 1996, a synthetic fiber mooring line consisting of PET, nylon, and HMPE was successfully installed on DeepStar platform. In 1997, a polyester rope + anchor chain catenary mooring system is first used for a FPSO in Brazil. Afterwards, more and more fiber mooring systems appeared in the Gulf of Mexico (Winkler and Mckenna 1994).

When oil and gas exploration and production was conducted in shallow to deep water, the most common mooring line configuration was the catenary mooring line consisting of chain or wire rope. For exploration and production in deep to ultra-deep water, the weight of the mooring line starts to become a limiting factor in the design of the floater. To overcome this problem, new solutions were developed consisting of synthetic ropes in the mooring line (less weight) and/or a taut leg mooring system (Vryhof 2005).

The major difference between a catenary mooring and a taut leg mooring is that where the catenary mooring arrives at the seabed horizontally, the taut leg mooring arrives at the seabed at an angle. This means that in a taut leg mooring, the anchor point has to be capable of resisting both horizontal and vertical forces, while in a catenary mooring the anchor point is only subjected to horizontal forces. In a catenary mooring, most of the restoring forces are generated by the weight of the mooring line. In a taut leg mooring, the restoring forces are

generated by the elasticity of the mooring line (API 2005).

Key Applications

The history of mooring system started from the development and use of the single point mooring device of catenary buoy by the US navy during World War II and then developed rapidly. Numerous different mooring systems have been developed over the years. Generally, the mooring systems are divided into single point mooring and spread mooring (DNV 2008).

Single Point Mooring

A single point mooring system connects all the lines to a single point. It links subsea manifolds connections and weather vaning tankers, which are free to rotate 360° . The single point system includes a buoy, mooring and anchoring elements, product transfer system, and other components.

Single point moorings are used primarily for ship-shaped vessels. They allow the vessel to weather vane. This is necessary to minimize environmental loads on the ship-shaped vessel by heading into the prevailing weather. There is wide variety in the design of single point moorings, such as Catenary Anchor Leg Mooring, Single Anchor Leg Mooring, Turret mooring.

CALM buoy – Generally the buoy will be moored using four or more mooring lines at equally spaced angles. The mooring lines generally have a catenary shape. The vessel connects to the buoy with a single line and is free to weather vane around the buoy.

SALM buoy – These types of buoys have a mooring that consists of a single mooring line attached to an anchor point on the seabed, underneath the buoy. The anchor point may be gravity based or piled.

Turret mooring – This type of mooring is generally used on FPSOs and FSOs in much harsh environments. Multiple mooring lines are used, which come together at the turntable built into the FPSO or FSO. The FPSO or FSO is able to rotate around the turret to obtain an optimal orientation relative to the prevailing weather

conditions. The turret can be mounted externally from the vessel bow or stern with appropriate reinforcements or internally within the vessel. The chain table can be above or below the waterline (Fig. 1).

Spread Mooring

The spread mooring system does not allow the vessel to weather vane, which means to rotate in the horizontal plane due to wind, waves, or current. Spread mooring is versatile as it can be used in any water depth, on any vessel, in an equally spread pattern or a group.

In a typical spread mooring system, groups of mooring lines are terminated at the corners of the vessel, holding a stable vessel heading. Since the environmental force on a semi-submersible or a spar is relatively insensitive to direction, a spread mooring system can be designed to hold the vessel on location regardless of the direction of the environment. However, this system can also be applied to ship-shaped vessels, which are more sensitive to environmental directions. The mooring line can be chain, wire rope, fiber rope, or a combination of the three. Either conventional drag anchors or anchor piles can be used to terminate the mooring lines (Baritrop 1998).

The catenary mooring system is the most commonly used system in shallow water. It gets its name from the shape of the free hanging line as its configuration changes due to vessel motions. At the seabed, the mooring line lies horizontally; thus, the mooring line has to be longer than the water depth. Increasing the length of the mooring line also increases its weight. As the water depth increases, the weight of the line lessens the working payload of the vessel. In that case, synthetic ropes are used. As water depth increases, conventional, catenary systems become less and less economical (ABS 2014).

The taut leg system typically uses polyester rope that is pre-tensioned until taut. The rope comes in at a $30\text{--}45^\circ$ angle on the seabed where it meets the anchor (suction piles or vertically loaded anchors), which is loaded vertically. When the platform drifts horizontally with wind or current, the lines stretch and this sets up an opposing force.



Mooring System, Fig. 1 (a) Typical internal turret mooring (b) typical external turret mooring

The semi-taut system combines taut lines and catenary lines in one system. It is ideally used in deepwater (SBM 2012).

Cross-References

- ▶ Catenary Anchor Leg Mooring
- ▶ Catenary Mooring
- ▶ Design of Mooring System
- ▶ Dynamic Analysis Method
- ▶ External Turret Single Point Mooring System
- ▶ Internal Turret Single-Point Mooring (SPM) System
- ▶ Single Anchor Leg Mooring
- ▶ Single Point Mooring
- ▶ Soft YOKE Single Point Mooring System
- ▶ Spread Mooring
- ▶ Station-Keeping System for VLFS
- ▶ Taut Mooring
- ▶ Turret Mooring

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Mooring System of Renewable Energy Devices

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Definition

A mooring system is a system of several mooring lines which are connected to a floating structure of renewable energy device at one end and fixed to the sea bed at the other end, to restrict the horizontal excursions and orientation of the floating structure.

General Purpose of Mooring System

A station-keeping system (ISO 2005) is to restrict the horizontal excursions of the floating structure

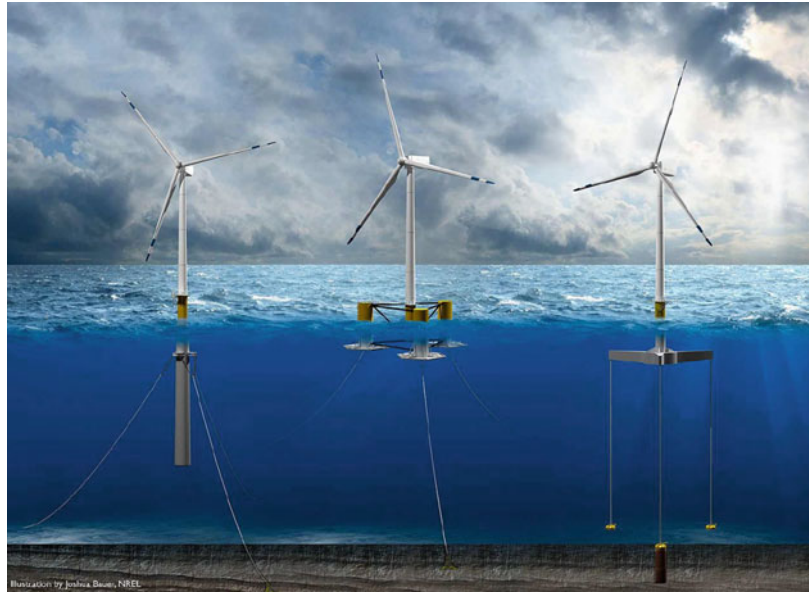
within prescribed limits, as well as to provide means of active or passive directional control when the structure's orientation is important for safety or operational considerations. For floating offshore renewable energy devices, including floating wind turbines, floating wave energy converters, and floating tidal turbines, a passive station-keeping system, i.e., a mooring system, is needed to limit the offset of the floater under environmental actions of waves, wind, and current, so that the desired normal operations, i.e., power production, can be maintained, without breaking the power cable. Active station-keeping system, such as dynamic positioning system with propellers and thrusters, is not relevant for offshore renewable energy devices because of the high cost and the high electricity used to keep the floating structure in position.

A mooring system typically consists of a group of mooring lines which can be configured as spread mooring or turret (or single point) mooring. Most of the mooring systems for offshore renewable energy devices are based on spread mooring configurations, as shown in Fig. 1 for floating wind turbines. Mooring lines are slender structures of chain links, steel wire, and/or fiber ropes, which are connected to the floater at its fairlead and to the anchor on the sea bed. Mooring systems for offshore oil and gas platforms are designed with redundancy so that failure of one line will not lead to failure of the whole mooring system and will not result into a drift off of the platform. This is because the consequence of such failure might be significant in terms of not only property loss but also fatalities and environmental pollution. However, floating renewable energy devices are usually unmanned, and there are less significant consequences due to mooring failure, which is mainly property loss. Most of the mooring systems used today for floating wind turbines just have three minimum lines with no redundancy.

Depending on the way that a mooring system provides the restoring effect on floater motions, we have catenary mooring, taut mooring, and tension legs. For example, catenary moorings are suitable for spar and semisubmersible floating wind turbines, and tension legs are designed for TLP floating wind turbines, as shown in Fig. 1.

Mooring System of Renewable Energy Devices, Fig. 1

Floating wind turbines and their mooring systems (From left: spar, semisubmersible, and TLP) (Graphic by Joshua Bauer at NREL)



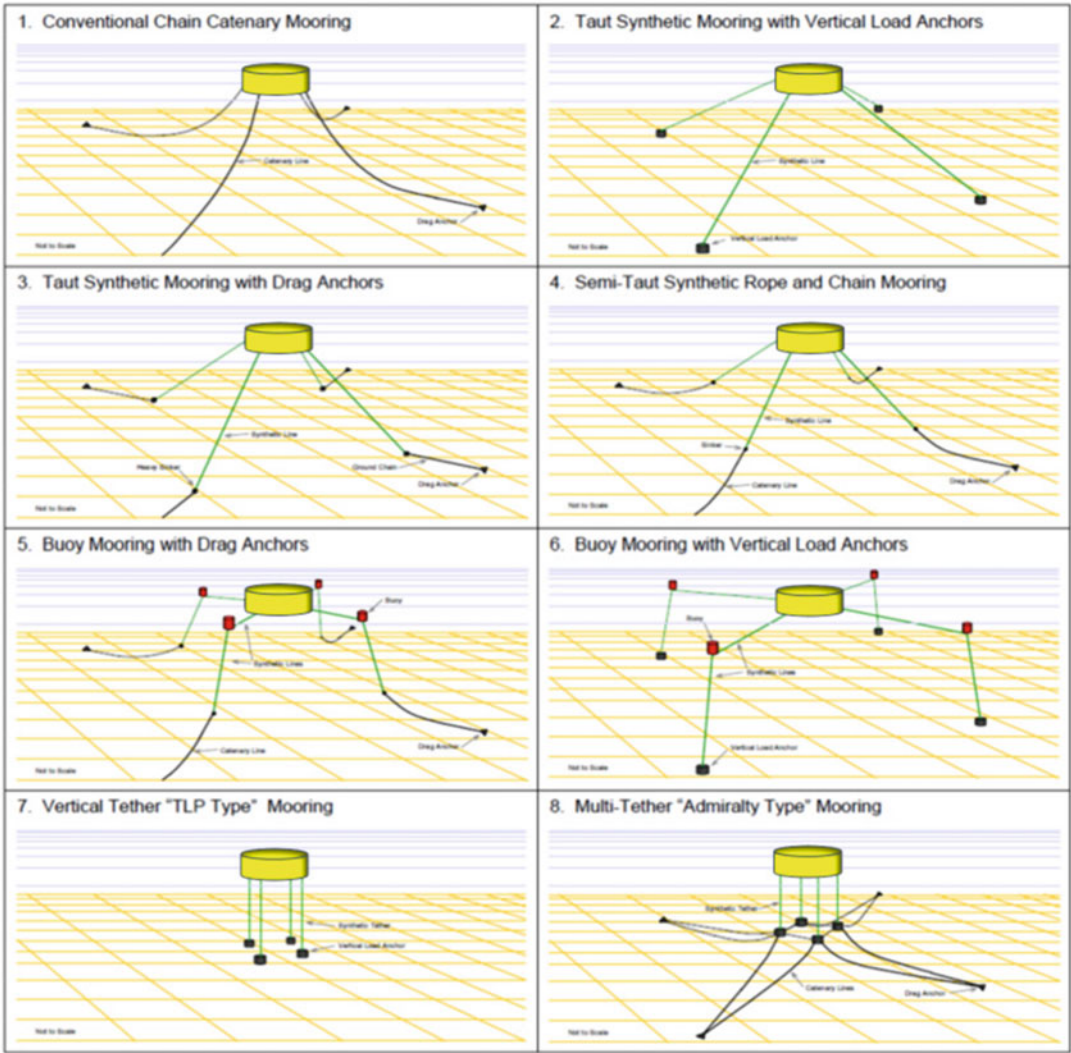
Catenary line is a series of chain links, and it provides the restoring effect via the change of the catenary shape under large gravity force. It has a large proportion of the line on the sea bed, to avoid a sudden increase in mooring stiffness when the whole mooring line is lifted up from the sea bed. When this happens, the stretching of the chain links will take place and give too large mooring stiffness. Taut lines of steel wire or fiber ropes and tension legs provide the restoring effect through the axial extension of the line. When a clump weight or a buoy is used, a catenary effect can be generated from the relative angles between the two segments that are connected by the clump weight or the buoy. An illustration of the different mooring configurations, typically for floating wave energy converters, is shown in Fig. 2.

For floating wind turbines in a farm configuration, it is important to reduce the footprint of each mooring system to limit the range of the offshore wind farm. Normally, catenary mooring systems have a much larger footprint as compared to the taut line or TLP systems. In a TLP platform, the pretension of the tendons is provided by the extra buoyancy force of the floater which is larger than the gravity of the whole system and should be large enough to avoid slack and large transient dynamics

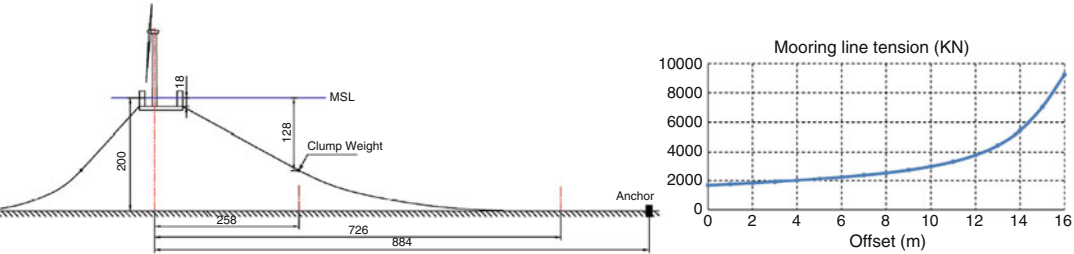
in mooring tendons considering the floater motions under wind, wave, and current loads.

Mooring system has a strong influence on the motions of floating structures. It provides a restoring effect on the horizontal motions of the floater, i.e., surge, sway, and yaw, for which there is no hydrostatic restoring effect. This is particularly important for floating wind turbines, which may have significant mean thrust force on the wind turbine under normal operations and may potentially lead to a large offset of the floater. As shown in Fig. 3, the mooring stiffness is normally characterized by the (horizontal) tension-offset curve, which is typically nonlinear when the offset is large.

The stiffness of mooring lines is designed so that the natural periods for horizontal rigid-body motions of the floater are in the order of 1–2 mins, which are much larger than the main wave periods, to avoid resonance due to the first-order wave loads. However, low-frequency resonant horizontal motions can be excited by either second-order wave loads or slowly varying wind loads, which are one order of magnitude smaller than the first-order wave loads. Both catenary mooring system and deep water taut mooring system have limited influence on the vertical motion modes of the floater, i.e., heave, roll, and



Mooring System of Renewable Energy Devices, Fig. 2 Mooring configurations for floating wave energy converters with catenary line, taut line, TLP as well as with clump weights and buoys (EMEC 2015)



Mooring System of Renewable Energy Devices, Fig. 3 Typical catenary mooring system for a floating wind turbine (left, units in meters) and mooring line characteristic curve (tension-offset relationship) (right) (Xu 2015)

pitch. While tension legs in a TLP are designed with high axial stiffness so that the natural periods of the vertical motion modes are in the order of 1–3 s, which are smaller than the main wave periods. In other words, the wave-frequency motions of the floaters are not constrained in a catenary and taut line system, and the first-order wave loads are compensated by the inertia loads of the floater, while such motions in the vertical modes are constrained by the tendons in a TLP, and the first-order wave loads in these modes are taken directly by the tendons, as the restoring effect.

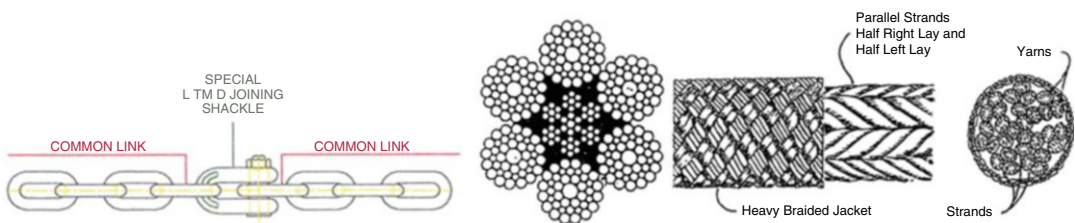
Mooring Line Components

Mooring lines may consist of chain links, steel wire, fiber ropes, and/or a combination of these; see Fig. 4. Chain links are welded and connected steel bars, with or without stud. In deep waters, because the total weight of chain links can be very large and have to be compensated by the additional buoyancy of the floater, they are used in combination with steel wire or fiber ropes with less weight for such conditions. Chain link has a good property against friction and abrasion and are often used at the fairlead and/or at the sea bed. Steel wire might be six-strand or spiral-strand constructions. Synthetic fiber ropes are polyester, aramid, or nylon, which are light materials and suitable for very large water depths. Chain links are normally used in a catenary mooring system. Steel wire can be used in both catenary and taut moorings, while fiber ropes are normally used for taut-line systems. A particular challenge for mooring system in shallow waters is that the mooring stiffness can increase significantly for a small

increase in the horizontal offset of the fairlead, which leads to a significant increase in mooring line tension. Therefore, clump weights or buoys might be placed in between the mooring line segments to increase the catenary effect for such conditions.

Anchors are placed at the lower end of mooring line to provide a fixation for the mooring line and also for the floater. Typically anchors are designed to take only horizontal loads, such as fluke anchor and plate anchor, while some anchors, like pile anchor, suction anchor, and gravity anchor, can take in addition vertical loads. For large oil and gas platform, winches are arranged at the platform level which allows the length change of individual lines and therefore the heading change of the platform. However, because of the cost issues, floating wind turbines typically do not have winches for mooring system, and mooring lines are fixed at the fairlead. Moreover, connectors are used when combining chain links with steel wire or fiber ropes.

Technical specifications in terms of dimension, weight, ultimate strength, and fatigue strength of mooring line components are provided in DNV (2010). In particular, fatigue design of mooring systems requires in the first step to have a global model for coupled mooring analysis considering both wind and wave conditions. Fatigue S-N (stress-number to failure) curves and very often T-N (tension-number to failure) curves for mooring lines are then used in combination with the global response analysis results for fatigue damage calculation. Fatigue property of chain link and steel wire is well-known, while fatigue tests show a large value and a big scatter of the material slope of the S-N curve for synthetic fiber ropes. Therefore,



Mooring System of Renewable Energy Devices, Fig. 4 Chain links (left), six-strand steel wire ropes (middle), and parallel-subrope fiber ropes (right) (DNV 2010)

fatigue design safety factors are typically chosen very large to reflect the big uncertainties in fatigue property of fiber ropes (DNV 2010).

Mooring System for Floating Wind Turbines, Wave Energy Converters, and Tidal Turbines

In this section, examples of mooring systems for floating renewable energy devices will be given.

Figure 1 shows the three types of floating wind turbines (spar, semi-submersible and TLP) and their mooring systems. In fact, there exist MW-scale floating spar and semi-submersible wind turbine demos, for example, Hywind spar wind turbine (Equinor 2018) and WindFloat semisubmersible wind turbine (PrinciplePower 2018). The third one is a TLP wind turbine with vertical tendons as mooring lines.

The mooring system of the Hywind demo is a taut-line mooring system in combination with clump weights. Near the fairlead of the mooring lines, a delta-line configuration of the mooring system is applied in order to increase the mooring stiffness in yaw so that the natural period of yaw motions is around 8 s. A wind turbine rotor in turbulent wind field may experience unbalanced rotor plane loads, which induces a yaw moment. If the yaw stiffness from the mooring system is too low, there will be a large yaw offset, which may strongly influence the power production of the wind turbine.

The mooring system for the WindFloat prototype is a four-line spread catenary mooring system. Two lines are attached to the column which supports the wind turbine, and the other two lines are for the two remaining columns. The fairleads of the three mooring lines are located at the columns, which is far enough from the central position of the wind turbine tower, to create a large yaw stiffness from the mooring system.

The TLP wind turbine as shown in Fig. 1 has three vertical tendons which are significantly pretensioned due to the excessive buoyancy of the floater as compared to its gravity. As a design requirement, the pretension in each tendon should

be large enough so that it will not go below zero under environmental loads due to wind, waves, and current, which will otherwise induce a transient dynamic effect and therefore large loads in the tendon. Single tendon mooring system is also proposed for one of the floating wind turbine concept, SWAY (2012).

Mooring system for floating wave energy converters depends strongly on the concept and size of wave energy converters. Catenary and single point mooring system is used for large-sized wave energy converters, such as the Wave Dragon overtopping device (2018). Such weather-vane mooring system allows the floater to rotate towards the favorable wave directions. Similar moorings are used for large platforms hosting multiple wave energy converters, for example, oscillating water columns. Most of the pointer absorber wave energy converters are small, and their mooring systems, as shown in Fig. 2, might have a strong interaction effect on the wave-frequency motions of the pointer absorbers and therefore the power production.

For commercial purpose, a big number of wave energy converters should be developed in a farm configuration, similar as those for offshore wind turbines. It has the advantage of smoothing power production based on an appropriate configuration and sharing infrastructures to lower the cost. The simplest way to share the infrastructure is to share the anchors for multiple wave energy converters. In such cases, anchors with vertical capacity are needed. Moreover, the wave energy converters can be interconnected so that the total number of anchors and mooring lines can be minimized to reduce the cost. However, the dynamic behavior of the whole system, including multiple wave energy converters and mooring system, can become quite complicated, and large loads in connecting lines might be induced by adjacent wave energy converters with opposite wave-frequency motions (Gao and Moan 2009).

Most of the ocean and tidal current turbines developed today are bottom-fixed. Some floating tidal turbine plants have been proposed, typically with a spread-moored floating platform which supports one or two turbines. Similar to floating

wind turbines, large mean thrust from tidal turbines will need to be compensated by the mooring forces. In addition, since the tides change directions every 6 h, the tidal turbines are normally designed to work in both directions. As a result, the floater may experience large offset in both directions, and the mooring system should be designed to consider this.

Cross-References

- ▶ [Design Rules and Standards](#)
- ▶ [Economic Assessment](#)
- ▶ [Offshore Wind Turbines](#)
- ▶ [Tidal and Ocean Current Turbines](#)
- ▶ [Wave Energy Converters](#)

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Morison-Force Model

- ▶ [Aquaculture Structures: Numerical Methods](#)

MOSES TLP

- ▶ [Tension-Leg platform](#)

Motion Control System

- ▶ [AUV/ROV/HOV Control Systems](#)

Motion Reference Unit (MRU)

- ▶ [Optical Compass](#)
- ▶ [Ultra-short Baseline Underwater Acoustic Location Technology](#)

MSS – Mineral Storage System

- ▶ [Surface Mineral Storage and Dump](#)

MSTS – Mineral Storage and Transport System

- ▶ [Surface Mineral Storage and Dump](#)

MTS – Mineral Transport System

- ▶ [Surface Mineral Storage and Dump](#)

Multidisciplinary Design

- ▶ [Detailed Design](#)

Multidisciplinary Design Optimization (MDO)

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Synonyms

Approximate technology; MDO method; MDO-computing framework; Optimization algorithm; System decomposition; System modeling

Definition

Multidisciplinary design optimization (MDO) is a design methodology for complex engineering systems, which aims to explore the mechanism of synergistic action among subsystems and design complex engineering systems. MDO focuses on the use of optimization algorithms for the design of complex engineering systems which involve a number of subsystems or disciplines. In MDO, the engineering system is divided into different subsystems or disciplines, then the mathematical model of each subsystem is established and the appropriate MDO method is applied to reorganize each subsystem model. Finally, the appropriate optimization algorithm is selected for optimization so as to achieve the optimal design of complex engineering system.

Scientific Fundamentals

Mathematical Model and Basic Concept of MDO Problems

The mathematical model of MDO problems is more complex than that of traditional optimization problems, because the coupling relationship among multiple disciplines or subsystems should be considered in MDO problems. Without loss of generality, for MDO problems that involve a number of disciplines, the mathematical model can be described as follows:

$$\text{Min} : f(x_s, x_i, z)$$

$$\text{S.t.} : g_i \left(x_s, x_i, z_i \left(x_s, x_i, y_{ji} \right) \right) \leq 0 \quad (i = 1, 2, \dots, N; i \neq j)$$

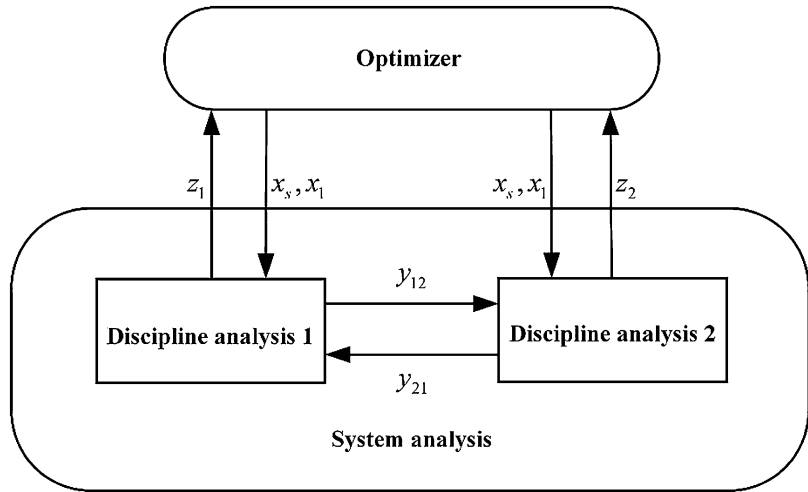
$$h_i \left(x_s, x_i, z_i \left(x_s, x_i, y_{ji} \right) \right) = 0 \quad (i = 1, 2, \dots, N; i \neq j)$$

where f represents optimization objective, g_i and h_i represent inequality constraints and equality constraints of discipline i , respectively. x_s refers to shared design variables, x_i and z_i refer to local design variables and state variables of discipline i , respectively, and y_{ji} is the coupled state variable that discipline j outputs to discipline i . Compared to traditional optimization problems, MDO problems involve more basic concepts. Combined with the MDO problem that involves two disciplines in Fig. 1, some basic concepts involved in MDO problems are introduced as follows:

1. Design variables: Design variables are a group of relatively independent parameters controlled by the designer during the design process. Design variables can be used to represent the basic characteristics of the engineering system. According to the scope of design variables, it can be divided into shared design variables x_s and local design variables x_1, x_2 (see Fig. 1).
2. State variables: State variables are a set of parameters used to represent the response of the engineering system, which can reflect the performance or characteristics of the engineering system. State variables are usually obtained from analysis or calculation models. State variables include system state variables such as the optimization objective, discipline state variables z_1, z_2 , and coupled state variables. The discipline state variables employed as the input to the analysis of another discipline are coupled state variables (i.e., y_{12}, y_{21} shown in Fig. 1).
3. Disciplines: Disciplines, also known as subsystems, represent the basic modules that maintain their relative independence in engineering systems. Different ways of decomposition lead to different disciplinary divisions.
4. Discipline analysis: Taking the design variables of discipline i and the coupled state variables of other disciplines as input, the process

Multidisciplinary Design Optimization (MDO),

Fig. 1 The MDO problem that involves two disciplines



of solving the state equation of discipline i to obtain the output state variable of discipline i is called discipline analysis.

5. System analysis: System analysis is also called multidisciplinary analysis. The system performs repeated iterative calculations with given input variables to coordinate the coupled state variables obtained from the analysis of various disciplines; this process is called system analysis.

Main Research Contents of MDO

Due to the coupling relationship between disciplines, there will be two problems in the multidisciplinary design optimization of complex engineering systems, including organizational complexity and computational cost. A series of technologies and methods on how to solve the two problems of MDO constitutes main research contents of MDO (Sobieszczanski-Sobieski and Haftka 1997), including system decomposition, system modeling, approximate technology, optimization algorithm, MDO method, and MDO computing framework.

System Decomposition

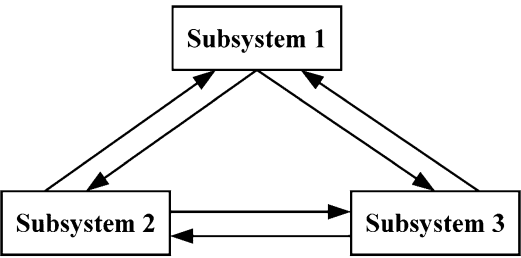
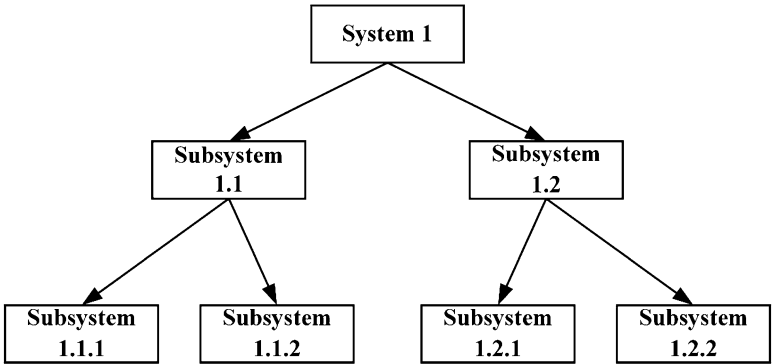
The design of complex engineering system involves high-dimensional nonlinear design variable space, and coupling relationship exists between disciplines; it is very difficult to design and optimize the system directly. Therefore, it

is necessary to use system decomposition technology to change the organizational structure of MDO problems. System decomposition refers to the decomposition of a complex system into several relatively independent subsystems (or disciplines), thus reducing the organizational complexity of the system and shortening the system design cycle. According to the different ways of data transfer within the system after decomposition, system decomposition methods can be divided into two categories:

1. Hierarchical decomposition: For a hierarchical decomposition system, as shown in Fig. 2, data transmission occurs between subsystems of the same level as each other, while there is no information exchange between subsystems of the same level, so parallel analysis and calculation can be carried out. This decomposition method is suitable for multidisciplinary design optimization problems with weak coupling, but difficult to deal with problems with strong coupling.
2. Nonhierarchical decomposition: As shown in Fig. 3, there is no hierarchical relationship in the nonhierarchical decomposition, and data is directly transferred between each subsystem. The decomposition method is suitable for multidisciplinary design optimization problems with strong coupling, which is in line with the practice of complex engineering system design.

Multidisciplinary Design Optimization (MDO),

Fig. 2 Hierarchical decomposition



Multidisciplinary Design Optimization (MDO),
Fig. 3 Nonhierarchical decomposition

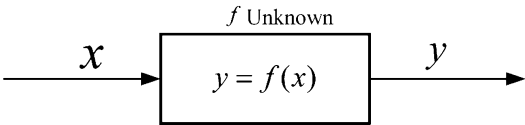
After the decomposition of the system, each subsystem can be analyzed and optimized independently, but coordination among subsystems is needed. There are various coordination methods, and consistency constraint (Haftka 1985) is often used to replace the connection between subsystems.

System Modeling

System modeling is realized by using mathematical language to establish the optimization model of the MDO problem and the analysis model of various disciplines. In addition, to clarify the way of data transfer between subsystems, the mathematical model of coupling relationship between subsystems is also needed. System modeling is the basis of multidisciplinary design optimization of complex engineering systems.

Approximate Technology

The approximation technique is the way to build approximate relationships between data based on existing data sets. As shown in Fig. 4, for a given



Multidisciplinary Design Optimization (MDO),
Fig. 4 Approximate technology

input x , there will be a corresponding output y , but the functional relationship between x and y is unknown. Given a large number of inputs ($x_1, x_2, \dots, x_n$), the corresponding output ($y_1, y_2, \dots, y_n$) can be obtained through analysis and calculation. Then, assuming that x and y meet a specific functional relationship (i.e., approximation model), the function relationship can be determined through the existing data set of input and output; this method is called approximation technique.

By replacing the complex discipline analysis process with simple and efficient approximation models, the computational work can be greatly reduced while ensuring a certain accuracy, and the key of approximation technology is to build an appropriate approximation model to replace the corresponding discipline analysis model. Common approximate models include artificial neural network, response surface model, etc. (Sobieszcanski-Sobieski and Haftka 1997).

For the MDO problem of complex engineering system, the calculation amount of a complete multidisciplinary analysis is huge. If the process is coupled into the optimization model for optimization, the calculation amount increases dramatically. In addition, for some subsystems that use

high-precision numerical software as the discipline analysis model, it is difficult to imagine the amount of computing tasks since the high-precision discipline analysis model of the subsystem will be repeatedly called during the whole multidisciplinary design optimization process. So, approximation technique is an essential research content in multidisciplinary design optimization.

Optimization Algorithm

After establishing the optimization model of MDO problem, it is necessary to search the design variable space to get the optimal design scheme. In this process, the optimization algorithm is needed to find the optimal solution.

MDO problem is an optimization problem in essence, and the appropriate optimization algorithm has an important impact on the final solution of the multidisciplinary design optimization problem, so the optimization algorithm is also an important research content of MDO.

Optimization algorithms can be divided into two categories: (1) traditional optimization algorithms (Nocedal and Wright 2006), such as conjugate gradient method, trust-region method, newton method, etc.; and (2) intelligent optimization algorithm (Simon 2013), such as particle swarm optimization algorithm, genetic algorithm, ant colony algorithm, etc.

MDO Method

Multidisciplinary design optimization method is an optimization solution strategy for complex engineering system design. MDO methods reorganize the system through system decomposition and establish the mathematical model of MDO problem by using system modeling technology. The MDO method starts from the engineering system design problem itself and involves several major research contents of MDO, such as system decomposition, system modeling, approximation technology, optimization algorithm, etc. MDO methods can be divided into single-level method and multilevel method. Single-level method only has an optimizer for optimization at the system level, and there is only analysis within each

subsystem (or discipline) without optimization. The multilevel method has an optimizer at the system layer and subsystem layer for optimization.

MDO Computing Framework

The MDO computing framework refers to the computer hardware and software environment that implements the MDO method. The MDO computing framework can integrate and execute the analysis and design of each subsystem, realize the data transfer between systems, and complete the multidisciplinary design optimization of complex engineering systems.

As multidisciplinary design optimization is widely used in the engineering field, the market demand for high-performance MDO computing frameworks increases sharply. A variety of commercial MDO computing frameworks are developed (Salas and Townsend 1998), such as iSIGHT (Engineous Software), ModelCenter (Phoenix), Optimus (LMS Technologies), etc. In recent years, some open source high-performance multidisciplinary design optimization platforms, such as pyMDO (Martins et al. 2009) and OpenMDAO (Gray et al. 2019), have been developed for MDO researchers to learn.

Historical Development

Multidisciplinary design optimization originates from the aircraft design direction in the field of aerospace research (Sobieszczanski-Sobieski and Haftka 1997). In the 1970s, aircraft design methods mostly adopted the traditional serial design method; this method has many drawbacks, such as ignoring the coupling relationship between disciplines, low efficiency, long design cycle, and high cost.

In order to overcome the drawbacks of traditional design methods, multidisciplinary design optimization was proposed as a new design methodology in the 1980s. Sobieszczanski-sobieski (1982) proposed the concept of multidisciplinary design optimization for the first time in a research paper on structural optimization. Since it was

proposed, MDO has received great attention from academia and industry. In 1986, AIAA, NASA, OAI, and USAF in the United States jointly organized the first symposium on “Multidisciplinary Analysis and Optimization.” Since then, the conference has been held every 2 years and has become a leading international academic conference in the field of MDO.

In order to promote the research and development of MDO, Sobieszczanski-Sobieski (1995) presented a review about MDO; this paper attempts to define the MDO as a new field of research endeavor and as an aid in the design of engineering systems. The Multidisciplinary Design Optimization Branch (MDOB), established by NASA’s Langley Research Center, proposed a test suite in 1996 to measure the performance of MDO methods (Padula et al. 1996). With the promotion of these scientific research institutions, the theoretical research of MDO entered the stage of rapid development in the 1990s, among which the multidisciplinary design optimization method is the core content and frontier of the theoretical research of MDO. In this stage, many classical MDO methods are put forward one after another, including multidisciplinary feasible method (MDF) (Cramer et al. 1994), all-at-once method (AAO) (Cramer et al. 1994), individual disciplinary feasible method (IDF) (Cramer et al. 1994), concurrent subspace optimization method (CSSO) (Sobieszczanski-Sobieski 1988), collaborative optimization method (CO) (Kroo et al. 1994), and bilevel-integrated system synthesis method (BLISS) (Sobieszczanski-Sobieski et al. 2000), in which MDF, AAO, and IDF belong to single-level MDO methods, and CSSO, CO, and BLISS belong to multilevel MDO methods.

In the twenty-first century, the theoretical research of MDO focuses on the improvement of the classical MDO method. Sobieszczanski-Sobieski et al. (2003) reconstructed the system optimization problem on the basis of BLISS and proposed the BLISS-2000 method, in which the information exchange between the system and the subsystem is realized by the approximate model of the subsystem optimization, and the objective function of the subsystem optimization is

represented by the weighted sum of the state variables of the subsystem. Based on the MDF method, Chittick and Martins (2009) proposed a new distributed MDF method, which is called Asymmetric Subspace Optimization (ASO). Aiming at the problem that the coupling variables are difficult to be resolved, Li et al. (2016) proposed a standard for determining the coupling variables, combined it with CO method, and proposed an improved collaborative optimization (ICO) method.

Key Applications

After nearly 40 years of rapid development, the multidisciplinary design optimization theory is becoming mature and gradually moves from the theoretical research stage to the engineering practical application stage. In addition to the successful application in the aerospace field, such as aircraft design optimization (Braun et al. 1996; Jun et al. 2006; Park et al. 2009; Setayandeh and Babaei 2020), multidisciplinary design optimization technology is also widely applied in the field of ship and ocean engineering. Wang et al. (2007) applied the concurrent subspace design method (CSD) to the shape design of a certain AUV. Luo and Lyu (2015) applied the collaborative optimization method to the hydrodynamic performance optimization of underwater unmanned vehicle (UUV). Liu et al. (2017) carried out the research on the multiple objective multidisciplinary design optimization of heavier-than-water underwater vehicle using CFD and approximation model. Bidoki et al. (2018) proposed the PSO-MDF method, by combining the particle swarm optimization algorithm and MDF method, and applied it to the multidisciplinary design optimization of an autonomous underwater vehicle (AUV). Chen et al. (2018) divided the design of AUV into three subdisciplines, including hydrodynamic discipline, control discipline, and energy and weight distribution discipline, and then applied the MDF method to the overall conceptual design optimization of AUV. Bidoki et al. (2019) presented an improved multidisciplinary design optimization methodology for conceptual design

of an autonomous underwater vehicle in both engineering and tactic aspects under uncertainty.

Cross-References

- [Concept Design](#)
- [Design of Submersibles](#)
- [Preliminary Design](#)

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Multihop Network

- ▶ Underwater Acoustic Sensor Network

Multi-source Information Fusion Navigation

- ▶ Integrated Navigation

Multiple Input Multiple Output (MIMO)

- ▶ Underwater Acoustic Communication

Multi-user Long Baseline (MULBL)

- ▶ Long Baseline Underwater Acoustic Location Technology

Multiple Line Moorings

- ▶ Spread Mooring System

Mussel Farm

- ▶ Traditional Aquaculture Structures

N

National Maritime Research Institute (NMRI)

- [Maneuverability of Polar Vessel](#)

National Maritime Research Institute of Japan (NMRI)

- [Ice Tank Test](#)

National Research Council Canada-Ocean

- [Ice Tank Test](#)

Navigation Buoy

Wanan Sheng
SW MARE Marine Technology and Consultation,
Cork, Ireland

Definition

A Navigation Buoy is a special floating buoy (or maybe fixed in occasional cases) that are set on waters for enhancing safety of sea activities

by providing lights and/or marks and other relevant information. For instance, it is of vital importance for marine traffic to obtain correct LOPs. It is also called Aids to Navigation buoy (AtoN buoy). Figure 1 shows a typical navigation buoy.

Scientific Fundamentals

The International Association of Marine Aids to Navigation and Lighthouse Authorities (IALA, previously known as International Association of Lighthouse Authorities (<http://www.iala-aism.org/>)) is an Intergovernmental organization founded in 1957 to collect and provide nautical expertise and advice. Since 1973, the principal work has been implemented as the IALA Maritime Buoyage System. This rationalized system was introduced as a result of two accidents in the Dover Straits in 1971 when the *Brandenburg* hit the wreck of the *Texaco Caribbean* off Folkestone; and shortly later the *Niki* struck the *Texaco Caribbean*. The combined loss of lives in these two accidents was 51 persons. As a result of the accidents, a new system was essentially established to replace some 30 dissimilar buoyage systems in use throughout the world with two internationally standardized systems.

Although the international agreement of 1982 implementing a harmonized buoyage system is a major achievement for the IALA Organization, through its committees carried out a lot of works in other directions resulting in innovating



Navigation Buoy, Fig. 1 Typical navigation buoy with a light and a chart symbol on top. (Adopted from <https://www.ybw.com/features/navigation-essential-buoys-marks-8611>)

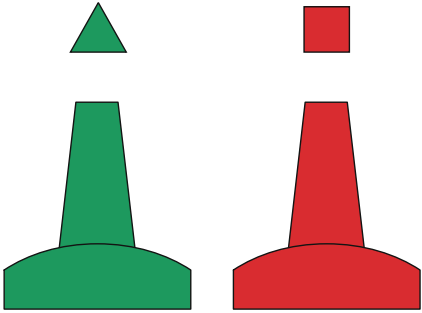
techniques being adopted all over the work, such as the AIS (Automatic Identification System), DGNSS (Differential Global Navigation System), and many others.

Key Applications

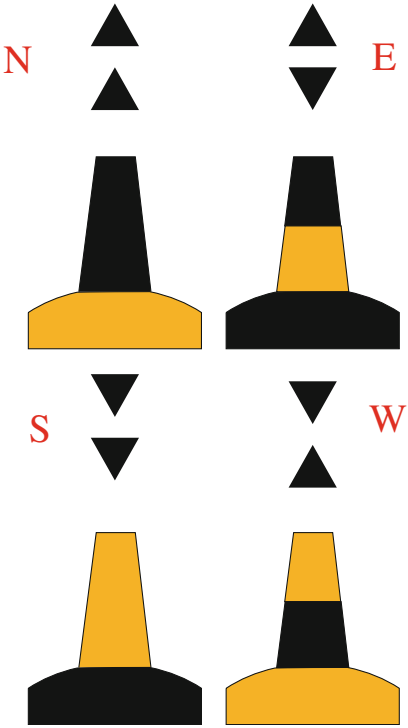
Regardless of which system you use, there are six types of navigational marks: lateral; cardinal; safe water; isolated danger; emergency wreck, and special. Now IALA is known for the IALA Maritime Buoyage Systems or sea mark systems that are used in the pilotage of vessels at sea (following information adopted from <https://www.irishlights.ie/>):

1. Lateral Marks

Lateral Marks are generally used for well-defined channels (Fig. 2). They indicate the port and starboard sides of the route to be followed. They are positioned in accordance with a Conventional Direction of Buoyage.



Navigation Buoy, Fig. 2 Lateral marks

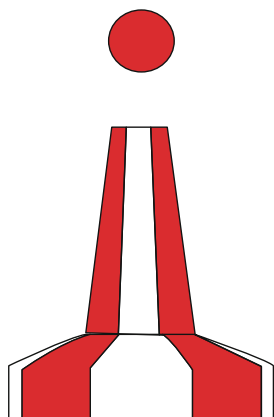


Navigation Buoy, Fig. 3 Cardinal marks

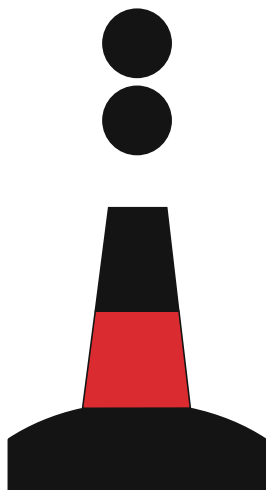
Lateral Marks in region A use red to denote port and green to denote starboard **when returning from sea**. In region B, the reverse applies.

2. Cardinal Marks

A Cardinal mark is named after the quadrant in which it is placed. The name of a Cardinal mark indicates that it should be passed to the named side of the mark. The mariner will be safe if they pass north of a North mark (“N”),



Navigation Buoy, Fig. 4 Safe water mark



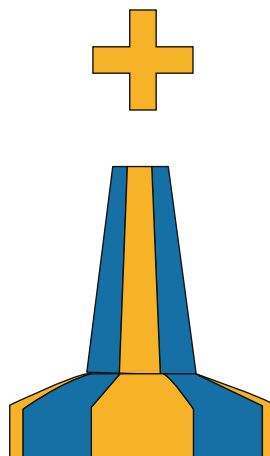
Navigation Buoy, Fig. 5 Isolated danger marks

south of a South mark (“S”), east of an East mark (“E”) and west of a West mark (“W”) in Fig. 3.

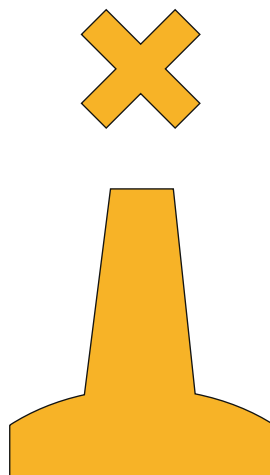
A Cardinal mark may be used, for example, to:

- Indicate that the deepest water in that area is on the named side of the mark.
- Indicate the safe side on which to pass a danger.
- Draw attention to a feature in a channel such as a bend, a junction, a bifurcation, or the end of a shoal.

Cardinal Marks are also used for permanent wreck marking whereby North, East, South, and West Cardinal buoys are placed around the wreck. In the case of a new wreck, any



Navigation Buoy, Fig. 6 Emergency wreck marking buoy



Navigation Buoy, Fig. 7 Special marks

one of the Cardinal buoys may be duplicated and fixed with a Radar Beacon (racon).

3. Safe Water Marks

Safe water marks serve to indicate that there is navigable water around the mark (Fig. 4). These include centre line marks and mid-channel marks. They may also be used to indicate channel entrance, port or estuary approaches, or landfall.

4. Isolated Danger Marks

Isolated danger marks are used to mark small, isolated dangers with navigable water

Navigation Buoy,
Fig. 8 Symbols and abbreviations for light characteristics

Description	Characteristic	Chart Abbreviation
Alternating		Alt. R.W.G.
Fixed		F.
Flashing		Fl.
Group flashing		Gp Fl.(2)
Occulting		Occ.
Group occulting		Gp Occ(3)
Quick flashing		Qk.Fl.
Very quick flashing		V.Qk.Fl.
Isophase		Iso.
Morse		Mo.(letter)

around the buoy. Typically used to mark hazards such as an underwater shoal or rock. They are red and black in color (Fig. 5).

5. Emergency Wreck Marking Buoy

These buoys mark new wrecks clearly and unambiguously. Used as a temporary response, they are colored in an equal number of blue and yellow vertical stripes and fitted with an alternating blue and yellow flashing light (Fig. 6).

6. Special Marks

Special marks are used to indicate a special area or feature whose nature may be apparent from reference to a chart or other nautical publication. Special marks are yellow. They may carry a yellow “X” top-mark and any light used is also yellow (Fig. 7). Uses of special marks include:

- Ocean Data Acquisition Systems (ODAS) marks
- Traffic separation marks where use of conventional channel marking may cause confusion
- Spoil ground marks
- Military exercise zone marks
- Cable or pipeline marks
- Recreation zone marks
- Boundaries of anchorage areas
- Structures such as offshore renewable energy installations
- Aquaculture

In addition, each type of mark has a distinctive color, shape, and possibly a characteristic light (see Fig. 8).

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Navigation of Polar Vessel

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Introduction

In terms of geodynamics, meteorology, hydrology, and geography, the polar regions are too unique to lead to the normality of the conventional navigation methods commonly used in the low-medium latitude. Therefore, it is impossible to reach the predetermined destination safely, accurately, and effectively for the carries and the individuals. With the further exploration of polar natural resources, the researches corresponding to the polar navigating methods have already been becoming a critically significant part of safe voyage in polar regions. Firstly, the article defines the polar region and polar navigation. Then, the challenges easily encountered by conventional navigation methods are

analyzed. Finally, the application status of existing methods is introduced in detail.

Introduction of Polar Navigation

Individuals and carriers, such as vessels, aircrafts, spacecrafts, terrestrial vehicles, etc., utilize the electric, magnetic, acoustic, optical, mechanical, and other methods in the polar regions. And, the target is to locate the moving body by measuring the corresponding kinematic parameters and then to guide it from one point to the destination along the predetermined route safely, accurately, and economically, which is called polar navigation.

Baidu Baike proposes the definition of the polar navigation: navigation in the Antarctic and the Arctic. However, the definition of polar regions corresponding to navigation differs from it defined by the geographic circle of the Antarctic and the Arctic. Hitherto, these two definitions have not agreed with each other yet. In Advisory Circular (issued on June 13, 2018), Federal Aviation Administration (FAA) defines the polar region for aviation navigation. 78°N and more is the Arctic region; 60°S and more is the Antarctic region. This is the most authoritative definition of polar regions.

Nowadays, the ice sheet in the Arctic and the Antarctic is melting rapidly along with global warming. With the background of economic growth and regional integration, polar regions weight heavily on aspects of strategy, economy, science, environmental protection, marine channel, natural resource, etc. With the geospatial advantages in military, resource exploration, and marine transportation, the polar area significantly values in sustainable economic development and national security. Therefore, the polar regions place the pivotal position in marine exploration and probe. However, due to the special geographic location and natural environment in the polar regions, the navigation methods, such as radio, satellite, geomagnetic, inertial, and celestial navigation, cannot work properly, which are commonly used in the low-medium latitude. For example, polar regions are near the South Pole

and the North Pole of the earth. Its unique location causes the convergence of geographic meridians at the geographic poles. Because of the acute convergence of meridian, it is significantly hard to define the heading, relative to the meridian. Moreover, all directions are even toward the south, overhead the North Pole. That, mentioned above, directly restricts the application of the traditional mechanism of the inertial navigation system in the polar regions. The geomagnetic poles are also located in the Arctic and the Antarctic. Its characteristics are the weak horizontal component of the geomagnetic field and the large change of magnetic variation over a large part of the polar regions. Then, it leads to the incapability of the magnetic compass to show reliable directions. Another factor is the abominable and capricious polar natural environment, such as geomagnetic storm, polar day and night, etc., which totally differs from the low-medium latitude. The working characteristic of navigation systems will change and even fail to perform properly. For instance, in polar regions, the ionospheric scintillation frequently occurs. The magnetic storm and solar flare periodically happen, directly triggering the cutoff effect. It is prone to affect the positioning capability of satellite navigation. Radio navigation requires an arrangement of navigation stations on the ground. However, because of polar landforms and climate conditions, its arrangement is restricted. Thereby, applications of radio navigation are limited as well. Besides, wind and low visibility of polar regions, as the major meteorological factors, limit the implementation of landmark navigation technique, visual navigation technique, and celestial navigation technique (Maclure 1949; Qi-Ju et al. 2014).

Navigation technology has always been a critical warranty for the safe and economic navigation of civil vehicles, while the development of polar navigation technology strongly supports the feasibility of polar navigation. In August 21, 1914, Y.I.Nagurskiy, the Russian Royal Navy Captain, flew to the high latitude for the first time. In 1945, Aries I completed the mission of aviation toward the North Pole. So far, western developed countries, especially in Europe and

America, and Russia already have a much more perfect strategy to aviate in the polar regions and even through at the poles. In 2001, it indicates that the polar navigation technology has already matured in Europe and America, when Boeing opened a polar commercial airline connecting North America and Asia. The latest models of large active commercial aircrafts, belonging to Boeing and Airbus, are able to achieve global flight. IRS (inertial reference system) is equipped as the main navigation system when flying over the polar regions. In addition, attribute to the Russian research about the polar navigation, its polar navigation technology grows more and more maturely. Due to the huge potential of polar routes, 2001 China Eastern Airlines 2350 Airbus A340 aircraft, Boeing 777-21BER B-2055 aircraft of China Southern Airlines was trialed for the polar flight. At present, Chinese scientific research ship "Xue Long" has completed eight times scientific research missions in the Arctic. Obviously, during the polar region navigating, polar navigation technology plays a critically important role. It can not only determine the location of the vehicle itself and its destination but also reduce the frequency of the accident occurrence.

Polar Navigation Methods

Along with the improvements and developments of polar navigation until now, the common navigation methods include:

Inertial Navigation

The earliest application of the polar inertial navigation could date back to 1958. American "Nautilus" nuclear submarine, equipped with N6A inertial navigation system, sailed underwater for 21 days and succeed to cross the North Pole. Its voyage is 8146 nautical miles, while its positioning error is only 20 nautical miles during the whole sailing (Lyon 1984). Compared with traditional navigation methods, like electro-magnetic navigation technique, and traditional navigation methods, the polar inertial navigation technology has the advantage of stable

performances in the harsh polar environment. And it is proved to be an optimal strategy to sail toward and in the polar regions. The inertial navigation systems can be arranged by different mechanisms. The mechanism propagated in the east-north-up frame is easy to understand and clear to physically expressed. However, its shortcomings in the polar regions are to hardly apply torques on azimuth gyro, to overflow in the system computer calculation, and to amplify errors of positioning parameters. To solve its problems, the inertial navigation mechanism proposed in wander angle frame can avoid the overflow caused by convergence of polar meridians. This mechanism was adopted by the early LN-51 aviation navigation system (DelCore and Nascro 1986). However, as for the wander angle introduced purposely, its error is also amplified with the increase of the latitude. Meanwhile, the position extracted from the cosine direction matrix (CDM) is almost singular, when it is very close to or just at the pole. Thus, the system performs badly. For solving problems of the wander angle frame in the polar regions, the mechanism of the earth-centered earth-fixed coordinate system (ECEF), grid coordinate system (Ignagni 1972), and transverse coordinate system (Dyer 1971) can be adopted reasonably. They are all with abilities of global flight, further to achieve polar navigating. In 1958, "Skate" nuclear submarine and "Sargo" just utilized the grid inertial navigation system and completed polar navigating missions. The MAPS inertial navigation device of Honeywell is used by the National Defense Department of Canada for autonomous navigation in the high latitude (84°N). The Russian Vega-M inertial compass system is able to guarantee the medium-precision output of attitude and heading between 85°S and 85°N. The icebreaker, equipped with the Vega-M system, arrived at the area of 89°N in 2003. It is proved that, when the operation exceeds the range of the system, the navigation errors will surge obviously (John et al. 2003).

Satellite Navigation

At present, four major global satellite navigation systems are GPS of the United States, GLONASS

of Russia, Galileo of Europe, and BDS of China. In the polar regions, the arrangement of the satellite navigation system constellations directly affects the positioning accuracy. BDS is mainly a high-orbit satellite, mainly above the Asia-Pacific. The average number of satellites over the medium-high latitude is about two, less than GPS, Galileo, and GLONASS. Its geometric arrangement is poor, and then positioning precision descends sharply. The number of visible satellites of GPS changes little with the latitude and longitude, all about 8–10 satellites. So is in the polar regions. The number of visible satellites of Galileo in the polar regions is always more than four. Moreover, with the completion of the full constellation in 2018, the number of its visible satellites in the high latitude will be more than that of GPS. Because Russia is located at the high-latitude area, the GLONASS is designed to place more satellites in the high-latitude region. So the number of visible satellites in the polar regions is the largest. Besides, its satellite inclination is the highest, which results in a better performance of constellation geometry in high latitude. Although more visible satellites are distributed in polar regions with the comparison of the low-medium latitude, most satellite's elevation angle is relatively low. In addition to constellation distribution, the tropospheric delay is another key factor for satellite positioning. The related research confirms that the tropospheric delay influences more on the observation of satellites of a low elevation angle. The polar ionosphere scintillation occurs frequently, while the daily fluctuation of the total electron content (TEC) is more fierce than low-medium latitude. Besides, due to the uniqueness of polar regions, it is hard to use satellite-based or ground-based augmentation systems to modify the positioning accuracy. For example, for satellite-based systems like EGNOS and WWAS, the polar regions cannot be covered by the geostationary earth orbit (GEO) satellites, only seen in part of regions and time. For ground-based augmentation systems, there have been a few observation stations in polar regions. Meanwhile, the harsh environment condition makes the establishment of long-time stations difficult, like lack of support of

sustainable energy and chain of real-time communication, etc. (Yuanxi and Junyi 2016). Therefore, the positioning performance of the satellite navigation system in the polar area is at subpar levels.

Celestial Navigation

Same as the polar inertial navigation technology, the polar celestial navigation technology is also autonomous. The polar celestial navigation technique uses the star sensor to track the relative celestial bodies' position in space, in which it is an accurate and passive system. Then, the navigation computer can calculate the navigation information through observed parameters. The advantage is that it is not influenced by the magnetic field deception jamming and has a good performance of concealment. Thus, since the development of the star sensor in the 1990s, it has been widely used in the military and civil (van Bezooijen 1994) and has increasingly played a key role in polar navigation as well. Even under some specific conditions, it is the only method that can be relied on. However, the special sailing conditions in the polar regions restrict the operation of polar celestial navigation technology to a certain extent. Especially the polar navigators might be caught in the long twilight which may last for several days, when neither sun nor stars are available. Meanwhile, because the update rate of the celestial navigation system is low, it lacks the capability of the provision of real-time navigation information. Thus, it is usually an adding method to correct accumulative errors of the inertial navigation system for autonomous, high-precision, and long-time navigation. It has been confirmed that the careful dead reckoning and hourly celestial bodies' observations are sufficient for reasonably accurate navigation (Molett 1954). Taking the American air force as an example, it has been contributing to this kind of research since the 1950s, such as the NAS-26 inertial/celestial integrated navigation system in the 1980s and the SAIN strapdown inertial/celestial navigation system in the 1990s. They are all the typical airborne inertial/celestial integrated navigation products (Ming 2019).

Underwater Acoustic Navigation

Polar underwater acoustic navigation technology helps increase the transparency of underwater space and enhance the survival ability of underwater vehicles. Research in navigation and localization algorithms, along with advances in acoustic positioning and seafloor relative Doppler velocity log (DVL) measurements, has significantly advanced the state of the art over the past two decades. However, operating in the polar regions presents unique challenges – first and foremost that surface roughness interacts with wind and current due to the polar sea ice, which tends to form underwater noise easily. The noise will extremely limit the performance of the sonar. Moreover, polar seafloor conditions are more unpredictable and severe. Under the unknown underwater condition, it is difficult for acoustic signal beacons to be installed and arranged all over the seafloor. By this very existence, vehicle and ship are hard to access to the free surface and, therefore, require fundamental changes in the methods of operation. At the same time, the navigation methodology employed must be modified as well (McFarland et al. 2015). In order to extend the voyage, Russia, America, and other countries are all active to develop innovative underwater acoustic navigation systems to adapt to the special underwater sailing circumstance in the polar regions. For example, Russia actively explores an innovative underwater acoustic navigation system, composed of the GLONASS navigation system, sonar buoy, and unmanned underwater vehicles. And, the system is located on the continental shelf of the Arctic Ocean of Russia.

Integrated Navigation

In order to further improve the accuracy and reliability of navigation, the vehicle can adopt a combined working mode as the integrated navigation, which means that it takes one of the navigation methods as the dominant with the aid of various navigation systems. Thus, the integrated navigation system can suppress respectively systematic errors contained in each system to achieve accurate positioning and navigating. Considering the mature technology of the integrated navigation of the foreign underwater

vehicles, the integrated navigation system is mainly the strapdown inertial navigation system plus Doppler sonar. Its positioning accuracy can reach 0.01% of its voyage. For example, with the aid of the same Doppler velocity log (DVL), one underwater vessel of the Bluefin Robotics Company of the United States carries the laser inertial navigation system, while another takes optical fiber inertial navigation system for the sea trial. And its results demonstrate that both two systems have relatively high precision (Tal et al. 2017). Because the performance of celestial navigation will be interrupted by the polar environment, it always collaborates with the inertial navigation system. For instance, the LN-120G inertial/celestial integrated navigation system of the United States can globally navigate. It is with capabilities of fully autonomous, high-precision, and long-time navigation. Its positioning precision even reaches 0.5 nautical miles per 18 h (<https://www.northropgrumman.com/Capabilities/LN120GStellarInertialNavigationSystem/Pages/default.aspx>). Until now, China has been also active to develop the integrated navigation system to adapt to polar navigation. For example, “Yong Sheng” owned by China Ocean Shipping (Group) Company (COSCO) first voyaged successfully through the Arctic in 2013. The GPS compass equipped on the ship is the high-precision integrated navigation system. The system is improved by the fusion of dual antenna azimuth solution technology of GPS and fiber optic gyroscope technology.

Cross-References

► [Polar Acoustics](#)

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Navy Oceanographic Meteorological Automatic Device

- [Wave Measurement Buoy](#)

NC (Numerical Control Cutting)

- [Introduction to Shipbuilding \(Shipyard\)](#)

Neighbor Discovery

- [Underwater Acoustic Sensor Network](#)

Net Hydrodynamic Characteristics

- [Net Structures: Hydrodynamics](#)

Net Present Value (NPV)

- [Christmas Tree](#)

Net Structures: Biofouling and Antifouling

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Synonyms

[2,3,5,6-tetrachloro-4-methylsulfonyl \(TCMS\)](#); [2-thiocyanomethylthiobenzo-thiazole \(TCMTB\)](#); [Dissolved oxygen \(DO\)](#); [International Maritime Organization \(IMO\)](#); [Netpen liver disease \(NLD\)](#); [Tributyltin \(TBT\)](#)

Definition

Biofouling, i.e., marine biological fouling, corresponds to the accumulation of microorganisms, plants, and aquatic animals on artificial surfaces immersed in seawater. Biofouling is an ocean phenomenon and an important scientific and technical term to understand what happens to artificial surfaces in seawater while exploiting ocean resources in terms of the following in aquaculture and mariculture: the biofouling community, fouling community, and foulant communities; the foulant, fouler, and fouling organisms; pens, artificial structures enclosed on all sides with the bottom formed by the sea bed, or cages, artificial

structures enclosing all submerged surfaces; the stocking density, the amount of fish in the cage or the total number of fish inside; and the mesh occlusion and mesh plugging abundance of fouling organisms.

Scientific Fundamentals

The structure, composition, and large surface areas of netting materials, especially multifilament meshes, are highly suitable for the colonization and development of biofouling (Ashraf et al. 2017). Biofouling often develops rapidly because the seawater surrounding aquaculture operations is enriched by inorganic and organic wastes (food residues, feces, and excretory material) generated by high-density fish populations (Navarro et al. 2008; Sanderson et al. 2008). The increases in carbon, nitrogen and phosphorus in the seawater surrounding mariculture farms facilitate the growth of annual filamentous algae (Ruokolahti 1988). The rapid growth of fouling in eutrophic seawater leads to the complete blockage of the netting mesh within 2–3 months (Neori et al. 2004). The biofouling of fish-cage netting can lead to significant operational problems in aquaculture. Mesh plugging and the resulting restriction on water exchange adversely influence the health of fishes owing to the reduction in dissolved oxygen (DO) and the accumulation of metabolic ammonia. Fouling is an additional concern because it significantly decreases cage flotation, increases structural fatigue and cage deformation, and may act as a reservoir for pathogens (Amaraa et al. 2018). The impacts of fouling vary dramatically depending on the season and location and are also influenced by the farming methods and practices.

The fouling of mariculture structures differs from that of many other marine industries, in particular, marine transport, in terms of the surface characteristics; specifically, the surface is rough, and consequently, the surface area is large (Bixler and Bhushan 2015). Moreover, the fouling of mariculture structures is not subject to the high water velocities and shear forces associated with ship hulls or the internal surfaces of pipes. Early

studies on the fouling on mariculture netting showed that the netting material and mesh size significantly affected the fouling rate, mesh occlusion, and density and abundance of fouling species (Rothwell and Nash 1977). According to these data and observations of mesh deterioration, materials were rated for their suitability in the construction and maintenance of fish cages. More recently, the effects of the net angle and microfouling development on multifilament meshes (Corner et al. 2007) were investigated. However, due to the importance of the aquaculture industry in global economic development, the number of studies on the impact of marine biofouling in all forms of aquaculture is limited and often disparate and sparse.

Development of Biofouling in Mariculture

In intensive commercial mariculture, multifilament netting bags or cages suspended from a floating frame are usually utilized. In particular, high standards of water quality must be maintained through water exchange, which depends on the speed and direction of water flow, which is affected by the salinity, temperature, and topography of the site (Beveridge 2004). The key nature of water exchange renders the influence of biofouling critical on all forms of aquaculture, especially cage-based aquaculture. The main effect of biofouling on cages is to restrict the water current, the effect of which increases as the biofouling communities develop from a biofilm to a complex three-dimensional climax community dominated by sessile invertebrates.

The development and composition of fouling communities on fish cages has been investigated for many types of mariculture (Cronin et al. 1999; Greene and Grizzle 2007). Multifilament mesh materials are generally regarded ideal fouling surfaces, and the succession of organisms attached to aquaculture networks has been specifically evaluated (Svane et al. 2006). Even after immersion for a short period of time, for instance, less than a month, macroalgae are usually found to be the most serious type of foulants on cages (Hodson



Net Structures: Biofouling and Antifouling,

Fig. 1 Hydroids fouling a testing net in Zhoushan during late summer. Hydroids are common and problematic fouling organisms, which are hard to dislodge and regenerate rapidly. These organisms reduce the flow and oxygen levels within cages, and constant management is required to prevent their impact on the growth and survival of cultured fishes in the cage. (Photograph by Hailong Zhang)

and Burke 1994). In contrast, several other marine organisms, such as bivalves, ascidians, and hydroids (Fig. 1), may be the predominant foulants on cages after immersion for longer periods, although these organisms may also cause serious fouling problems even in a brief period, especially during periods of high larval settlement (Greene and Grizzle 2007).

Variation in Biofouling Between Sites: On a large scale, considerable differences exist in the biofouling between sites, for example, between temperate and tropical waters (> 1000 km). However, spatial variation also occurs on a smaller scale (< 100 km). The differences in sites may represent variations in the environmental conditions, such as differences in the seawater quality, salinity and temperature, as well as differences in the abundance of larval stages (Santhanam et al. 1983). For instance, the fouling communities on polyethylene netting differ between cages immersed in brackish and marine waters. Cages in brackish water (24.5–33.8%) are usually colonized by the algal genera *Enteromorpha* and *Ectocarpus*. However, the cages in marine conditions (36%) are usually colonized by a more diverse community, including bivalves, sea anemones, solitary and colonial ascidians, algae (*Caulerpa* spp., *Codium*

spp., and *Gracilaria* spp.), amphipods, barnacles, and polychaetes (Santhanam et al. 1983).

The Variation in Biofouling Within Sites is mainly driven by the effectiveness and availability of light and water flow, which are related to the depth and orientation of the cage (Cronin et al. 1999). The differences in the photosynthetic biomass between the cage sides are detectable only near the surface where light intensity variation is most pronounced; for instance, the significant fouling changes that occur at a depth of 0.5 m become nonsignificant at a depth of 2.0 m. Overall, the fouling mass decreases significantly with increasing depth.

The direction of immersed surfaces affects the development of fouling organisms, and a significant difference exists between vertical and horizontal substrates. This aspect was demonstrated in a comparison of vertical and horizontal net panels. The vertical panels fouled more rapidly, developed a greater fouling biomass, and exhibited an increased abundance of compound ascidians and tubeworms. However, barnacles and oysters were found to be more abundant on the horizontal frame (Cheah and Chua 1983). The increase in biomass on the vertical panel was believed to reflect a greater interception of horizontally moving planktonic larvae, which increased the larval settlement. However, an increase in collisions with suspended material increases the nutrition of filter-feeding organisms. Compared with the vertical surface, communities on the horizontal surface are more prone to siltation and predation, and vertical or stacked species are favored. However, on vertical surfaces, in which critical competition for space occurs and the predation pressure is less, colonial growth is more effective (Harris and Irons 1982). In addition, the relative abundance of individual bivalves was observed at the bottom of a cage, and more individuals were observed to grow on the outer than on the inner surfaces of a cage (Lee et al. 1985).

Netting and Biofouling

Because the waters around mariculture sites are rich in organic and inorganic wastes produced by

high-density fish populations, biofouling usually occurs extremely rapidly. The fouling of mariculture structures is different from that of structures in many other marine industries, especially marine transportation, because of the rough surface and high surface area. Moreover, such fouling is unaffected by the high flow velocities and shear forces associated with the surface of ship hulls and the inner surface of pipes. Early studies on the surface fouling of mariculture cages showed that cage materials and mesh sizes considerably influence the fouling rate, mesh occlusion, and density and abundance of fouling organisms (Rothwell and Nash 1977). Based on these data and the observed mesh deterioration, materials are evaluated to determine their suitability for the construction and maintenance of fish cages. Recently, the influences of the mesh angle and development of microfouling on multifilament mesh have also been investigated (Corner et al. 2007).

Influence of the Mesh Size of a Cage

In commercial fish farming, a variety of mesh sizes are adopted, ranging from 12–40 mm, 60–90 mm, and 100–150 mm for salmon cages, bluefin tuna cages, and predator fences, respectively. Larger grids usually have thicker gauges; however, a smaller mesh size corresponds to a larger surface area per square meter. Therefore, smaller grids usually incur more fouling organisms and total biomass. In general, the fouling rate, fouling quality, species diversity, and species richness increase as the grid size decreases (Cheah and Chua 1983). Consequently, to maintain acceptable water exchange, small grid nets must be cleaned more frequently than larger nets. High sediment loading does not require an excellent substrate for the settlement and growth of foulants because the accumulation of silt makes the surface of the multifilament mesh rougher. Thus, the grid size and total surface area are believed to interact and influence the development of biofouling.

Influence of the Mesh Structure

The distribution and type of the initial fouling process were reported to be influenced by the micromorphology of the multifilament mesh

(Hodson and Burke 1994). The cylindrical shape of the grid strip resulted in different light intensities on the upper and lower surfaces of the horizontal submerged grid strip. Therefore, a community of phototrophic organisms (such as diatoms) formed on the upper surface of the horizontal strip, and a heterotrophic protozoan community developed on the lower surface.

The large crevices and many filaments on the surface of netting rope are likely to contribute to the attachment of fouling organisms, either because of the entrapment of suspended matter or because several larvae of fouling invertebrates and spores of common fouling organisms preferentially settle in small depressions. For many organisms, a preferred settlement size exists, which may affect the community composition of micro- (Scardino et al. 2006) and macrofoulants (Scardino et al. 2008).

The three-dimensional structure of a net mesh directly affects the development of fouling on netting. The occurrence of biofouling attachment has been noted to be preferential at grid intersections (Tseng and Yuen 1979); for instance, researchers observed the appearance of a large aggregation of mussels at the intersection and the presence of bryophytes, barnacles, and green algae mainly at the intersection. This preferential attachment is presumably due to the larger surface area and turbulence changes in certain sea areas.

Influence of the Mesh Material

A number of suitable materials have been employed for the construction of fish cages. These materials have different degrees of antifouling capabilities. In this respect, some studies have proven the relative capacity of many types of netting: multifilament polymer nets, extruded polymer meshes, metallic hardware cloths, and extruded metal meshes.

After examining ten mesh types, polymer-fiber nets and galvanized meshes were found to be the most and least vulnerable to marine fouling, respectively. After 4 months of immersion, the polymer fiber mesh was completely blocked by the growth of marine organisms (Fig. 2), and the weight of the test panel (0.4 m²) increased from 5.5 kg (clean) to more than 15.5 kg. In contrast, a



Net Structures: Biofouling and Antifouling, Fig. 2 Settlement of marine organisms on the small mesh net. As a monospecific group, hydroids usually breed rapidly, increasing the weight of nets, and can easily block the water flow through nets of small mesh sizes. (Photograph by Hailong Zhang)

reasonable flow of water passed through the galvanized material mesh, and the weight of the panel increased from approximately 7 kg to 9 kg. Furthermore, nylon and polyethylene meshes were found to foul at a significantly greater rate than metal meshes. In addition, the polyethylene mesh and galvanized mesh exhibited the greatest and least fouling, respectively, after 5 months (Rothwell and Nash 1977).

In addition, the fouling community composition differed among various grid types. First, algae propagated on most nets but became most abundant on nylon nets and nets with ineffective antifouling paints. All the panels were rich in serpulid tubeworms; however, these organisms were not widely observed on extruded polymer mesh- and PVC-coated chain links, on which barnacles were abundant after 5 months (Rothwell and Nash 1977).

According to this discussion, the material, mesh structure, and mesh size of cages have important impacts on the biofouling rate, mesh blockage, and fouling biological density and quantity. Fouling communities in cages are usually developed and characterized by a large quantity of biomass. For example, the species composition of a biofouling community immersed in seawater for 4 months is almost the

same as that immersed for 2 months, although the weight of the former will be doubled (Cheah and Chua 1979). Nevertheless, large amounts of silt are often deposited and associated with those biofouling communities. For example, one study reported that 58% of the total fouling biomass with an area of 4.5 kg/m^2 is silt (Lee et al. 1985).

Impact of Biofouling on Mariculture

Factors Limiting Water Exchange

The main concerns pertaining to fishnet cage fouling relate to the mesh blockage and change in the water quality due to the limitation on the water flow. The water exchange inside and outside the cage is influenced by the external velocity (Edwards and Edleston 1976) and the angle between the mesh and flow (Gularte and Huguenin 1984). The differences in transmission measurements may be due to the methods used to quantify the flow, the stocking density of the cages, and the circulating flow generated by the fish. The transmission of a clean net is related to the mesh size and usually varies between 50% and 80%. Due to the biofouling of the mesh and the grouping of the cages, the transmission decreases significantly. After 52 days, 80 days, and 120 days in the ocean, the transmission rate (57.5%) of a clean 13 mm mesh decreased to 23.4%, 18.7%, and 13.1%, respectively, with the corresponding fouling biomass weights being 1.85 kg/m^2 , 2.84 kg/m^2 , and 4.98 kg/m^2 (Wee 1979).

When three 9 mm mesh cages were placed parallel to the flow, the flow decreased in a serial manner, and the transmission rate decreased from 70% for the first cage to 35% and 18% for the second and third cages, respectively (Inoue 1972). As the cages were arranged in a series and the netting became fouled by marine organisms, these two effects dramatically reduced the exchange of water (Aarsnes et al. 1990). Hence, grouped cages are suggested to be oriented perpendicular to the current, with no more than two or three cages in a series being placed parallel to the current (Beveridge 2004).

In the fish farming industry, a common practice is to increase the size of the net pen and place large

nets in the sea throughout the production cycle (Sunde et al. 2003). In this scenario, the effects of biofouling on the water exchange and net deformation are expected to become more severe because larger cages and nets have smaller surface area to volume ratios than smaller nets, resulting in lower water exchange rates (Lader et al. 2008).

Factors Limiting Water Quality

Water exchange is essential to replenish dissolved oxygen and remove excess feed and waste materials to maintain the water quality. Many studies have demonstrated a decrease in the oxygen concentration from the outside to the inside of cages and clarified the relationship between the oxygen reduction and short-term water exchange (Wee 1979). In addition, an increased stocking density increases the oxygen consumption inside cages (Kadowaki et al. 1978). Therefore, combined with low current flow and significant mesh blockage and a high density of fish, dissolved oxygen may rapidly decrease to critical levels (Edwards and Edelston 1976).

Kennedy et al. (1977) reported the mortality of fish due to anoxia in severely fouled cages, in which the dissolved oxygen (DO) concentration was less than 4.0 mg/L. Low water exchange was found to directly cause this low DO concentration in the cages. After installing a clean net, the concentration of DO increased to 8.25 mg/L. For salmon aquaculture, the oxygen concentration is recommended to be greater than 7 mg/L; notably, a concentration less than 5 mg/L will have a negative impact on the growth and respiration of fish, and a concentration less than 2 mg/L will lead to mortality (Boyd 1982).

Many factors contribute to the total supply and consumption of DO in the cage (Silvert 1994; Cronin 1995). In particular, oxygen supply occurs primarily through water exchange but also through the photosynthesis of foulant communities and atmospheric diffusion. Although oxygen is mainly consumed by fish, to some extent, it is also consumed by the biochemical oxygen demand of the immediate environment and fouling communities. The maximum stocking density of fish depends almost completely on the exchange of water and can be calculated

according to oxygen consumption and the oxygen supply rate. Although the oxygen levels in cages are mainly controlled by the water exchange, the production or consumption of oxygen by the fouling community can affect the oxygen concentration (Cronin et al. 1999). Reduced water exchange may also affect the fish health, as the ammonia levels in cages may increase compared to those of the surrounding waters.

Factors Influencing the Fish Disease Risk

Biofouling has been reported to be a risk factor. Fouling communities may pose health risks to cultured fish species because they can act as reservoirs for pathogenic microorganisms that exist within large fouling species or in a wide range of microbial communities on cage netting (Meyers 1984). Marine aquaculture may lead to unusual parasitic infections caused by farming in new geographic areas or in net pen environments (Kent 2000). The occurrence of caged fish diseases is also linked to the consumption of fouling organisms by cultured fish species (Andersen et al. 1993). Fouling communities may directly affect fish by causing physical damage to the cultured species. Moreover, a severely fouled net can also support the existence of the free swimming stage of sea lice (*Lepeophtheirus salmonis*) (Kent et al. 1995).

Nevertheless, biofouling organisms may also reduce the disease risk. The potential for the mussel *Mytilus edulis* to harbor bacterial kidney disease makes the survival of the bacterium *Renibacterium salmoninarum* unlikely. These mussels kill the majority of *R. salmoninarum* during digestion and may reduce the level of pathogenic bacteria in the cage environment (Paclibare et al. 1994).

Cage Deformation and Structural Fatigue

Exposure to currents causes net cages to change their shape through deflection and deformation (Fredheim 2005). The extent of this shape change depends on the flow velocity, the original shape and construction of the cage, the placement of weights, the type of net, and the degree of biofouling (Lader et al. 2008). An increase in mesh occlusion will significantly increase the drag

forces on mesh netting. Milne (1970) measured the current forces on clean and fouled nets under different current velocities and showed that the forces on a fouled net may be 12.5 times greater than those on a clean net. Consequently, unless the cage is heavily weighed, the shape of the cage may be severely distorted by the current flow (Osawa et al. 1985). Aarsnes et al. (1990) calculated the deformation rate of a 12,000 m³ cage with a bottom weight of 400 kg and found that the cage volume was reduced by 45% (to 6600 m³) and 80% (to 2300 m³) under velocities of 0.5 m/s and 1 m/s, respectively. Wee (1979) observed a 50% reduction in the volume of a heavily fouled cage in use. Reducing the cage volume may affect the fish health, as the oxygen consumption and ammonia production are expected to increase per unit volume and crowding is likely to stress the cultured fishes.

Lader et al. (2008) analyzed the influence of incoming currents of varying velocities on the deformations of net cages. They found that a current velocity of 0.35 m/s resulted in a 40% reduction in the cage volume. Highly deformed nets increased the structural stress of the cage. Although increasing the cage weight could reduce the deformation, the structural stress will increase (Anon 1993). Tomi et al. (1979) indicated that the weight added to cage corners caused a two- to six-fold increase in the horizontal forces on the cage. Specifically, with heavy weighting, waves cause the floating frame to move upward, and the weight pulls the net down. The structural loading and fatigue may further increase when predator netting is attached to cages.

The biomass of the fouling community usually directly impacts the static load of net cages, which may increase the weight of cages by a factor of 200 (Beveridge 2004). This additional load must be considered in the design of floating and mooring systems. Failure to do so may incur net failures or breakage, which have been devastating for commercial mariculture (Huguenin 1997). In addition, Swift et al. (2006) measured the increase in the fluid resistance in net cages due to biofouling. Biofouling resistance coefficients generally increase with increasing solids (ratio of projected area to defined area) and biomass, and the

resistance of fouled nets may be more than three times that of clean nets.

Summary of Biofouling Impacts

Biofouling in marine cages can lead to mesh clogging, resulting in decreased productivity and health risks to fishes as well as structural fatigue and cage deformation. Furthermore, biofouling is a critical management issue. If the impact of biofouling is not adequately considered, the costs of operation and maintenance will increase.

Surprisingly, there is little information about the effects of fouling, a problem of such high impact, and this aspect is only now beginning to be discussed in detail. Considering the limited products available to control aquaculture fouling, a quantitative study of the spatiotemporal variation of biofouling communities at the species level, as well as the impacts of animal husbandry and farm management on fouling development, can help the mariculture industry to choose the most cost-effective fouling control method.

The rapid development of cage aquaculture in tropical and subtropical areas, in which biofouling occurs most rapidly, and the shift to offshore cage mariculture with limited coastal space poses new challenges to understanding the impact of biofouling, especially to implementing successful marine fouling control strategies.

Principles of Controlling Biofouling

Physical Approach

In view of the serious impact of biofouling in commercial fish farms employing cage culture, protective measures typically involve a variety of methods to control net fouling, usually including the utilization of farm management techniques, namely, frequently changing and cleaning the netting, and the use of antifouling coatings. In the following, these methods are reviewed along with the upcoming options for the nonbiocidal control of biofouling in marine aquaculture. This aspect is particularly important because of the restrictions on the application of traditional metal-based coatings in the aquaculture industry.

Net Changing

In almost all parts of the world, fouling develops rapidly at a certain stage in use. Frequently replacing and cleaning nets are essential practices to maintain water exchange within cages. The replacement rate of a large mesh cage is relatively low because critical mesh blockage or occlusion occurs only under a considerable amount of biofouling. Although nets are frequently changed in temperate and tropical regions, cages immersed in offshore locations and extremely cold water can be immersed in water for a considerably long period without cleaning. Moreover, the cleaning frequency can be delayed to a certain extent by raising the top several meters of the cage above the water surface (Needham 1988); however, this technique is only applicable to the case in which the fouling is limited to the upper area of the cage.

Net changing may incur considerable costs in mariculture, and it is necessary to purchase a large number of nets and support dedicated net replacement and cleaning teams. In addition, frequent net changes may result in the damage or loss of fish in the cages and interrupt or disturb feeding, which may reduce the fish growth rate. However, the economic impacts of fouling and fouling control on the aquaculture industry cannot be quantified at present.

Shore-Based Net Cleaning

Usually, biofouling communities are removed from netting cages by replacing the fouled nets and transporting them to the shore for manual or semiautomatic cleaning (Lewis 1994a). However, the frequent replacement of nets in standard floating cages is labor- and capital-intensive, and large-scale cages require marine hydraulic cranes. Furthermore, washing procedures and the handling of nets often cause damage to nets and shorten their service life.

Underwater Net Cleaning

Underwater cleaning is feasible and commercially plausible; however, this approach usually requires diving, and thus, it is more expensive and dangerous than shore-based cleaning. An Australian company tested the efficacy of an underwater net cleaner, which could prevent biofouling for

10 weeks in summer (Hodson et al. 1997). However, due to physical constraints, the fouling organisms in the net grid or crevices were not completely removed, leading to the rapid regrowth and recolonization of the fouling organisms.

Many cleaning practices have been adopted in Norway. Three main strategies are employed to address biofouling on fishing nets, including cleaning nets onshore and in situ. Copper-based coatings are also used on cage netting in combination with regular on-site cleaning.

Biological Control

Increased profitability and sustainability can be achieved through the smart use of herbivorous fish or invertebrates to control biofouling (Beveridge 2004) and the use of benthic/debris feeders to remove uneaten food (Angel et al. 2002). The concept of biological control is limited by the dramatic changes in the fouling types of algae and invertebrates, which indicates that only herbivores and omnivores with extensive feeding habits are successful control agents. Furthermore, continuous grazing may provide a selective environment for nonfood species, and thus, polyculture may only reduce the frequency of net changes. For example, using sea urchins and resident crabs for biological control has been demonstrated to be effective in controlling the fouling of suspended shellfish systems (Lodeiros and Garcia 2004; Ross et al. 2004). In the case of finfish, biological control of fouling has been successful with the coculture of other finfish (Chua and Teng 1980).

In salmon aquaculture, detritivorous animals such as red sea cucumbers, *Parastichopus californicus*, which feed on broken or rotten food, have been shown to effectively reduce the growth of fouling organisms. However, due to the negative effects of wave-generated undulation, sea cucumbers cannot maintain their position on both sides of the cages suspended with buoys, although they can maintain their position in rigid frame pens (Ahlgren 1998). The advantage of polyculture with the sea cucumber is that sea

cucumbers are a key commercial aquaculture product, and there is a large demand for sea cucumbers in Asian markets (Conand and Sloan 1989).

Alternative Cage Designs

Another alternative to frequent net changing and underwater cleaning is the utilization of totally enclosed rotating cages (Willinsky et al. 1991). Such cages are either horizontally mounted cylinders rotating on a central axis (Menton and Allen 1991) or rectangular boxes with inflatable buoyancy devices at each corner. The rectangular cage rotates gradually by changing the buoyancy of the corners in turn (by inflation and deflation or by displacement and water filling). When using a rotatable cage, no area of the net needs to be immersed for a long time, and the net can be brought to the surface to air dry, thereby killing the attached biofouling. In addition, the cage can easily remove biofouling and help repair the net; moreover, significant growth of biofouling can be avoided by keeping the net immersed for only a short period of time. Blair et al. (1982) noted that a cage rotation of 90° per week was sufficient to keep cages basically free from biofouling. Geffen (1979) reported that cage rotation every 3 days could help keep cages completely clean.

Although completely enclosed and rotating cages have many other benefits, such as preventing birds from predation and avoiding storms and ice by cage submergence, rotating cages are not widely utilized. To hold volumes of fish comparable to conventional floating collars with a circumference >90 m, it would be necessary to construct extremely large rotating cages. Furthermore, commercial rotating cages are more expensive than traditional designs, and continuous exposure to direct sunlight can hasten the degradation of nets (Beveridge 2004).

Protective Coatings with Antifoulants

In the last 50 years, antifouling coatings have been intensively investigated. Chemical antifouling agents in coatings prevent the formation and establishment of marine biofilms through the leaching of bactericide, which can produce a thin layer of toxic solution around the net. Some of the

earliest published cage antifouling tests indicate that the antifouling agent keeps the net completely clean for 5 months, during which the untreated fishing net is completely blocked (Koops 1971).

However, only a few antifouling products have been specially designed for aquaculture cages, and the industry has typically learned from other marine industries, especially the shipping industry. Therefore, several antifouling products that contain chemicals and heavy metals clearly recognized as harmful to the environment have been used in the aquaculture industry. Sufficient public concern has been voiced regarding the use of these chemicals in aquaculture, particularly antibiotics and antifouling agents (Costello et al. 2001). Therefore, the broader aquaculture industry, including producers, regulators and research providers, must conceptually and realistically develop a safe industrial alternative.

Toxicity of Heavy Metal Organic Compounds

Tributyltin (TBT), one of the most widely used and “successful” (in addressing biofouling) antifouling agents on ship hulls since the 1960s (Yebra et al. 2004), is a broad-spectrum algicide, fungicide, insecticide, and acaricide. Because of its antifouling effect, TBT is also widely used as a cage coating in mariculture.

However, the use of TBT antifouling agents has demonstrated several hazards related to toxic coatings in mariculture (Terlizzi et al. 2001). TBT seeped from an impregnated net was recorded in the water around the treated cage (Balls 1987); in addition, the release of TBT from a newly coated cage was measured, with Sn contents of 1 mg/m³, 0.1 mg/m³, and 0.005 mg/m³ discovered after 1 day, 2 weeks, and 5 months, respectively.

The use of TBT-impregnated nets in salmon aquaculture resulted in histopathological effects (Bruno and Ellis 1988) and mortality (Lee et al. 1985). Moreover, salmon raised in treated nets could rapidly bioaccumulate TBT. Short and Thrower (1986) reported bioaccumulation after 3–4 days of exposure to 1.5 pg/L. The authors recorded wet weights of 6.4, 1.9, and 0.3 pg TBT/g for the liver, brain, and muscle, respectively.

Moreover, TBT antifouling products pose an unacceptable risk to nontarget species, which cannot be identified when introduced in the market. Furthermore, TBT induces sexual aberrations in gastropods (Gibbs et al. 1991) and has since been found in fish, bivalves, seabirds, and marine mammals (Terlizzi et al. 2001 review).

The adverse effects of the widespread use of TBT have led to a ban on its use (Evans 1999). In 1986, the National Farmers' Union of Scotland voluntarily participated in the prohibition of its use on fish cages, and in 1987, the Scottish government banned its retail sale (Balls 1987). The International Maritime Organization has banned the use of TBT in coatings since 2003 (Julian 1999), and many governments have thus prohibited the use of organotin in antifouling coatings (Costello et al. 2001).

The aquaculture industry is facing future developmental challenges. To ensure that similar TBT scenarios are not repeated, the six criteria followed by the aquaculture industry are as follows (Lewis 1994b): (1) must be effective for a wide range of fouling groups, (2) must be harmless to the environment, (3) must exert no negative impact on the cultured species, (4) must leave no residues in cultured species, (5) must be capable of withstanding onshore handling and cleaning, and (6) must be economically feasible.

Feasibility of Copper-Containing Coatings

After the implementation of the TBT ban, people soon focused on copper- and copper-containing coatings, which have a long history of applications in shipping and mariculture (Lewis 1994b). The main active components in copper-based antifouling agents are cuprous oxide and copper. Copper is leached from the coating of impregnated nets into seawater. To prevent barnacles and diatoms from adhering and growing, leaching rates of 10 and 20 $\mu\text{g}/\text{cm}^2/\text{day}$ should be applied for response treatments and measures, respectively (Callow 1999). However, relatively low concentrations of copper are known to be harmful to fish and many marine organisms, especially invertebrate larvae, and toxicity studies have reported diverse effects (Brooks and Mahnken 2003). Furthermore, in salmon farms, the

intestinal copper content of green sea urchins was noted to be increased (Chou et al. 2003), and copper bioaccumulation was found in the hepatopancreas and oysters of crayfish sampled near the salmon farm (Chou et al. 2000).

Fish are likely to be exposed to antifouling agents for long periods, up to several months. Therefore, from a "clean and green" marketing perspective, the use of toxic metal antifouling agents, i.e., simply using copper as an antifouling compound, is relatively undesirable. At present, most countries are committed to reducing the use of copper-based antifouling agents in the short term.

Other Biocides

Many other biocides are currently used as antifouling agents worldwide; although these products are not necessarily used in mariculture (Konstantinou 2006), they are potential candidates to supplement or replace copper as an antifouling agent. The most commonly used biocides include chlorothalonil, dichlofluanid, diuron, irgarol 1051, sea-nine 211, thiram, TCMS (2,3,5,6-tetrachloro-4-methylsulfonyl), pyridine, pyrithiones, TCMTB (2-thiocyanomethylthiobenzo-thiazole), zinc pyrithione, and zineb (Konstantinou 2006). Coatings using isothiazolinones as the sole biocide have been successfully tested in Australia (Svane et al. 2006); however, little peer-reviewed literature exists on the efficacy of other biocides tested in aquaculture environments.

The known chemical and physical properties of common biocides vary widely, and their properties, toxicity, environmental fate, and knowledge gaps in the aquatic environment have been extensively reviewed (Konstantinou and Albanis 2004; Konstantinou 2006). A potential danger remains because the listed biocides are largely untested in some cases, and they may be less efficient and/or more harmful to the environment than TBT or copper (Evans 1999). Moreover, the biocides released during the cleaning process persist in the environment when associated with paint particles (Thomas et al. 2003). Considering the gaps in our understanding of the long-term effects of biocides, it is difficult to assess their impacts

Net Structures: Biofouling and Antifouling, Table 1 Key data for a comparative environmental assessment of relevant biocides

	Persistence ^a	Toxicity	Bioaccumulation	Performance ^b
Chlorothalonil	No			Poor
Dichlofluanid	No			Poor
Diuron	Yes	Least	No	Poor
Irgarol 1051	Yes	Least	Yes	Poor
Sea-Nine 211	No		No	Poor
Zinc pyrithione	No			Best
Zineb			No	Best
Thiram			No	
Pyrithiones		High		
Pyridine				Poor
TCMTB				Poor
TCMS				Poor

^apersists in the water column

^bhighest performance in terms of environmental parameters; TCMS pyridine and TCMTB demonstrate environmental characteristics similar to TBT (Voulvoulis et al. 2002)

and risks on the aquatic environment; therefore, effective environmental policies must be formulated according to precautionary principles. Key data are presented in Table 1 regarding a comparative environmental assessment of biocides (Voulvoulis et al. 2002).

To date, all developed synergistic biocides affect the water environment, and no ideal substitute exists for TBT or copper. Moreover, the aquatic environment is affected by booster biocides, and no ideal replacement exists for either TBT or copper. The focus of future research and development has shifted to both effective and environmentally friendly antifouling agents in terms of their chemical (nontoxic coatings) and physical properties (fouling release coatings and nonleaching biocides) (Yebra et al. 2004).

Development of Environmentally Benign Coatings

Natural Biocide Products

Natural products have had a long history in aquaculture. Before using modern polymer-based netting, farmers in Malaysia soaked cotton nets with tannins extracted from the bark of mangroves (*Rhizophora sp.*). Tannins are toxic and function as natural biocides. These tannins have a high

absorption capacity for traditional fiber nets but low absorption capacity for synthetic materials, and the effectiveness is usually short term (Lai et al. 1993).

However, the potential for the use of natural antifouling agents is limited because they are simply another chemical entity, and considerable commercial development time is required in the regulatory and commercial environment to prepare surface effect coatings. This kind of compound must be synthesized in large quantities at a reasonable cost, added to the coating matrix, and subject to the same regulatory assessment as biocides by environmental agencies before its practical application can be formally adopted. Fusetani and Clare (2006) comprehensively reviewed antifouling compounds from natural sources, including potential effective compounds.

Fouling Release Coatings

In recent years, research has focused on biocide-free low surface energy siloxane elastomers and fluoropolymers, which may provide a nontoxic alternative to control biofouling and facilitate the application of such coating products (Yebra et al. 2004; Chambers et al. 2006). These “fouling release” coatings are designed to reduce or prevent the adhesion of biofouling, and silicone-based coatings are not toxic to any tested

organism (Karlsson and Eklund 2004). Nevertheless, nets and panels coated with nontoxic silicone effectively reduce the initial stage of fouling development and make it easier to remove the accumulated biofouling (Hodson et al. 2000; Terlizzi et al. 2000). At present, these materials are considered substitutes for toxic coatings on the surface of ship hulls, for which the speed of the vessel produces the hydrodynamic shear force required for loosely attached biofouling to fall off (Yebra et al. 2004).

However, the commercial development of these technologies and their application in fixed aquaculture infrastructure are difficult, and research to solve these problems requires additional time. The principle of superhydrophobicity has been further developed in the field of fouling release technology (Genzer and Efimenko 2006; Marmur 2006). This principle can likely be combined with the abovementioned technologies to solve the biofouling problem of relatively static cage nettings.

Nonleaching Biocides Coatings: Contact Type

Biocides that irreversibly bind (via covalent bond grafting) to the surface of antifouling coatings or antifouling nets are called nonleaching biocides (Clarkson and Evans 1993, 1995). Although this approach can limit environmental contamination, it has not been successfully implemented, possibly due to technical problems and a wide range of fouling organisms, many of which may not respond to bound biocide entities. These technologies have been effectively used to inhibit the biological attachment and growth of bacteria on biomedical devices (Hume et al. 2004; Zhu et al. 2008). Under legislation restricting antifouling technology to a nonrelease mechanism, this aspect may be promising with broad prospects (Silva et al. 2019).

Microtexturing of Surfaces

Research on the identification of physical defenses against biofouling in specific animals and plants may also have commercial applications in antifouling technologies. Such methods are characterized by topography and microstructure surfaces to prevent the attachment of common

fouling organisms. For instance, natural regular rippled surface features have been visualized on blue mussels (Scardino et al. 2003), and these structures significantly inhibit the settlement and development of biofouling (Scardino and de NYS 2004).

Surface microtopographies, many with a bioinspired design, inhibit the attachment and growth of specific fouling organisms and promote their release (Scardino et al. 2006, 2008). In many cases, the efficacy depends on the proportion of morphology, i.e., micro/nanostructures, relative to the size of the settled larvae or propagules, which limits the surface effectiveness to a limited range of fouling organisms. To enhance the effectiveness of prevention or broaden their deterrent impacts, surfaces with multiple scales of topography are now being developed (Schumacher et al. 2007). This nonleaching surface effect technology can be used in combination with fouling release coatings and may be effective in preventing biofouling in aquaculture, in which the dominant single species in the fouling community can be targeted.

Conclusions

At present, methods of protecting against fouling on fishing nets and other aquaculture structures are relatively limited, and most of them are based on the release of copper and zinc by adding booster biocides. The use of zinc and copper is restricted by the International Convention on the elimination of zinc, and thus, the use of such products may be limited as well. Consequently, specific biocides with a low environmental impact are the only mechanism for biofouling control. However, new products can be developed with emphasis on antifouling technologies based on low surface energy (foulant-release) coatings, textures, and surface-bound compounds. Unfortunately, the fouling release technology mainly depends on the hydrodynamic force to remove fouling organisms with low adhesion on the fouling-release surface, making such methods less suitable for aquaculture. Moreover, with the continuous improvement of vessel antifouling technology, the transfer of technology to

aquaculture industries is expected to become more feasible, and product development is expected to be targeted at larger aquaculture industries. Another alternative to heavy metal and biocide technologies, biological control, has yet to be proven to have broad-spectrum efficacy in controlling biofouling.

Although such methods are industry- and site-specific and it is difficult to envision their wide application, certain industry sectors may significantly benefit. On the other hand, with the withdrawal of technologies based on metals and biocides from the market, the aquaculture industry is expected to become greener and environmentally friendly. More and better technologies are expected to emerge as solutions for biofouling protection.

Cross-References

- [Aquaculture Structures: Experimental Techniques](#)
- [Aquaculture Structures: Numerical Methods](#)
- [Modern Aquaculture Structures](#)
- [Net Structures: Design](#)
- [Net Structures: Hydrodynamics](#)
- [Traditional Aquaculture Structures](#)

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Net Structures: Design

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Synonyms

Copper Alloy Net; Metal Net; Polyamide (PA) Net; Polyethylene (PE) Net; Ultrahigh Molecular Weight Polyethylene (UHMWPE) Fiber Net

Introduction

A net is a porous structure composed of net twine constructed by twisting and weaving. Nets are commonly applied in fishery production (e.g., fishing industry, cage aquaculture, and net enclosure aquaculture) and engineering construction (e.g., building protection and area isolation). In this chapter, we describe the application design of

a net in cage aquaculture and net enclosure aquaculture (NEA).

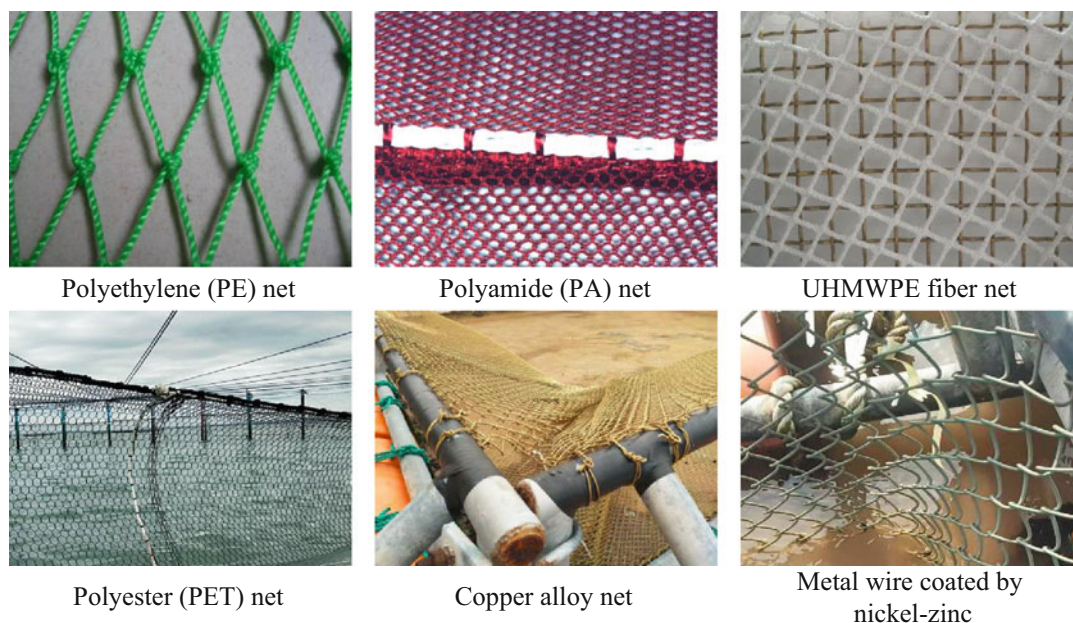
Net Types

In practice, synthetic fibers, metals, and other new types of materials are commonly used for net making. Among these materials, polyethylene nets, polyamide (PA) nets, ultrahigh molecular weight polyethylene (UHMWPE) fiber nets, polyester PET (polyethylene terephthalate) nets, copper alloy nets, and coated metal wire nets have been widely used in cage aquaculture and NEA (see Fig. 1).

Polyethylene (PE) Net: Polyethylene monofilaments are twisted to form net twines and later woven to form a net. Polyethylene is flexible and exhibits a high tensile strength, high impact resistance, and low density (less dense than water, approximately 953 kg/m^3). The net twine used in PE nets is normally composed of 3 strands, and the number of monofilaments per strand is set according to the mesh size and twine strength. PE netting is one of the most widely used net types in cage aquaculture.

Polyamide (PA) Net: Polyamide, commonly known as nylon (nylon), was invented in the early 1930s and rapidly became the leading industrial product of polymer synthetic fibers. Its favorable characteristics include its high strength, high abrasion resistance and good elasticity, and the density (approximately 1140 kg/m^3) is slightly higher than that of water. Generally, the single-yarn diameter of PA fibers used as net materials is 0.66–2.22 tex, considerably smaller than that (36 tex) of PE monofilaments. Under the same diameter, the breaking strength of PA fibers is generally slightly higher than that of PE fibers.

Ultrahigh Molecular Weight Polyethylene UHMWPE Fiber Net: UHMWPE fiber was successfully developed by the DSM company (Netherlands) in 1979. This net has many advantages, such as a high strength, low density, high fatigue resistance, and low temperature resistance. This material was once used to prepare bullet-proof jackets. Research has demonstrated that the strength of a net cable woven from UHMWPE fibers is 3–4 times that of a PE net cable, demonstrating a considerable advantage in strength.



Net Structures: Design, Fig. 1 Nets commonly used in the farming industry (by Fukun Gui)

However, the UHMWPE fiber weaving process is more complicated than that of PE netting, and the price of a UHMWPE fiber net is 5–6 times that of a PE net. Therefore, UHMWPE fiber nets are often applied in important aquaculture projects with high manufacturing costs (such as cage aquaculture projects in the open sea and large-scale NEA projects).

Polyester (PET) Net: The PET net is a newly developed net that exhibits a hexagonal mesh structure made of a monofilament structure through a special twisting process. The diameter of a PET net monofilament is generally approximately 2.5–4 mm, and the strength is approximately 1–1.5 times higher than that of an ordinary PE net with the same diameter but lower than that of a UHMWPE net. PET nets exhibit an excellent fatigue resistance strength. Owing to the monofilament structure, the surface of the net is smooth and has no voids. Moreover, the biofouling rate is lower than that of other fiber nets, which helps to relieve the impacts on nets from currents and waves. Furthermore, the price of a PET net is close to that of a UHMWPE net, and the comprehensive cost-performance ratio of

PET nets is higher. The PET net has been gradually applied in key farming projects.

Metal Net: Metal nets in farming projects are usually made of copper alloy wire or steel wire coated by nickel-zinc. The abovementioned metal nets exhibit excellent antifouling characteristics, and thus, they have been applied in cage aquaculture and NEA projects. However, the application of copper alloy nets is restrained by their high cost compared with other types of nets currently used in farming projects.

Key Technologies in Net Design

The design of the net system in aquaculture projects generally requires several key procedures, such as the net model design and material selection, net load calculation, net strength design, and net assembly design.

Net Model Design and Material Selection

The selection and design of nets are mainly based on the functional requirements of aquaculture. Generally, the following main factors are considered:

Mesh Size Generally, the mesh size of the net should be chosen according to the actual size and body characteristics of the farming fishes. For small fish fry, the mesh size of the net can be small. For large fish fry, a larger mesh size should be selected to ensure sufficient water exchange. In the entire farming process, as the body length of the fishes increases, the nets should be replaced over time by nets with larger meshes. Therefore, it is necessary to rationally design nets with several different mesh sizes.

Preliminary Selection of the Twine Diameter First, the net load strength must be calculated to determine the diameter of the net twine. The selection of the twine diameter depends on not only the strength requirements but also the weaving process of the mesh. A certain relationship exists between the twine diameter and mesh size of the net. Smaller meshes generally adopt a smaller diameter of twine. Choosing a twine diameter that is excessively large may lead to the failure of net fabrication. Under normal circumstances, the twine diameter is initially determined according to the mesh size and later optimized according to the load strength of the net. In terms of the twine diameter of the polyethylene net, Table 1 (Shi et al. 2016) can be referred to for the preliminary selection.

When the mesh size exceeds 40 mm, the number of twine strands varies according to the practical use. The corresponding twine diameter can be estimated according to formula (1):

$$D = 0.022 \times G + 1.1865 \quad (1)$$

where D represents the twine diameter and G represents the number of twine strands.

Selection of the Net Material When choosing the net material, several key factors should be carefully considered, such as the cost of the net, the demanded strength and toughness, and the antifouling capability. Among the nets commonly used in aquaculture projects, copper alloy nets exhibit a favorable antifouling performance. However, they are the most expensive option, and their toughness is insufficient. UHMWPE fiber nets exhibit excellent strength, but the cost is relatively high, and their antifouling function is inadequate. PET nets have a relatively high tensile strength and excellent fatigue strength, and their antifouling performance is higher than that of other fiber nets; however, the cost is also high. For the PE net, the overall strength is lower than that of the abovementioned materials, and the antifouling functionality is lacking; however, PE nets are inexpensive, which contributes to their higher cost-performance ratio. Therefore, PE nets have been widely used in aquaculture projects. In recent years, the development of large-scale deep-sea aquaculture equipment and ultra-large NEA equipment has imposed higher requirements on the strength of the net; therefore, UHMWPE fiber nets and PET nets are attracting increasing attention.

Calculation of the Net Load

The net load is an important basis for the net design. The selection of the net material and twine diameter is based on the calculated net load. Three methods can be used to calculate the net load: theoretical analysis, physical model testing, and numerical simulation. Regardless of which method is adopted, the design standard of

Net Structures: Design, Table 1 General specifications of the PE net

Mesh size (mm)	Twine specification	Twine diameter (mm)	Weight (g/100 m)	Breaking strength (kgf)
0.5–1	0.23/1	0.23	4.36	2.37
1–2	0.23/1×2	0.46	9.33	3.55
3–10	0.23/1×3	0.53	14.0	5.32
10–13	0.23/2×2	0.67	17.0	6.62
13–20	0.23/2×3	0.78	28.0	9.94
20–25	0.23/3×3	0.96	42.0	14.9
25–30	0.23/4×3	1.13	56.0	18.4
30–40	0.23/5×3	1.29	67.0	23.0

the farming project must be determined first. The selection of the design criterion for a farming project depends on the conditions of the sea area in which the farming facility is located and the importance and influence of the farming project. For general near-shore cage culture, the design criterion is generally set to once every 25 years. For aquaculture projects in offshore waters, a design criterion of once every 50 years is recommended. For large-scale sea cage aquaculture in the outer sea and NEA projects, the design criterion is generally set to once every 100 years due to their significance and influence.

Theoretical Analysis Method The core purpose of the net load calculation is to determine the load strength of the key parts in the net system. Necessary simplifications and assumptions to the calculation model are often implemented when carrying out a theoretical analysis of the net load in aquaculture engineering.

The flow load acting on the net unit can be calculated by the general hydrodynamics equation (2):

$$F_D = \frac{1}{2} \rho \cdot C_D \cdot A \cdot u^2 \quad (2)$$

where F_D represents the flow force acting on the net; ρ represents the fluid density; C_D represents the drag coefficient of the net; A represents the projected area of the net; and u represents the fluid velocity.

The inertial force produced by unsteady flow dynamics can often be ignored due to the small-scale structure of the net, and thus, formula (2) can be applied to a net force analysis under wave conditions with u representing the fluid point velocity. The net load is calculated by the Stokes finite amplitude wave theory and Airy linear wave theory in deep-water sea cage aquaculture engineering. The third- or fifth-order Stokes finite amplitude wave theory can be used to calculate the net load when cage aquaculture or NEA projects are constructed in shallow waters. However, when the water depth is extremely shallow or there is a shore-linking section (such as shore-linked NEA), the theoretical calculation may

induce a large error, and it is generally necessary to carry out a special study.

The drag coefficient C_D is related to the impact angle of the current, Reynolds number, and solidity of the net and other parameters. According to the existing research literature (Li and Gui 2005), the drag coefficient of the mesh twine can be calculated by the following formula when analyzing the force of the net:

$$C_{D90} = 3(\text{Re})^{-0.04} \quad (3)$$

$$C_{D\theta} = C_{D90} (1 - \cos^2 \theta \cos^2 \varphi + \sqrt{S_n} \cos^2 \theta \sin \varphi \cos^3 \varphi - 3S_n \cos^3 \theta \sin^2 \varphi) \quad (4)$$

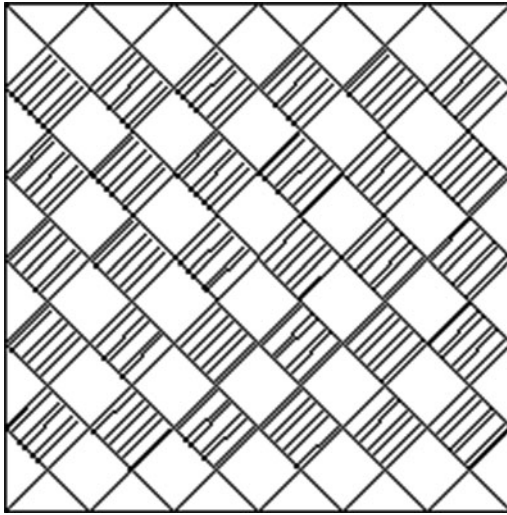
Regarding the drag coefficient of the knot, Fredheim and Faltinsen (2003) suggested that it is reasonable to use a drag coefficient in the range of 1.0–2.0 when modeling the knot part as a sphere.

The projected area of the mesh is related to the impact angle of the mesh unit and mesh structure. In general, the net mesh has rhomboidal, rectangular, and hexagonal structures, which can be classified into nodular and nonnodular types. Rhomboidal and hexagonal meshes are the most commonly used. A rhomboidal mesh consists of four wires and four knots, as shown in Fig. 2. A knot is shared by four meshes, and a twine is shared by two meshes; specifically, a mesh consists of a knot and two mesh twines, as shown in Fig. 3. Therefore, this definition is used when calculating the projected area of the mesh.

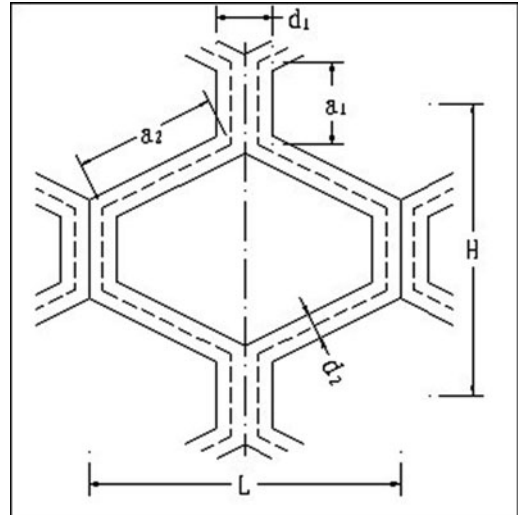
The inclination should be considered to calculate the projection area of a rhomboidal mesh when the mesh is slanted. Assuming that the angle between the mesh and direction of water flow is θ , the projected area of the mesh unit can be calculated as follows (Gui 2006):

$$A = a \cdot d \cdot \sqrt{\mu_1^2 + \mu_2^2} \cdot \sin^2 \theta \cdot N \quad (5)$$

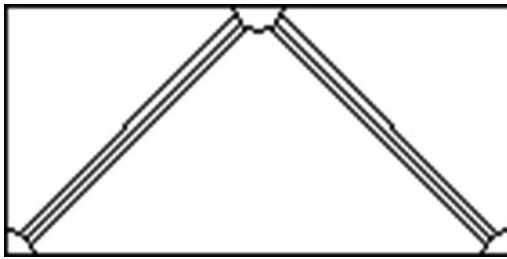
where A represents the projection area of the net; a and d represent the length and diameter of the twine, respectively; μ_1 and μ_2 denote the hanging



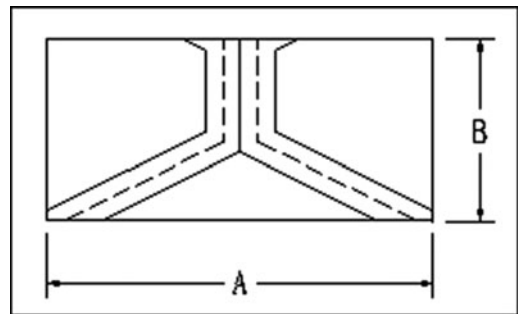
Net Structures: Design, Fig. 2 Sketch of a diamond mesh (Gui 2006)



Net Structures: Design, Fig. 4 Sketch of a hexagonal mesh (Gui 2006)



Net Structures: Design, Fig. 3 Definition of a single diamond mesh (Gui 2006)



Net Structures: Design, Fig. 5 Definition of a single hexagonal mesh (Gui 2006)

ratio in the horizontal and vertical directions, respectively; and N refers to the total number of meshes in the net unit.

For a hexagonal mesh without knots, as shown in Fig. 4, the calculated features mainly include the height H and width L of the mesh, length a_1 and diameter d_1 of the bevel mesh, and length a_2 and diameter d_2 of the vertical mesh. In the case of a certain inclination angle, the expressions to calculate the projected area of different mesh segments are different, and these aspects should thus be considered separately in the calculation. Similar to a mesh with knots, one mesh is equivalent to one twine with diameter d_1 and length a_1 and two twines with diameter d_2 and length a_2 , as shown in Figs. 4 and 5. The calculation of the projected area of a mesh can start from the calculation of the projected area of a single mesh. Assuming that

the impact angle between the plane net and the water flow is θ , the projected area of the plane net at this impact angle can be calculated according to the following formula.

$$A = \left(a_1 d_1 \sin \theta + d_2 \cdot \sqrt{(L - d_1)^2 - (H - 2a_1)^2 \cdot \sin^2 \theta} \right) \cdot N \quad (6)$$

where A is the projected area of the mesh, θ represents the water impulse angle, and N denotes the total number of meshes in the net unit.

A general biofouling situation will occur in the fiber net after marine farming for a certain period,

leading to an increase in the projected area of the net. In the actual calculation, the effect of biofouling should be considered.

Physical Model Tests

Modeling Criteria for Fishing Nets in Experiments Model experiments are highly useful for examining the hydrodynamic characteristics of fishing nets; however, it is challenging to establish reasonable simulation criteria for the model tests of fishing nets. For such model tests, the geometric scale λ is usually greater than 20:1. If the model net is designed strictly according to geometric similarity principles, two challenges arise. First, the yarn used for the model net will be extremely thin, which renders the manufacturing process difficult. Second, the Reynolds number (Re) for the model net will change significantly because the mesh is modeled to be extremely small, leading to a large difference in the hydrodynamic behavior between the model and prototype nets. Thus, special simulation methods must be used. Tauti's criteria (Tauti 1934), which were developed in the 1930s during research into fishing net problems, are used only for conditions where a current is present.

Tauti's Simulation Criteria Tauti's simulation criteria involve two geometric scales for nets, λ and λ' , which represent the global and model mesh scales, respectively:

$$\lambda = \frac{L_p}{L_m} \text{ and } \lambda' = \frac{d_p}{d_m} = \frac{a_p}{a_m}, \quad (7)$$

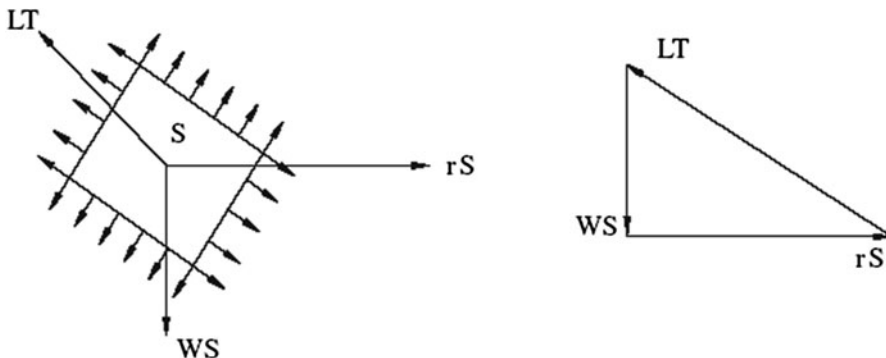
where d is the filament diameter, a is half the mesh size, and subscripts p and m denote the model and prototype sizes, respectively. Based on the geometric similarity and dynamic similarity, the following expressions can be obtained according to Fig. 6:

$$\frac{W_p S_p}{W_m S_m} = \frac{r_p S_p}{r_m S_m} = \frac{T_p L_p}{T_m L_m} = \frac{F_p}{F_m}, \quad (8)$$

where W and r denote the gravity and hydrodynamic force per unit area on the net, respectively, and T is the force per unit length on the net edge. When the net material is the same for the model and the prototype, the equation can be replaced by

$$\lambda^2 \lambda' = \lambda^2 \frac{V_p^2}{V_m^2} = \lambda \frac{T_p}{T_m} = \frac{F_p}{F_m} \quad (9)$$

In trawl experiments, Tauti's simulation criteria are suitable and widely used. However, in model tests of net cages, two limitations arise for practical experiments. First, in model tests, the model mesh scale cannot be excessively large, and thus, a greater current velocity is required. Second, Tauti's simulation criteria cannot be applied to model tests under wave conditions because Eq. (9) is not satisfied if geometric similarity is applied in the presence of waves. Thus, for model tests of net cages under wave and current conditions, the following gravity simulation criteria are recommended.



Net Structures: Design, Fig. 6 Model to calculate forces acting on a micro-segment of a fishing net (Gui 2006)

Extended Gravity Simulation Criteria (Gui 2006) The main principles for an extended gravity simulation are as follows:

1. Two geometric scales for the nets are used as in Tauti's simulation criteria: λ and λ' represent the global scale and the model scale for the yarn diameter and net mesh size, respectively.
2. Since the porosity ratio for the model net is the same as that for the prototype, the external forces acting on the net will follow the gravity simulation criteria by a scale of λ^3 .
3. The model net weight simulated by a scale of λ^3 is modified as follows:

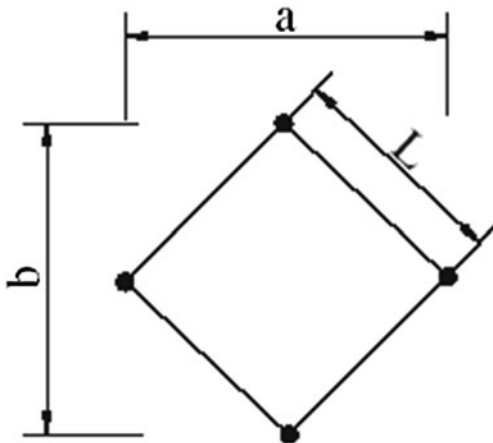
$$\Delta W = \left(\frac{1}{\lambda'} - \frac{1}{\lambda} \right) \cdot \left(\frac{\pi d_1^2}{4a_1\mu_1\mu_2} \times 10^4 \right) \cdot (\rho - \rho_w) \cdot q \cdot S, \quad (10)$$

where ρ is the density of the net material, ρ_w is the water density, q is the packing fraction of the filament, S is the area of the net, and μ_1 and μ_2 are the hanging ratios (see Fig. 7) defined in Eqs. (11) and (12), respectively.

In the crosswise direction of the net, the hanging ratio is defined as

$$\mu_1 = a/2L \quad (11)$$

In the longitudinal direction, the ratio is



Net Structures: Design, Fig. 7 Definition of mesh properties (Gui 2006)

$$\mu_2 = b/2L. \quad (12)$$

Based on the gravity simulation criteria, the following equation can be derived:

$$\lambda^3 = \lambda^2 \frac{V_p^2}{V_m^2} = \lambda \frac{T_p}{T_m} = \frac{F_p}{F_m}. \quad (13)$$

Numerical Simulation Method

It is necessary to establish an appropriate calculation model when using a numerical simulation method to analyze the net load. At present, the most widely used method for flexible fiber nets is the lumped-mass model. The finite element method is generally used for more rigid nets, such as those involving a copper mesh, using mature commercial software such as Ansys. Here, we simply introduce a lumped-mass model for flexible fiber nets.

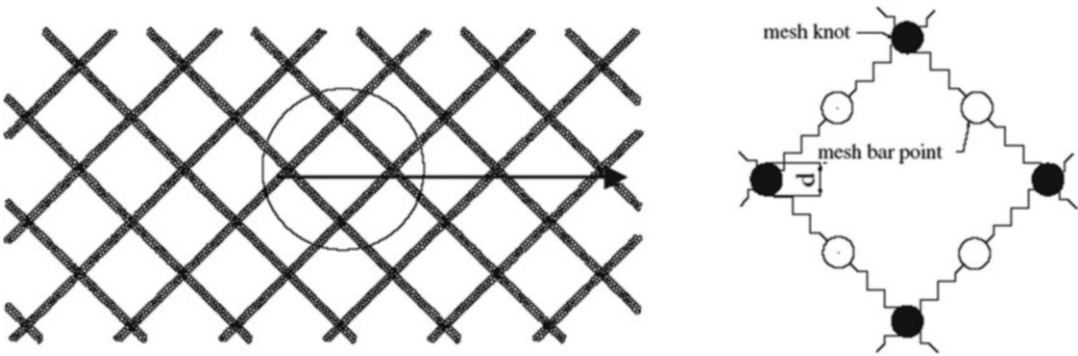
Lumped-Mass Model By applying a lumped-mass model (Li et al. 2006), a fishing net is assumed to be a connected structure with limited masses and springs. The lumped point masses are set at each knot and at the center of each mesh twine (see Fig. 8).

Depending on the use of knots, two different types of nets are commonly used: nets with knotted meshes and nets with knotless meshes. In the model, the diameter of the sphere d reflects the physical properties of the net. For a mesh with knots and a knotless mesh, d is considered to be 3.14-fold and 1.5-fold greater than the diameter of the twine, respectively.

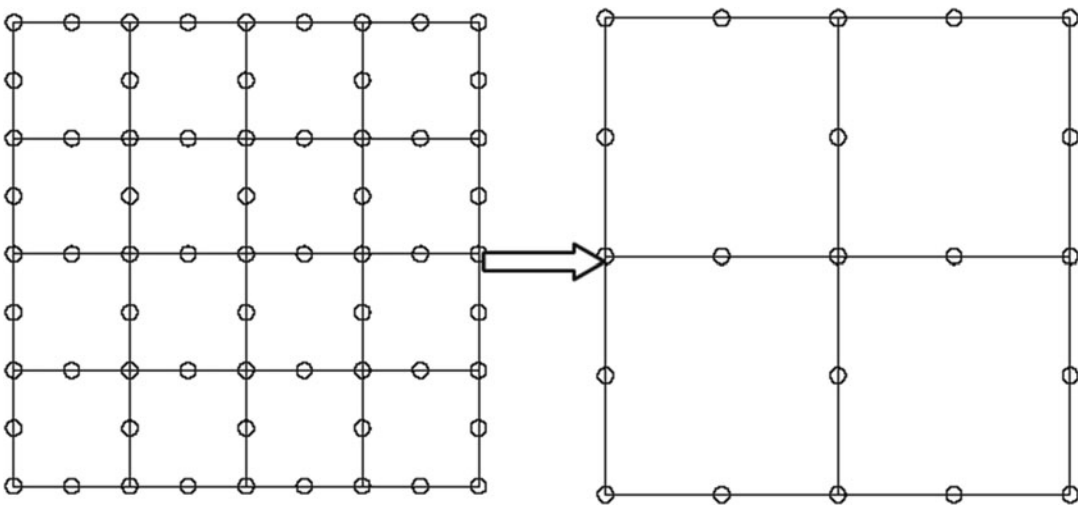
According to Newton's second law, the equation of motion for a lumped mass i in waves can be expressed as

$$M_i a = \sum_{j=1}^n \vec{T}_{ij} + \vec{F}_D + \vec{F}_I + \vec{W} + \vec{B}, \quad (14)$$

where M_i is the lumped mass of point i , a is the acceleration of mass point i , $\vec{T}_{ij}$ is the vector of the tension in twine ij (j represents the knots at the other end of twine ij), n is the number of knots adjacent to point i , $\vec{F}_D$ is the drag force, $\vec{F}_I$ is the inertial force, $\vec{W}$ is the gravity force, and $\vec{B}$ is the



Net Structures: Design, Fig. 8 Schematic of the mass-spring model (Li et al. 2006)



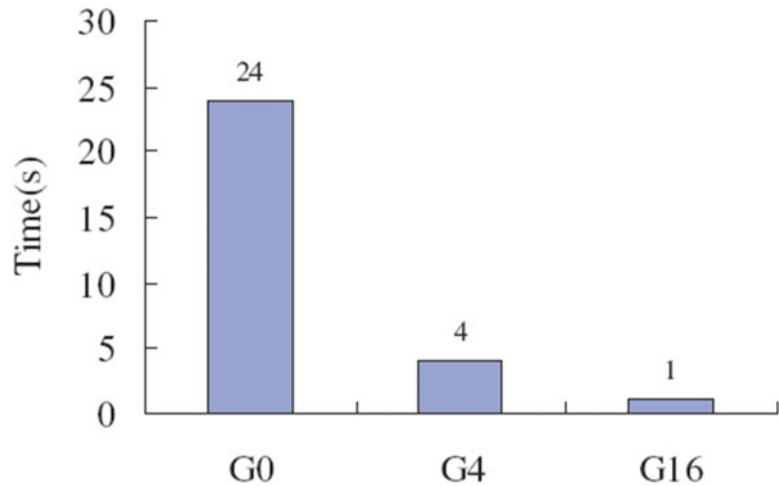
Net Structures: Design, Fig. 9 Sketch of the mesh grouping method (Zhao 2007)

buoyancy force. According to research, the inertial force $\vec{F}_I$ on a fishing net is rather small under wave conditions compared to other external forces and can thus be omitted. The drag force $\vec{F}_D$ can be calculated using equation (2), with the drag force coefficient C_D calculated by equations (3) and (4).

Equation (14) can be solved using Runge-Kutta methods. The net shape and force distribution at each time step can be calculated numerically by solving these ordinary differential equations for a given initial condition.

Mesh Grouping Method When all the knots in the mesh and at the twine are used as mass points,

more than 10,000 mass points exist for a common plane net, for which the calculation time in a practical simulation is extremely large. To reduce the computational cost, a mesh grouping method can be employed in the calculations. The method consists of modeling a given number of actual meshes as a fictitious equivalent mesh that has the same physical qualities as the actual meshes, such as the projected area of the net, specific mass, and weight. Figure 9 shows an example of a 4×4 mesh with 65 lumped mass points that is approximated to a 2×2 mesh with 21 lumped mass points. According to calculations by Zhao (2007), using 1×1 (G0), 2×2 (G4), and 4×4 (G16) meshes to model the same panel yields

Net Structures: Design,**Fig. 10** Comparison of the calculation time for different mesh grouping methods (Zhao 2007)

similar results for the predicted hydrodynamic forces for current-only loading, although with a considerable time reduction, as shown in Fig. 10.

Strength Design of the Net

The wire with the maximum tension in the net is generally considered to have the basic properties for the net strength design after the net load distribution is obtained. The net strength is determined mainly by the diameter of the mesh but is also affected by the knot form and mesh shape. It is still necessary to consider the attenuation of mesh strength caused by aging and biofouling when considering the service life of the net. Therefore, the safety factor K , which equals 3–4, is recommended for net design; specifically, the strength of the net selected for use should be 3–4 times the calculated load. Stress concentration is strictly forbidden to appear when performing net design. If the situation is unavoidable, special protection or reinforcement should be introduced.

Net Assembly Design**Net Sewing**

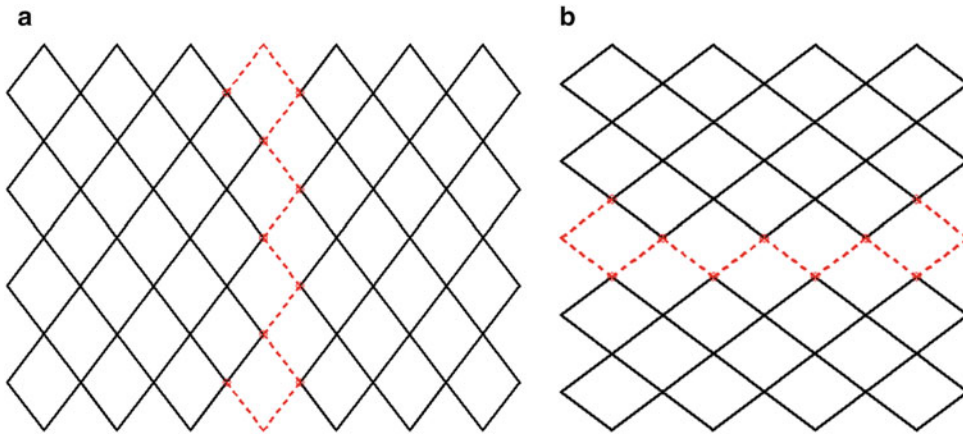
First, a certain width of net panel is established during net manufacturing, and net cutting and sewing are performed according to the overall structure of the cage or NEA. The common methods of net suturing in aquaculture engineering include braiding and winding.

Braiding involves connecting two pieces of net panels and can be categorized as longitudinal and transverse braiding according to the direction of the braiding mesh, as shown in Fig. 11. The material and diameter of the net line required for knitting should be the same as that of the net. The size of the mesh formed by knitting and the knotting method must also be the same as those for the original net. The net forms a complete and integral unit after knitting, which involves strict requirements in terms of manual weaving techniques.

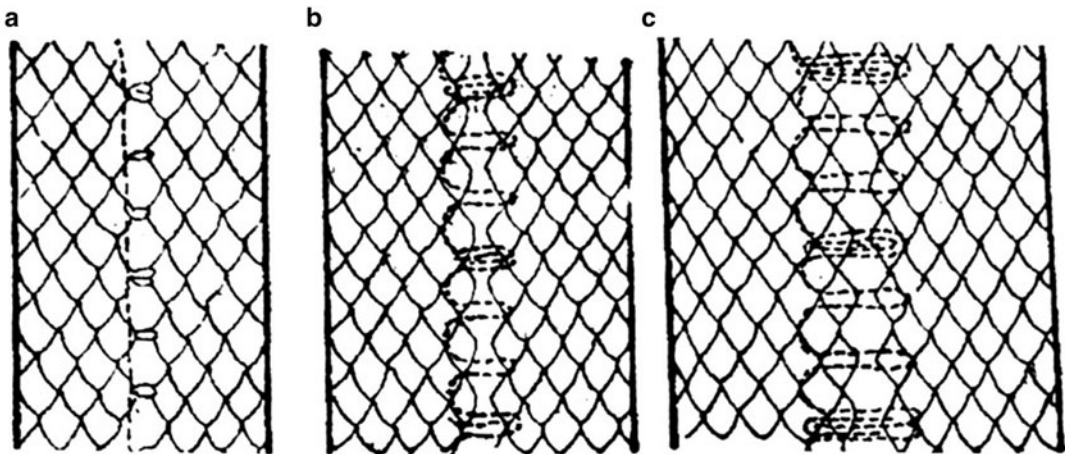
Winding involves the connection of two net panels. The requirements for winding are relatively approximate, and such methods are generally used for emergency repair or sewing processes, which do not involve high requirements. Winding can be categorized as ordinary winding (see Fig. 12) and cutting winding (see Fig. 13).

Rope Assembly

After the net is sewn, it needs to be assembled with the rope. The rope is an important structure that transfers the net load to the cage frame or NEA posts. It is necessary to consider the net load and select the rope strength by referring to the design principle of the net strength. After the rope has been selected, it is necessary to sew the net evenly on the rope with an appropriate method and ensure that the net forms the expected shrinkage shape. This process is known as rope assembly.



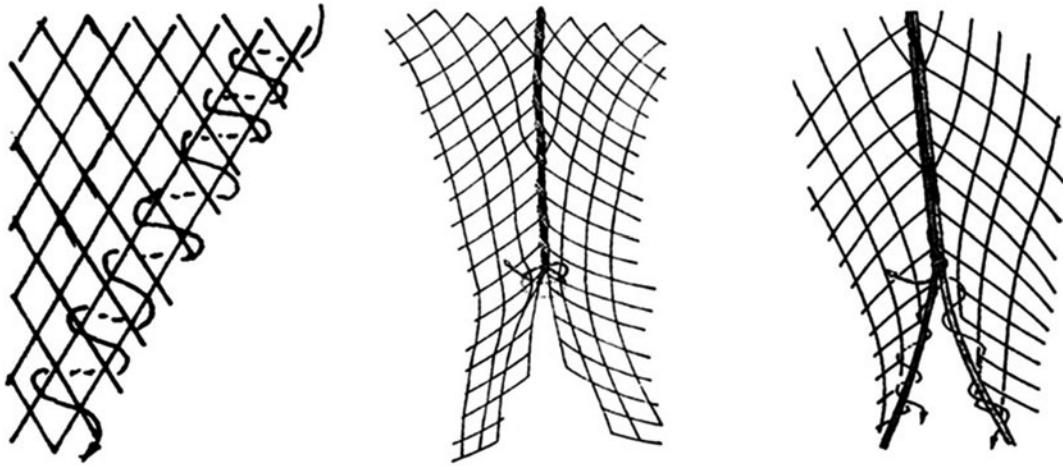
Net Structures: Design, Fig. 11 Net sewing methods (by Fukun Gui) (a) Longitudinal sewing (b) Transverse sewing



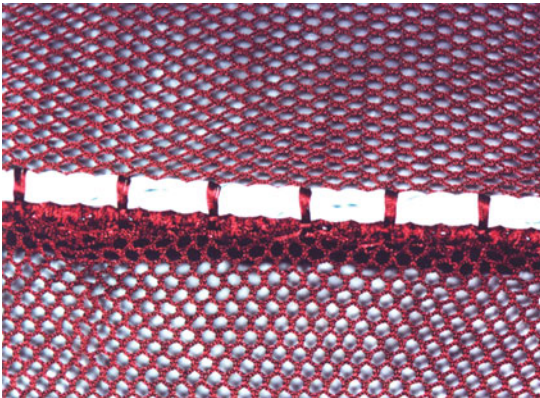
Net Structures: Design, Fig. 12 Ordinary winding methods (Shi et al. 2016) (a) Half mesh winding (b) One mesh winding (c) Multimesh winding

The net and rope are generally divided into several parts according to the mesh size and rope length during the course of assembly (see Fig. 14). The parts are marked at equal points and then assembled segment by segment to ensure the uniformity of the mesh distribution. For items with higher overall requirements for the rope assembly, the net is stretched and fixed according to the design size, and then the ropes are sewn onto the net. The ductility of the rope should be considered in the net assembly. The rope should be pre-tensioned before assembly to eliminate the plastic deformation of the rope.

The rope may or may not pass through the mesh during the assembly (see Fig. 15a–c). To ensure a more uniform stress distribution, the rope should ideally not pass through the mesh (see Fig. 15d–e). The material and diameter of the wire used for sewing the net and rope are generally required to be the same as those of the original net. Sewing is usually performed mesh by mesh, and a dead knot is introduced every several meshes when assembling and sewing to ensure that the whole rope does not fall off the net in case a sewing line breaks (see Fig. 15f).



Net Structures: Design, Fig. 13 Cutting winding methods (Shi et al. 2016)



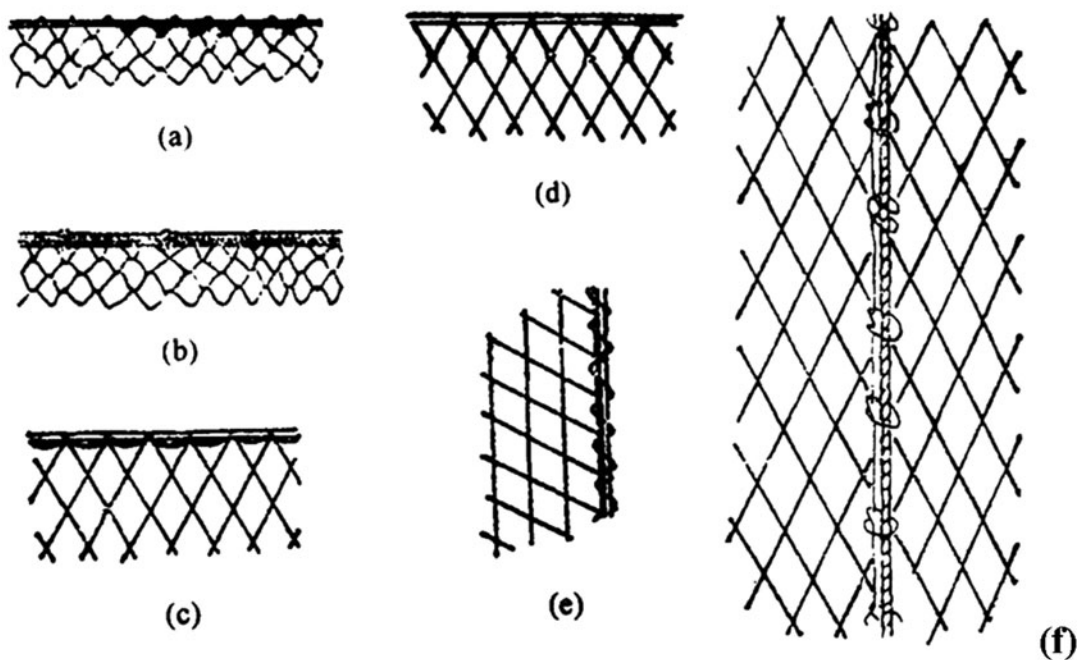
Net Structures: Design, Fig. 14 Physical map of the rope assembly (by Fukun Gui)

Final Assembly of the Net

The final assembly design is performed after the net has been assembled with the rope. Generally, the net has the main force direction. For example, in the case of a fiber net, the trend of the net system is generally the main force direction, as shown in Fig. 16. It is necessary to identify the main direction of the net load to ensure that the main direction of the load of the net twine is consistent with the main direction of the net load of the whole system during the design and final assembly.

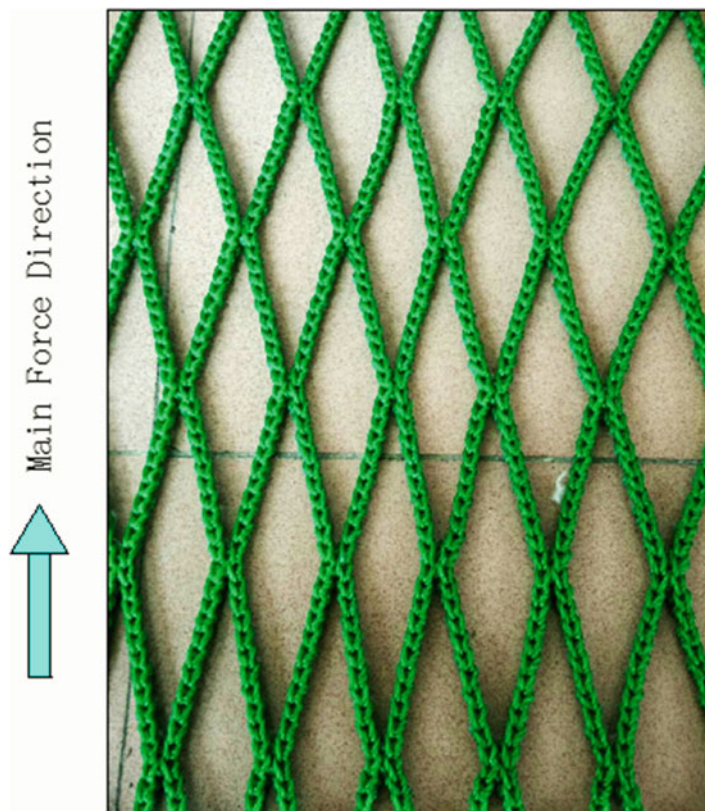
In general cage farming projects, the net is generally suspended from the cage floating frame system with the weights attached at the net

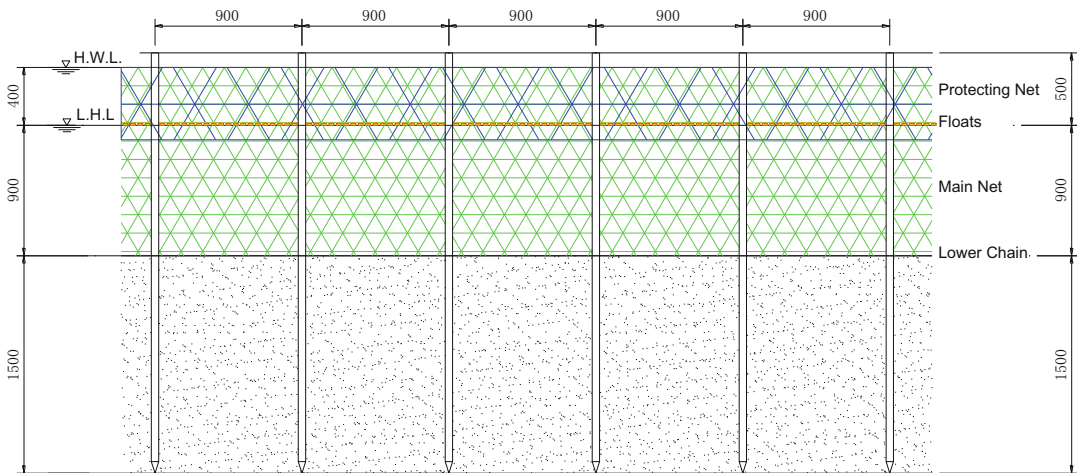
bottom. The main direction of the net load is the vertical direction, and thus, the main direction of the net line stress should be in the vertical direction. Therefore, the net is normally assembled vertically in cage aquaculture systems. For a pile-post NEA project, the stress of the net is generally borne by the pile, and the main direction of the net load is transverse. Therefore, the main direction of the stress of the net is horizontal, and the net panels need to be assembled horizontally. For sea cage aquaculture in the open ocean, the main assembly direction of the net can be determined according to the length-width ratio of the net unit. Generally, both horizontal and vertical assemblies are acceptable.



Net Structures: Design, Fig. 15 Schematic of the rope assembly (Shi et al. 2016)

Net Structures: Design,
Fig. 16 Main force
 direction of the net
 (by Fukun Gui)





Net Structures: Design, Fig. 17 Protective design of the net in a pile-post NEA project (by Fukun Gui)

Regardless of what kind of assembly method is adopted for the net, a uniform assembly should be ensured, and the stress concentration phenomenon should be strictly avoided during the net assembly. In assembly situations in which stress concentration cannot be avoided, special design and reinforcement should be implemented.

Protective Design of the Net

In addition to dynamic factors such as water flow and waves, the impact of floating objects should also be considered during net design. A protection net is mainly used to prevent the impact of floating objects such as bamboo and wood with the culturing net. Therefore, the strength of the protection net must be relatively high. Generally, PE ropes with a diameter of more than 6 mm are recommended for use, and the mesh size (2a) is expected to be more than 10 cm according to the main type of floating objects in the sea area.

For cage aquaculture, a protection net can be generally arranged along the cage periphery at a certain distance from the cage group to achieve effective protection and avoid any influence of normal management operations. For pile-post NEA projects, the protection net can be generally arranged on the peripheral piles, and the layout height of the bottom of the protection net is 1 m below the lowest tide (see Fig. 17).

Cross-References

- ▶ [Aquaculture Structures: Experimental Techniques](#)
- ▶ [Aquaculture Structures: Numerical Methods](#)
- ▶ [Modern Aquaculture Structures](#)
- ▶ [Net Structures: Biofouling and Antifouling](#)
- ▶ [Net Structures: Hydrodynamics](#)
- ▶ [Traditional Aquaculture Structures](#)

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Net Structures: Hydrodynamics

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Synonyms

Fluid dynamics of net structure; Fluid mechanics of net structure; Net hydrodynamic characteristics

Definition

Net hydrodynamics	Describes the hydrodynamic forces acting on the associated fluid mechanics around and the deformation of a net structure under external environmental loads, such as currents and waves.
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Scientific Fundamentals

Historical Development

Aquaculture is expanding all over the world, and net cages are becoming increasingly prevalent in the aquaculture industry. A thorough understanding of the hydrodynamic characteristics of a net cage is important for the design of the net cage and the welfare of the fish. An important component of a net cage is the fishing net, where the fish are kept and grown. A fishing net is completely submerged and infinitely flexible. When exposed to a current, a fishing net changes its shape to reduce the hydrodynamic force acting on it, and the deformed net in turn affects the flow field around the fishing net. In this way, the flow field and deformation of the fishing net interact and

mutually influence each other. Therefore, it is necessary to consider the hydrodynamics of the fishing net when designing a net cage.

Over the past decade, many researchers have attempted to determine the magnitude of forces caused by the fluid flow through a fishing net (Balash et al. 2009; Bi et al. 2014a; Dong et al. 2008; Fredheim 2005; Patursson 2007; Tsukrov et al. 2011; Zhao et al. 2013a). Some previous studies (Endresen et al. 2013; Lader et al. 2014) investigated the drag characteristics of net structures for fisheries and aquaculture by considering a cylindrical bar as the basic element. However, simply summing the forces acting on all the cylinders/twines of a net structure without representing the geometrical features of the intersections/knots in viscous flow yields unreliable predictions of the total drag (Fredheim 2005). As an alternative approach, a cruciform element that corresponds to one intersection of two cylinders with the length of a mesh bar can be considered as the basic element of the net structure (Bi et al. 2018).

Overall, the reduction in the current downstream of the fishing net has been considered in examining the hydrodynamic force and the effective volume of the net cage. However, in most numerical studies, only a reduction factor was estimated to represent the shielding effect of the fishing net. A numerical model that can consider the fluid–structure interaction between the flow and a flexible net has not been established. As an extension of previous works (Zhao et al. 2013a, b), a coupled fluid–structure model is developed by combining the porous-media fluid model and lumped-mass mechanical model. Using an appropriate iterative scheme, the fluid–structure interaction between the flow and fishing nets can be clarified.

Porous-Media Fluid Model

The porous-media fluid model can simulate the flow field around a plane net. Therefore, the porous-media fluid model is introduced to model a planar fishing net, and the finite volume method is used to solve the governing equations of the numerical model. A brief review of the porous-media fluid model is presented in this section.

The governing equations of the porous-media fluid model are the Navier–Stokes equations:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (1)$$

$$\begin{aligned} \frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_j} = & -\frac{\partial P}{\partial x_i} + \rho g_i \\ & + \frac{\partial}{\partial x_j}(\mu + \mu_t) \\ & \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + S_i \end{aligned} \quad (2)$$

where t is the time; μ and ρ denote the viscosity and density of the fluid, respectively; μ_t is the eddy viscosity; $P = p + (2/3)\rho k$, where p is the pressure and k is the turbulent kinetic energy; u_i and u_j are the average velocity components; g_i is the acceleration due to gravity; $i, j = 1, 2, 3$ (x, y, z); and S_i is the source term for the momentum equation.

The porous-media fluid model is a hypothetical model that produces the same water-blocking effect as the fishing net by setting the coefficients of the hypothetical porous media. For a flow through porous media, a pressure gradient exists:

$$\nabla p = a\vec{u} + b|\vec{u}|\vec{u} \quad (3)$$

where a and b are constant coefficients and $\vec{u}$ is the flow velocity. This expression was proposed by Forchheimer in 1901 based on Darcy's law. In the case of large porosities (e.g., an array of fixed cylinders), turbulence will occur, and the quadratic term for the frictional force will completely dominate over the viscous term (the linear term). In this case, the linear term is only a fitting term that has no physical meaning and can thus be neglected.

Outside the porous media, the source term for the momentum equation is defined as $S_i = 0$. Inside the porous media, S_i is calculated by the following equation:

$$S_i = -C_{ij} \frac{1}{2} \rho |\vec{u}| \vec{u} \quad (4)$$

where C_{ij} is given by

$$C_{ij} = \begin{pmatrix} C_n & 0 & 0 \\ 0 & C_t & 0 \\ 0 & 0 & C_t \end{pmatrix} \quad (5)$$

where C_{ij} represents the material matrices consisting of porous media resistance coefficients and C_n and C_t represent the normal and tangential resistance coefficients, respectively.

The drag force (F_d) and lift force (F_l) acting on the flow through the porous media can be expressed as follows:

$$\vec{F}_d = C_n \frac{1}{2} \rho \lambda A |\vec{u}| \vec{u} \quad (6)$$

$$\vec{F}_l = C_t \frac{1}{2} \rho \lambda A |\vec{u}| \vec{u} \quad (7)$$

where λ and A represent the thickness and area of the porous media, respectively.

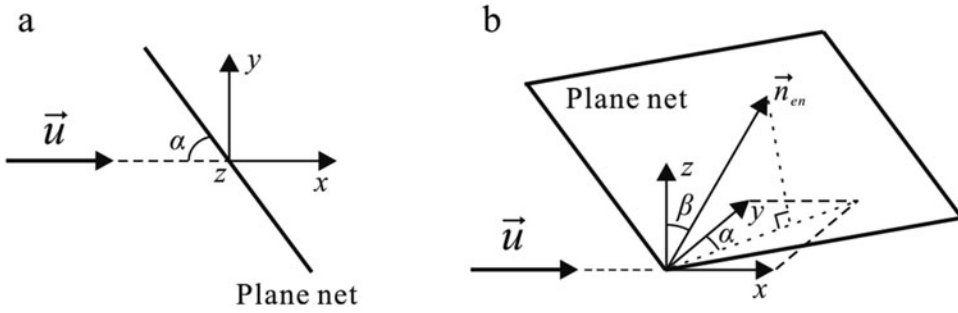
The porous coefficients in Eqs. (6) and (7) can be calculated from the drag and lift forces. Moreover, the drag and lift forces acting on a fishing net are strongly correlated with the features of the net. Therefore, the connection between the net features and the flow field is established using the porous coefficients in the numerical model. Generally, the drag and lift forces of the plane net can be obtained from laboratory experiments or calculated using the Morison equation:

$$F_d = \frac{1}{2} \rho C_d A u^2 \quad (8)$$

$$F_l = \frac{1}{2} \rho C_l A u^2 \quad (9)$$

where C_d and C_l are coefficients that can be calculated using empirical formulas (Bi et al. 2014b).

The porous coefficient C_n can be calculated from a curve fit between the drag forces acting on a plane net and corresponding current velocities using the least squares method when the plane net is oriented normal to the flow. The other coefficient C_t can be obtained when a plane net is oriented at a certain attack angle (see Fig. 1a). In this case, the coefficient C_t should be transformed to $C_{t\alpha}$ using Eq. (10) (Bear 1972). Next, $C_{t\alpha}$ can be



Net Structures: Hydrodynamics, Fig. 1 Definition of the angles α and β : plane nets (a) in the horizontal plane and (b) at an arbitrary position (Bi et al. 2014b)

calculated from a curve fit between the lift forces acting on a plane net and corresponding current velocities using the least squares method.

$$C_{tx} = \frac{C_n - C_t}{2} \sin(2\alpha) \quad (10)$$

where α is the attack angle, defined as the angle between the flow direction and plane net in the horizontal plane, and C_{tx} is the tangential inertial resistance coefficient for the plane net at an attack angle α .

When a plane net is oriented at an arbitrary position (see Fig. 1b), the porous coefficients should be transformed using Eq. (11):

$$\begin{pmatrix} C'_n \\ C'_t \end{pmatrix} = \begin{pmatrix} \sin^2 \alpha \sin^2 \beta & \cos^2 \alpha & \sin^2 \alpha \cos^2 \beta \\ \frac{1}{2} \sin 2\alpha \sin^2 \beta & -\frac{1}{2} \sin 2\alpha & \frac{1}{2} \sin 2\alpha \cos^2 \beta \\ \frac{1}{2} \sin 2\beta & 0 & -\frac{1}{2} \sin 2\beta \end{pmatrix} \begin{pmatrix} C_n \\ C_t \end{pmatrix} \quad (11)$$

where β is the angle between the normal vector of the plane net ($\vec{n}_{en}$) and z-direction. The cosines of α and β can be calculated between the corresponding two vectors.

Lumped-Mass Mechanical Model

The lumped-mass mechanical model is introduced to simplify the fishing net, and the motion

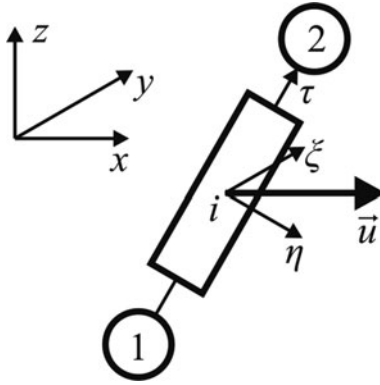
equations can be established mainly based on Newton's second law. Given an initial net configuration, the motion equations can be solved numerically for each lumped-mass point. Finally, the configuration of the net in current can be simulated. The calculation method for the model has been explained in previous studies (Li et al. 2006; Zhao et al. 2007b), and thus, only a brief outline is provided in this section.

Under a uniform current, the motion equations of lumped-mass point i can be expressed as follows:

$$(M_i + \Delta M_i) \vec{a} = \sum_{j=1}^n \vec{T}_{ij} + \vec{F}_d + \vec{B} + \vec{W} \quad (12)$$

where M_i and ΔM_i are the mass and added mass of lumped-mass point i , respectively; $\vec{a}$ is the acceleration of the point; $\vec{T}_{ij}$ is the tension force in twine ij (j is the index for the knot at another end of the bar ij); n is the number of knots adjacent to point i ; and $\vec{F}_d$, $\vec{W}$, and $\vec{B}$ denote the drag force, gravity force, and buoyancy force, respectively.

To consider the direction of the hydrodynamic forces acting on mesh bars, local coordinates (τ , η , ξ) are defined to simplify the procedure (see Fig. 2). The origin of the local coordinate system is set at the center of a mesh bar, and the η -axis lies on the plane, including the τ -axis and flow velocity $\vec{u}$. The drag force on a lumped-mass point in the τ -direction can be obtained from the Morison equation as follows:



Net Structures: Hydrodynamics, Fig. 2 Local coordinate system for a mesh twine (Bi et al. 2014a)

$$F_{D\tau} = -\frac{1}{2} \rho C_{D\tau} A_\tau \left| \vec{V}_\tau - \vec{R}_\tau \right| \left(\vec{V}_\tau - \vec{R}_\tau \right) \quad (13)$$

where $C_{D\tau}$ is the drag coefficient in the τ -direction, A_τ is the projected area of the twine normal to the τ -direction, and $\vec{R}_\tau$ and $\vec{V}_\tau$ are the velocity components of the lumped-mass point and water particle in the τ -direction, respectively. Similar expressions can be applied to the other two components of the drag force, $F_{D\eta}$ and $F_{D\xi}$.

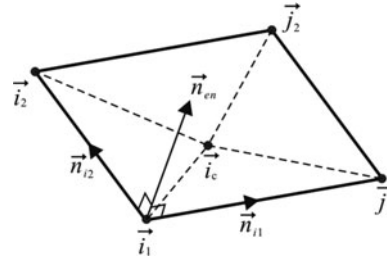
The mesh-grouping method (Tsukrov et al. 2003; Zhao et al. 2007a) is used to reduce the computational effort of the net. Each grouped mesh is defined as a plane-net element (see Fig. 3). The normal vector of the plane-net element ($\vec{n}_{en}$) is calculated as follows:

$$\vec{n}_{en} = \frac{\vec{n}_{i1} \times \vec{n}_{i2}}{|\vec{n}_{i1} \times \vec{n}_{i2}|} \quad (14)$$

where $\vec{n}_{i1}$ and $\vec{n}_{i2}$ are given by

$$\vec{n}_{i1} = \frac{\vec{j}_1 - \vec{i}_1}{|\vec{j}_1 - \vec{i}_1|}, \quad \vec{n}_{i2} = \frac{\vec{i}_2 - \vec{i}_1}{|\vec{i}_2 - \vec{i}_1|} \quad (15)$$

Because of the deformation of the net, the area of a plane-net element is not constant, which influences the solidity of the plane-net element. Consequently, the solidity S_n must be corrected



Net Structures: Hydrodynamics, Fig. 3 Schematic of a plane-net element (Bi et al. 2014b)

based on the varying area of the plane-net element:

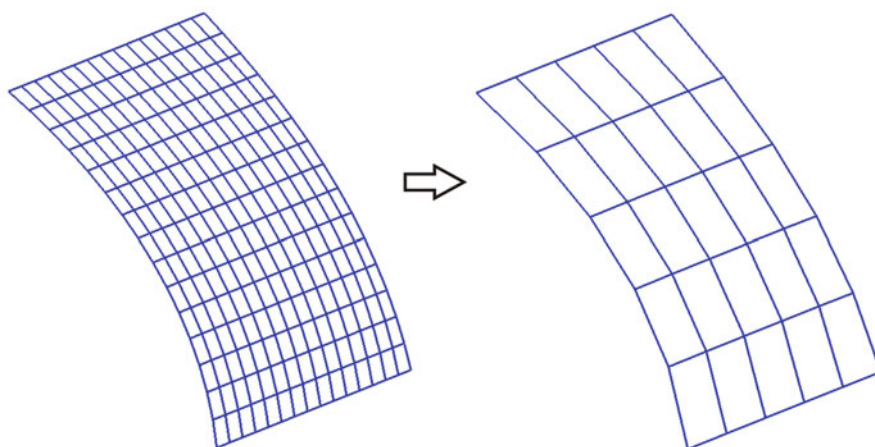
$$S'_n = \frac{A}{A'} S_n \quad (16)$$

where S_n and A denote the solidity and area of the undeformed plane-net element, respectively, and S'_n and A' are the corrected solidity and area of the plane-net element, respectively. To calculate the area of the plane-net element, the four-sided element is divided into four triangles, as shown in Fig. 3. The area of the element can be obtained by the sum of the four triangles.

Coupled Fluid–Structure Interaction Model

Based on the above work, the interaction between the flow and fishing net can be simulated by dividing the net into many plane-net elements (see Fig. 4). The main concept of the numerical approach is to combine the porous-media fluid model and lumped-mass mechanical model to simulate the interaction between the flow and flexible nets (Bi et al. 2014a, b). Considering the example of a single flexible net, the calculation flow chart of the numerical approach is shown in Fig. 5. The details of the calculation procedure are as follows.

The calculation procedure includes four steps. Step 1 involves the simulation of the flow field around a vertical plane net without considering the net deformation using the porous-media fluid model. Step 2 involves the determination of the drag force and the configuration of a flexible net in a current using the lumped-mass mechanical model. The given initial flow velocity is the



Net Structures: Hydrodynamics, Fig. 4 Schematic of the net model

average value exported from the upstream surface of the plane net in the last step. In Step 3, according to the configuration of the net, the orientation and the porous coefficients of each plane-net element can be obtained, thereby determining the flow field around the fishing net. In Step 4, the drag force and configuration of the net are calculated again. The flow velocity acting on each lumped-mass point is the average value exported from the upstream surface of the corresponding plane-net element in the last step.

The terminal criterion of the calculation procedure is defined as $\Delta F = (F_{i+1} - F_i)/F_{i+1}$, where F_i and F_{i+1} are the drag forces of two adjacent calculations. If $\Delta F < 0.001$, the force acting on the fishing net is stable, and the configuration of the net is the equilibrium position considering the fluid–structure interaction. Thus, the calculation procedure is considered complete. Otherwise, the next iteration of the procedure is implemented, and Steps 3 and 4 are repeated until the criterion meets the precision requirement. Finally, the fluid–structure interaction problem can be solved, and the steady flow field around the fishing net can be obtained (see Fig. 6).

Other Related Studies

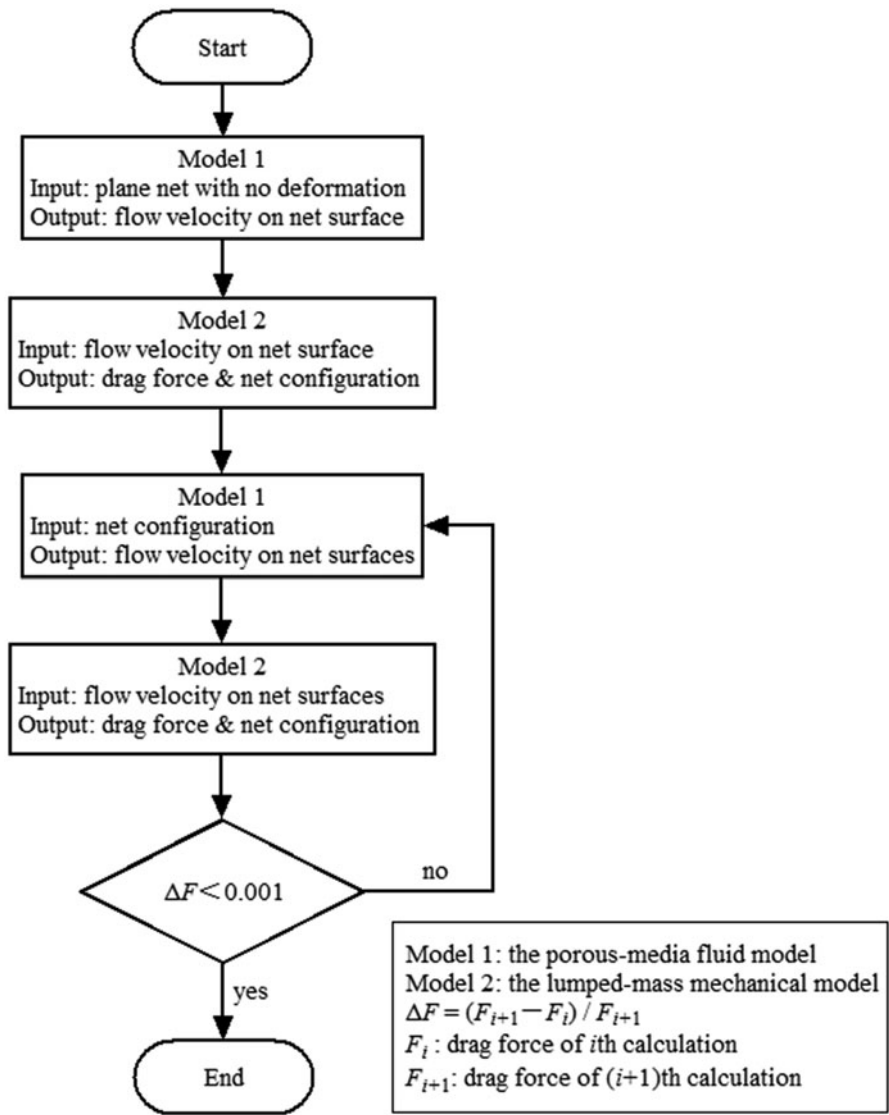
Hydrodynamics of the Net Element

To analyze the drag force on and the flow pattern around a cruciform element of the net structure, Bi

et al. (2017a) used computational fluid dynamics (CFD) to examine the drag effects and flow around T0 and T45 meshes oriented at angles of attack (AOAs) ranging from 90° to 0° (see Fig. 7). Two models were considered: (i) a one-cruciform element and (ii) a four-cruciform assembly composed of one element subjected to the wake effect of the upstream elements. The fluid flow around and drag of two cylindrical cruciform patterns, namely, a conventional (T0) pattern and a pattern rotated 45° in its own plane (T45), were numerically investigated by solving the three-dimensional Reynolds-averaged Navier–Stokes equations within the subcritical flow regime over AOAs ranging from 90° to 0° . First, the drag for the one-cruciform element was assessed. Next, the four-cruciform assembly (“square mesh”) was analyzed to consider the wake effect of tandem elements (see Fig. 8).

Effects of Biofouling on the Net Hydrodynamics

Biofouling is an inevitable problem in marine aquaculture that adversely influences the hydrodynamic behavior of the aquaculture facility (Bloecher et al. 2013; Gansel et al. 2017; Lader et al. 2015). In the study of Bi et al. (2018), the dominant biofouling on the netting of fish farms in the Yellow Sea of China was determined by field sampling (see Fig. 9). Plane nets with various levels of biofouling were obtained by submerging nets at different depths and for different durations.

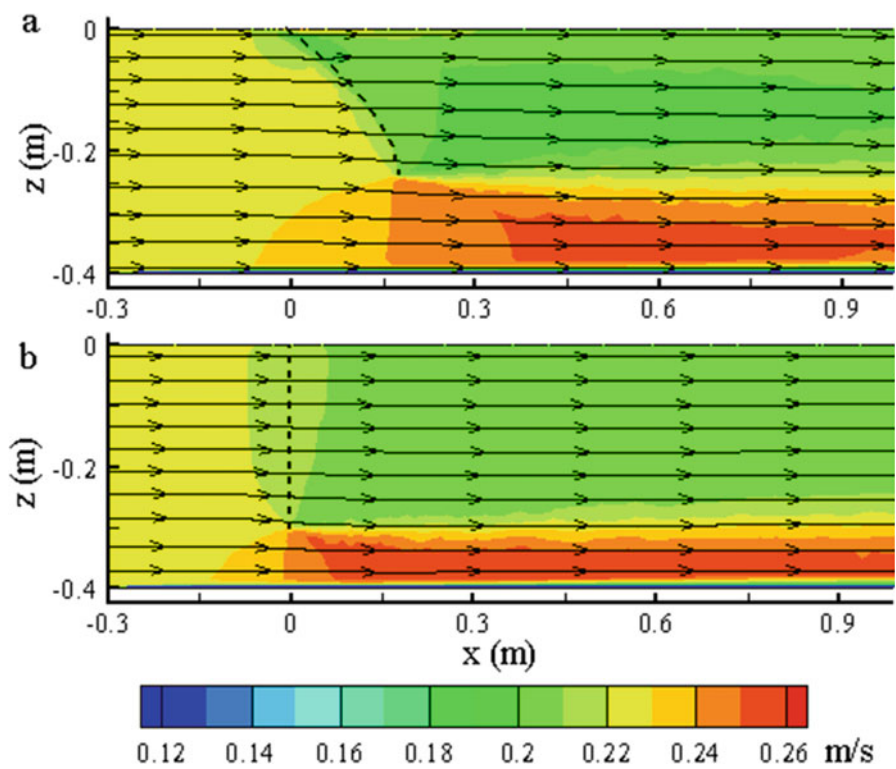


Net Structures: Hydrodynamics, Fig. 5 Flow chart of the coupled fluid–structure interaction model. (Modified from Bi et al. 2014a)

The drag acting on biofouled nets in currents was investigated by conducting laboratory experiments. The results indicated that the drag acting on the net increases with increasing levels of biofouling. Compared to a clean net, the accumulation of biofouling can lead to more than 10 times the hydrodynamic load on the net (see Fig. 10).

The effect of biofouling on the hydrodynamic characteristics of the net cage is of particular interest, as biofouled nets can significantly reduce

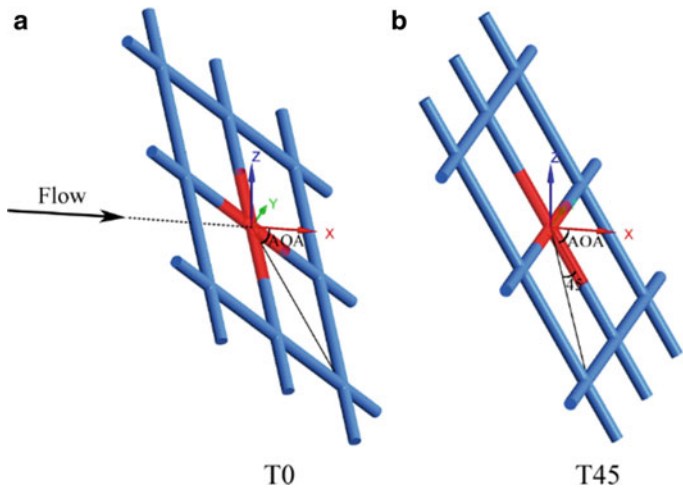
the flow of well-oxygenated water reaching the stocked fish. The drag forces on and the flow fields around five plane nets with different levels of biofouling are calculated using the proposed model (Bi et al. 2015). Flow-through full-scale net cages with the same level of biofouling as the tested plane nets are modeled. The flow fields inside and around biofouled net cages are analyzed, and the drag force acting on a net cage is estimated using a control volume analysis



Net Structures: Hydrodynamics, Fig. 6 Flow-velocity distribution on the vertical plane $y = 0$ around the net calculated by performing a numerical simulation for the incoming velocity of 0.226 m/s: (a) considering the FSI between the flow and fishing net and (b) without considering the FSI (Bi et al. 2014a)

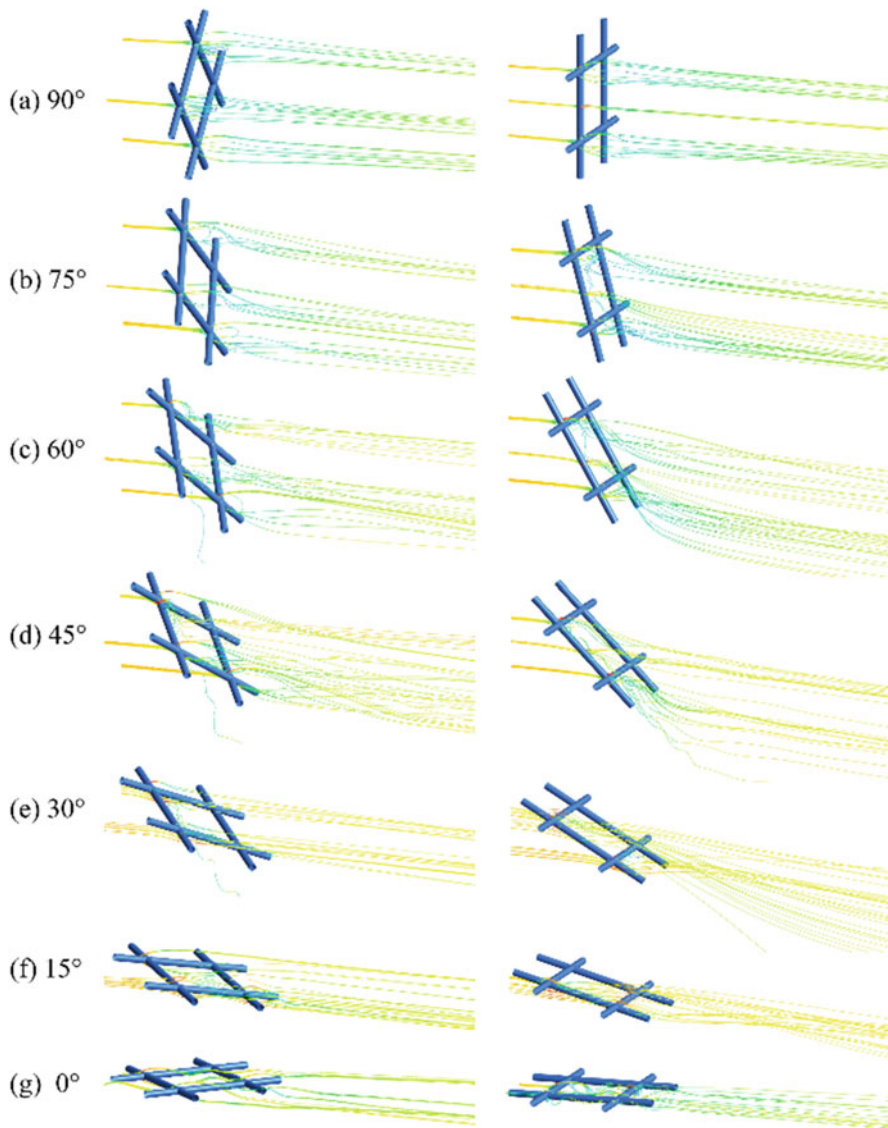
N

Net Structures: Hydrodynamics, Fig. 7 Definitions of the (a) angle of attack (AOA) and (b) T0 and T45 mesh orientations. The mesh orientation angle is the angle between the mesh axis and directional cosine of the flow vector on the fishing net



method. According to the numerical results, the empirical formulas of the reduction in the flow velocity and the load on the net cage are derived as a function of the drag coefficient of the corresponding biofouled netting.

To analyze the wave attenuation, the wave elevation both inside and around the cage arrays is numerically analyzed considering the effect of biofouling. According to the results of the present study, biofouling considerably influences the



Net Structures: Hydrodynamics, Fig. 8 Streamlines around T0 (left) and T45 (right) four-cruciform assemblies for discrete AOAs (Bi et al. 2017a)

wave elevation, and the damping effect of the cage array increases with increasing levels of biofouling. In addition, the incident angle of the waves has a significant effect on the wave field inside and around the cage array. From a wider perspective, the results of this study have implications for the transport of dissolved nutrients and hydrodynamic performance of the integrated multitrophic aquaculture configuration (Bi et al. 2017b).

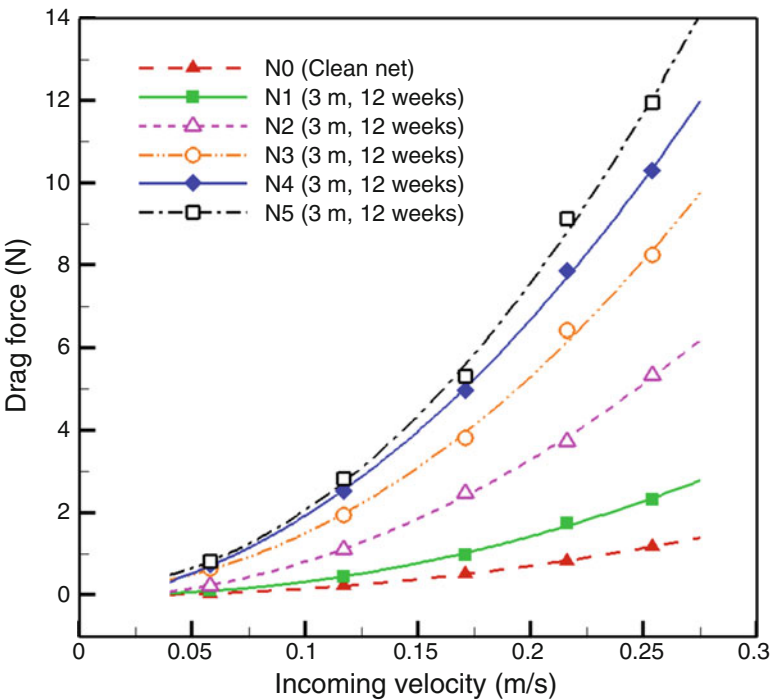
Effects of Fish on the Net Hydrodynamics

A computational fluid dynamics study is performed to analyze the impact of cultured fish on the flow field through a net cage and the deformation of the net cage (Tang et al. 2017). The shear stress turbulent k- ω model is applied to simulate the flow field through the net cage, and the large deformation nonlinear structure model is adopted to conduct the structural analysis of the



Net Structures: Hydrodynamics, Fig. 9 Images of the biofouled nettings of net cages in a fish farm (Bi and Xu 2018)

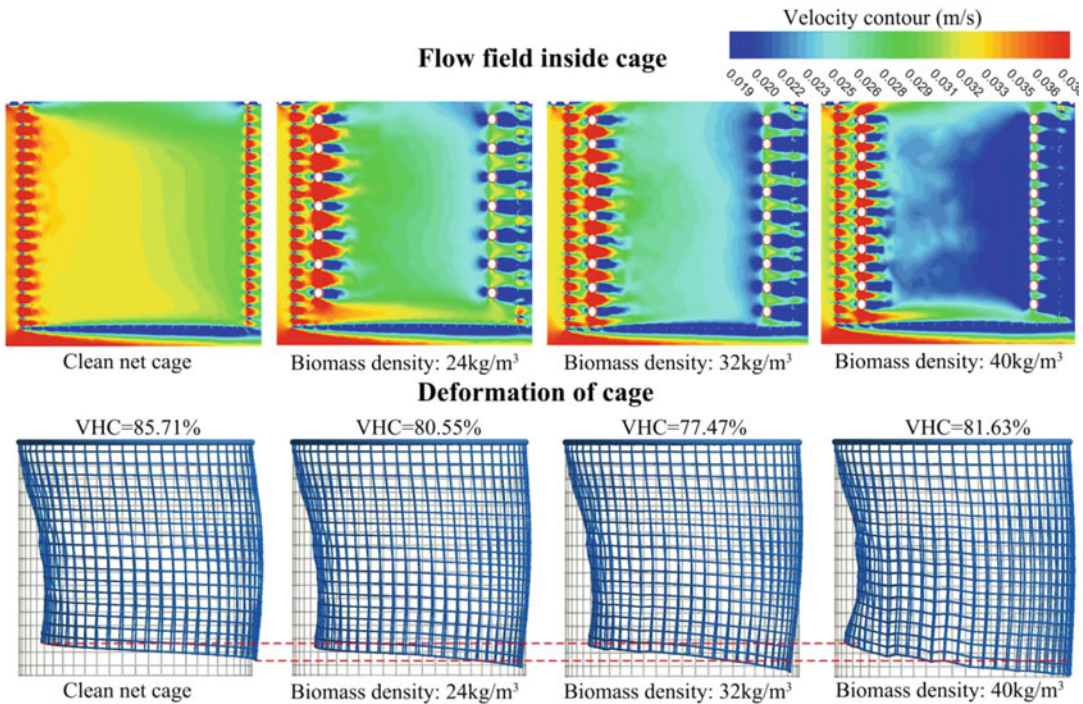
Net Structures: Hydrodynamics, Fig. 10 Drag forces on nets with different levels of biofouling for various incoming velocities. Nets N0 to N5 represent the various levels of biofouling (Bi et al. 2018)



flexible net cage. A fish model is used to simulate the effect of the fish behavior on the flow pattern around the net cage and deformation of the net cage (see Fig. 11). In addition, the flow fields around the net cage in the current are investigated considering different fish group structures, fish swimming speeds, fish distributions, and fish stocking densities. The results indicate that the circular movement of fish in still water leads to a low-pressure zone at the center of the net cage, which causes a strong vertical flow along the centerline of the net cage. The drag force on the net cage significantly decreases as the fish stocking density increases; however, the most severe

deformation of the net cage occurs in the case of a medium fish stocking density.

Notably, a net cage is a type of floating structure anchored to the seafloor by mooring lines. The change in the flow field inside and around the net cage owing to swimming fish may further affect the hydrodynamic forces acting on the net cage (He et al. 2018). Recently, the effects of farmed fish on both the drag force acting on and the flow patterns inside and around the net cage were investigated experimentally by Bi et al. (2020). The laboratory experiments were conducted using a circular net-cage model with juvenile carp (*Cyprinus carpio*). The considered



Net Structures: Hydrodynamics, Fig. 11 Flow field around the net cage and net deformation in the case of different fish densities (Tang et al. 2017)

stocking density of the net cage ranged from 5.3 to 37.2 kg/m³ depending on the number of fish. Here, four net-cage configurations are designed for a laboratory experiment. The drag force acting on the net cage and flow velocities and the turbulence characteristics inside and downstream of the net cage are investigated. The results indicate that the biomass, flow speed, net-cage configuration, and net deformation have various effects on the drag force and fluid flow. Moreover, the potential influence of the fish size, swimming pattern of the fish school, and fish-net contact are also discussed.

The swimming pattern of the fish school is a key factor that influences the water inside and around the net cage. Fish schools swimming in an on-current pattern, similar to that in the present study, mainly change the flow pattern downstream of the net cage. However, certain fish tend to form a torus-shaped school and swim in a circular pattern (see Fig. 12). To maintain a circular pattern, the fish must accelerate toward the center of the net cage. This type of centripetal force is balanced

by the water volume outside the net cage, resulting in a change in the flow direction.

Key Applications

Global fish production has grown steadily in the last five decades, with the food fish supply increasing at an average annual rate of 3.2% and an increasing number of people relying on fisheries and aquaculture for food and as a source of income. The prediction tools for net hydrodynamics can be applied to enhance the energy efficiency of trawling and optimize the design of floating fish cages. Net hydrodynamics, such as the drag force acting on a net structure and the associated flow field, are of considerable interest over a wide range of practical applications, such as heat exchangers, damping screens, and various onshore and offshore structures.

The related outputs have broad application prospects in the development and utilization of



Net Structures: Hydrodynamics, Fig. 12 Circular swimming pattern of *Takifugu rubripes* in a square net cage. (Photo courtesy of Sheng-Cong Liu of Dalian Tianzheng Industrial Co., Ltd)

marine resources. This study can provide a new theoretical basis for the research, development, and design and safe operation of offshore aquaculture facilities, trawl nets, floating breakwaters, jacket platforms, and trash racks for coastal nuclear power plants, including net-structure components.

Cross-References

- [Aquaculture Structures: Experimental Techniques](#)
- [Aquaculture Structures: Numerical Methods](#)
- [Modern Aquaculture Structures](#)
- [Net Structures: Biofouling and Antifouling](#)
- [Net Structures: Design](#)
- [Traditional Aquaculture Structures](#)

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Netpen Liver Disease (NLD)

► Net Structures: Biofouling and Antifouling

Network Experiment

► Underwater Acoustic Sensor Network

Neural Network Control

► Intelligent Control Algorithms in Underwater Vehicles

Neutral Float (NF)

► Profiling Float

New Technologies in Auxiliary Propulsions

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Synonyms

Auxiliary propulsions; Biofouling; Lightweight construction; Machinery maintenance; Optimization of propellers; Ship energy efficiency; Waste recovery system; Wind propulsions

Definition

Shipping is a substantial emitter that accounts for about 3% of total global CO₂, 1–3% of SO_x, and more than 10% of NO_x (Bauhaug et al. 2009). Because of the strong growth of maritime transport sector in world trade, emissions from shipping are expected to increase 150–200% by 2050 (EC 2011). The effect of air emission of NO_x and SO_x from shipping on the environmental and health impact is widely realized and attracts a lot of critics from social public. The International Maritime Organization (IMO) has established regulations to reduce fuel consumption and air emission (EEDI, EEOI, SEEMP, ECA, etc. IMO (2013, 2014, 2016)). The inclusion of emission tax in the future bunker price will be likely implemented in the future shipping market (DNV 2011). According to the future fuel cost scenarios for the reduction of CO₂ and SO_x emissions by Eide et al. (2011) and DNV-GL (2014), the added premiums may push the bunker price up to 2000 USD/ton in 2040. Due to the increasing pressure on the uncertain bunker prices, stricter regulations on air emissions from shipping, as well as ambitious market plans to strengthen the branding, trade flexibly, and attract external stakeholders, the current shipping industry is putting great efforts and investment on technologies to improve ship energy efficiency and reduce fuel consumption and air emissions.

Overview of Technologies to Improve a Ship's Energy Efficiency

Many innovative technologies potentially aiming at reducing the fuel consumption and increasing a ship's energy efficiency have been investigated by various stakeholders and policy makers in the shipping industry, e.g., Bauhaug et al. (2009), Wärtsilä (2010), ABS (2015), DNV-GL (2014), etc. A detailed summary of those technologies is provided in Table 1, which categorize the technologies for new building and existing ships. It should be noted that simple summation of potential savings from all the listed technologies will greatly overestimate the benefits of

implementing multiple energy-efficient technologies for a given ship (Kesicki and Strachan 2011 and Kesicki and Ekin 2012). For example, considering the characteristics of operational profiles and structural configurations of ships in different segments, some of those technologies may be only efficient to certain types of ships. In addition, some technologies are mutually exclusive. According to an extensive market survey by DNV-GL (2014), shipping companies are more willing to implement mature and well-known technologies, and the most attractive technologies are those of little investment and easily realized benefits, for example, the slow-steaming, weather routing, energy performance monitoring system and voyage optimization, etc. It also indicates that for the current shipping companies, the biggest incentive to implement energy-efficient technologies is to be compliant with the regulation requirements, rather than the market-focused goals, such as strengthen market position and economic profits, etc. Part of the reason is that there is no transparent approach to evaluate the actual energy savings by implementing current technologies, while the indicated benefits are mainly claimed by technology providers' or manufacturers' data and analysis.

In general, new building vessels have already been optimized by ship design and construction companies to allow for the best operation performance. For the existing ships, any changes and installation of those energy efficiency technologies are often needed to be negotiated with shipyards and/or technology service providers and classification societies. These installations are often associated with quite large amount of investment cost to ship owners, while the energy savings due to the implementation of innovative energy efficiency technologies will eventually benefit most to the ship chartering companies. Most often, ship owners who pay for the extra installations might not benefit from the fuel saving made by these installed technologies. As is explained, some interest conflicts are often present regarding the cost of technology investments between different stakeholders of the shipping industry. These conflicts will also affect the investment wiliness for certain stakeholder to put

New Technologies in Auxiliary Propulsions, Table 1 Various energy efficiency measures used in the current shipping industry

Concept categorizes		Detailed measures	Potential savings (%)	Barriers
New building vessels	Hull form optimization and propeller configuration	Main dimension optimization, e.g., slender and larger ships, and less ballast water volume	4–9	Policy and regulation
		Ship energy system optimization based on actual planned operational profiles, fuel price, trade routes, loading conditions, etc.	3–5	Information and organization
		Optimization of stern bigle, rudder and propeller for inflow considerations (e.g., better skeg design, better propeller blade design)	3–4	Information, economic, organizational, and technical
		Optimization of hull shape and bulbous bow	2–5	Information and technical
		Better design with consideration of added resistance due to wind and waves in open sea	1–2	Technical and economic
		Influence from IMO EEDI on ship design	10–20	Policy barriers, organization barriers
	Light weight construction	High strength steel	1–3	Technical
		Composite material	2–7	Technical and economic
Existing vessels	Machinery system	Waste heat recovery system	5–10	Economic, organization, and information
		Alternative fuel for power	2–4	
		Hybrid auxiliary power generation	2	
		Optimum heating and cooling systems	1–2	
		Adaptive pump and power management systems	5	
	Energy-saving devices	Propulsion improving devices	3–6	Technical and information
		Skin friction reduction	3–5	Technical and economic
		Wind power and other renewable devices for auxiliary propulsion	5–20	Technical, economic, and organization
	Ship operation management and optimization	Ship speed reduction	20–25	Organization
		Optimized ship maintenance schedule, hull and propeller cleaning	5	Information and economic
		Autopilot adjustment	4	Technical
		Intelligent engine adjustment according to weather and loading conditions	5	Technical and economic
		Trim, ballast and rudder control optimization, air lubrication	2–8	Technical and information
		Weather routing and voyage optimization	5–10	Information and technical

^aIt should be noted that the indicated energy savings from the listed technologies can differ significantly from various analysis approaches; therefore the intention of the table is to give a basic reference and relative saving potentials for different technologies

on the development and implementation of innovative energy efficiency technologies. Other barriers associated with the research and innovation of energy efficiency technologies will be further described in the next section. In the following, the definition and ideas behind various technologies listed in Table 1 will be briefly presented. More focus will be put on the technologies using wind power as auxiliary propulsion resources because of their big potential recognized in the shipping market.

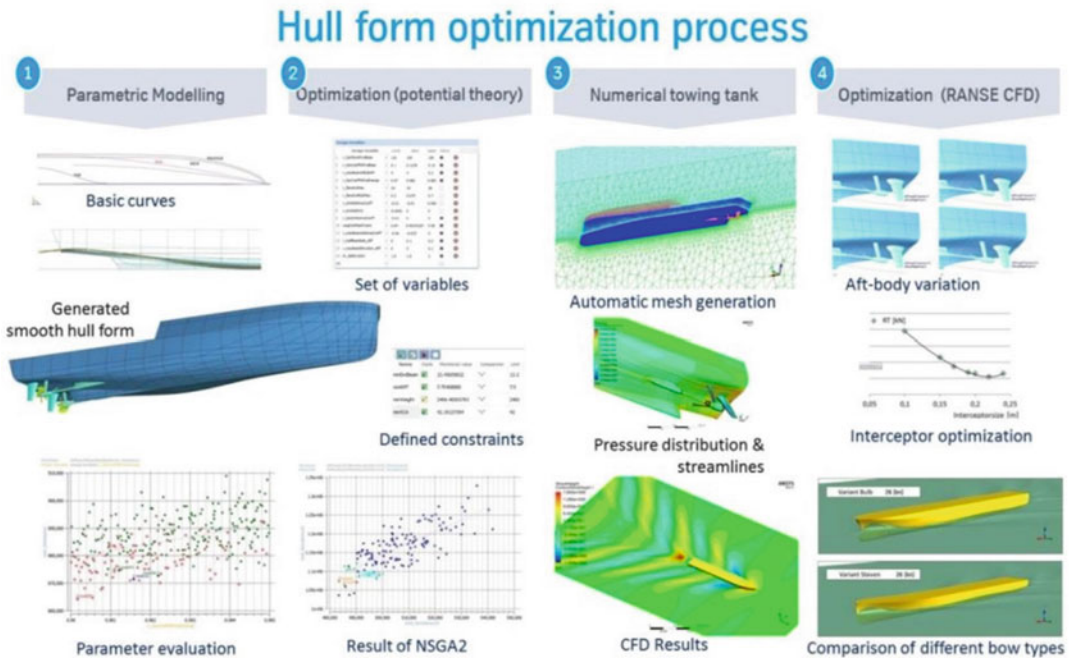
Energy Efficiency Technologies from Ship Design/Construction Perspectives

For new building ships, the first measure to increase her energy efficiency is to optimize a ship's hull to minimize the ship resistance considering her life cycle service conditions. Often, a new ship construction starts with a standard hull form and propulsion system (that is often well optimized) provided by a shipyard. Then the hull form could be modified and further optimized to fulfill the ship owner's specific expectation on the ship's operational profile. The basis for a specific optimization is the ship's main particulars, such as ship dimensions and service speed. After a ship's main dimensions and operational conditions have been defined, more detailed hull optimization can be carried out from two different aspects, i.e., reduce resistance propulsion and increase propulsion efficiency. The hull optimization can be performed by designing better hull lines in particular for the design of forebody, afterbody, and appendages. The forebody optimization often needs to consider a ship's bulb design, waterline entrance, forward shoulder, and transition to the turn of the bilge. The afterbody optimization aims to reduce the added resistance due to the aft shoulder crest wave and the deep wave trough waves. It can also help to optimize the fluid flow entering the ship propeller and to avoid eddy effects, which can be used to improve the ship's propulsive efficiency. In addition, for certain types of ships of big superstructures, the wind resistances also play an important role in the hull optimization process.

Conventional hull optimization is often carried out for calm water sailing conditions, but an oceangoing ship seldom encounters sea conditions of no wind and waves during her service life. Currently, the computation capability has been increased dramatically; various computational hydrodynamic codes of different levels of difficulties have been developed to simulate/approximate a ship's energy performance in various operational and environmental conditions. With the computer aid for ship design, a typical new ship design and optimization process can involve four steps of analysis, from the basic empirical formula to the most advanced CFD (computational fluid dynamics) analysis as in the following Fig. 1 (Friendship 2018).

Optimization of Propeller Configuration to Increase Propulsion Efficiency

In addition to reduce a ship's resistance, the ship's propulsion system can also be optimized to increase her propulsion efficiency by configuring more proper propeller design suitable to the ship's actual operational conditions. For example, a conventional fixed pitch propeller is optimized to a ship's design speed and loading conditions when operated in calm water conditions with a specific engine RPM. Due to a ship's encountered sea environments, consideration of expected time of arrival, cargo safety, etc., a ship's speeds often vary a lot during her whole voyage. Therefore, the propeller is not always running on its design condition with the most optimum propulsive efficiency. To allow a ship's propeller to run their optimum setting for various operational conditions, a variable-pitch propeller (VPP) can be propulsive efficient for a wide range of engine RPM speeds for various encountering sea environment and load conditions. This is because the pitches of the propeller blades can be adjusted to transmit the highest effective power from a marine engine. For example, a ship requires more effective propulsive power to push the ship forward when she is fully loaded and encountering heavy sea conditions than she is sailing in light conditions (ballast in calm sea conditions).



New Technologies in Auxiliary Propulsions, Fig. 1 A typical hull form optimization process from Friendship 2017

Therefore, higher ship energy efficiency can be achieved by setting a ship's propeller blades to the optimal pitch. In general, the cost of a VPP is much more expensive than the FPP, but a ship installed with a VPP can accelerate, stop, and back faster, and it can also improve a ship's maneuverability by controlling the fluid flow onto the ship's rudder (Kasten 2018) (Fig. 2).

In order to increase a ship's propulsion efficiency, a lot of innovative ideas have been developed to change the fluid flow sucking into the propeller. For example, a device named as the inventor Grothues-Spork (1988) called Grothue's spoilers contains a set of fins installed on a ship's afterbody hull ahead of the propeller. The finds can help to straighten the flow right so that the flow could be redirected horizontally toward propeller (Fig. 3), and they may also help to avoid the flow separation, leading to an improved propulsion efficiency. Another similar idea is to put a big extra flow duct ahead of the propeller (Fig. 3 shows an example of the well-recognized Becker Mewis Duct for ship propulsion systems); it could also help to organize the water flow

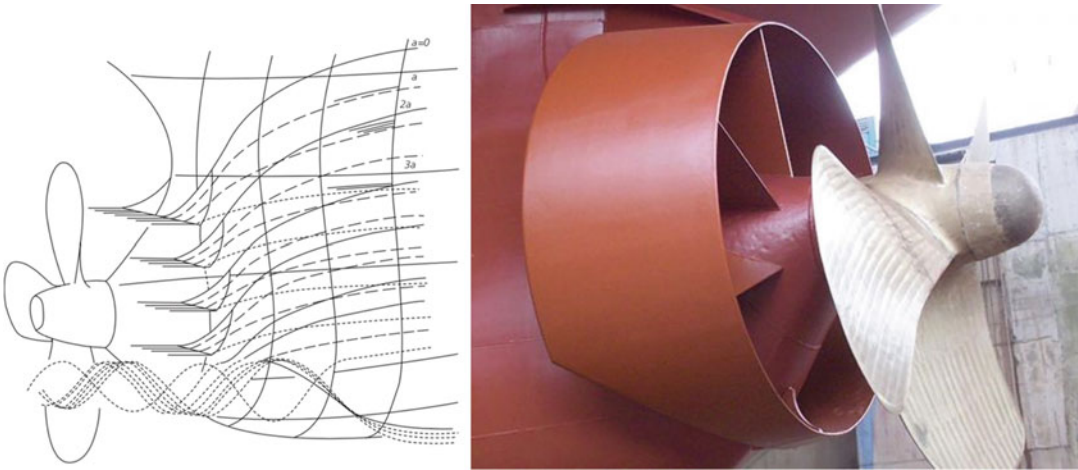
entering toward the propeller. It has the potential to save up to 8% of fuel cost for certain ship types. Many other innovative ideas have been proposed to increase the propeller's work efficiency, from both the shape of propeller blade design, configuration of rudders, etc. A good summary can be referred to ABS (2015).

Lightweight Materials in Ship Constructions

From a ship's structural integrity perspectives, a ship should be constructed by enough materials to provide strength to overtake the forces caused by cargoes and environmental loads. Normally, the larger a ship is to be built, the more cargoes it is supposed to take, the more materials/steels (more lightweight) it will need to use to provide enough ship strength, and consequently, the more fuel is needed to consume for each voyage. One way to reduce a ship's lightweight, which will help to reduce the fuel consumption per transported cargo tonnage, is more lightweight



New Technologies in Auxiliary Propulsions, Fig. 2 A fixed pitch propeller (FPP) to the left and a variable-pitch propeller (VPP) to the right installed in ships



New Technologies in Auxiliary Propulsions, Fig. 3 Basic ideas of the Grothue's fins work principles (ABS 2015) and an example of a Becker Mewis Duct mounted ahead of a ship's propeller (Becker Mewis 2018)

materials such as composite materials, or high tensile steels that can provide the same strength as mild steel but needs less materials. As the increase of ship size, more and more percentages of HTS have been realized in today's ship construction. For large bulk carriers, the average of HTS usage has reached to 60%, while for

the large containers, it is about 55%, a bit less for the tankers. The usage of high tensile steel can save a ship's lightweight up to 12%.

Another big potential for a ship's lightweight reduction is to use the composite materials such as the fiber-reinforced plastic (FRP) or fiber glasses, which are actually widely used in small high-

speed craft and in the superstructures of certain ro-ro/ropax ships. The application of FRP for ship construction can help to reduce a ship's weight by 30–70%, contributing to significant fuel savings. However, the application of composite materials for a wide range of conventional ship construction is not so optimistic, due to construction cost, recycling cost, and fire safety in compared with the steel.

Energy Efficiency Technologies from Ship Operation Perspectives

In comparison with the new building ships, which have already been quite well optimized to follow the environmental regulations on energy efficiency, it is rather an urgent demand to develop energy efficiency technologies for existing ships. In addition to install some extra equipment as the previous subsection to increase a ship's propulsion efficiency, more straightforward and efficient way is to develop technologies or management/maintenance strategies to assist a ship's operation.

Machinery Maintenance and Update Plan

A ship's fuel is directly consumed by a ship's complex machinery system, in which a lot of improvement could be achieved by better planning, optimized control, and configuration setting. For example, in addition to the energy used to provide propulsion force to move a ship forward, a lot of energy onboard a ship is actually used to provide electric power, such as heating the oil tank for chemical ships, powering the air conditions to containership and passenger ships, etc. The electric power onboard ships can be provided by either a generator attached to the main propulsion engine or by generators driven by independent diesel engines. To have an optimum setting/selection of engine generator, careful consideration should be made toward a ship's actual operational conditions and engine power usage. Normally, for ships with constant electric power requirements, i.e., during both sailing and anchor conditions, an auxiliary diesel engine generator is a more suitable option. While sometimes, the

mixture usage of diesel engine generator and generator mounted on a ship's main engine can provide even higher efficient operation through an optimal power management plan.

There are often many valves/pumps in a ship's machinery systems to control the bunker flow into the ship's main engine. The pump systems are often controlled by electronically controlled engines. In the shipping market, some companies also use the computer-aided management system to automatically adjust the work frequency of the pumps according to specific loading and operational and environmental conditions. Those electronically controlled engines and turbo-charger control equipment connected with a ship's main engine can use low-speed diesel engines because low to medium load engines are often more energy efficient. This kind of choice will be significantly energy efficiency beneficial in particular for operating consistently at lower speed/loading conditions than the design service or heavy sea operation conditions.

Waste Heat Recover System

Large heat energy is generated by a ship's machinery system, in particular when running with full load and heavy weather conditions. For example, normally, about 5% of a ship's marine engine consumed energy/fuel is used for engine cooling system, and another 25% is simply emitted as exhaust gas such as CO₂. A waste heat recovery system is an energy recovery heat exchanger that transmits heat contained at exhausted gas at high temperature to other parts that can absorb or utilize the heat energy for their purposes to increase a ship's total energy efficiency. One traditional usage of this heat energy is, for example, to generate fresh water and steam for heating onboard ships. Currently, due to the increase of ship size leading to larger installed marine engine, it means that the exhausted gas (heat energy) contains much more energy than it can be used for fresh water and heating. Modern waste heat recovery systems in the market can be used to transfer the heat energy to generate additional steam to power a steam turbine, which can directly add mechanical force on the engine shaft to provide propulsion forces to ships.

Ship Performance Monitoring System

In order to optimize a ship's energy consumption, one of the key elements is to understand the ship's actual performance, which can be used to identify possible solutions/technologies to increase her current performance during the operation. Now a lot of companies are providing service to record, collect, monitor, and analyze a ship's performance in real time, such as the Marorka and Kongsberg systems. By continuously monitoring and analyzing a ship's performance, it cannot just help onboard crews become aware the importance to operate the ship in a more energy efficiency manner and understand the outcome of performance change caused by both internal and external factors but also provide background/evidence for fleet management team to plan a company ship energy-efficient management plan. Among other parameters, some fundamental parameters will be monitored by the system, for example, fuel consumption rate (per hour/nautical mile), specific fuel oil consumption (SFOC), ship speed, shaft power, shaft torque, RPM, etc. Another important parameter to be monitored is the so-called weather impact, i.e., how much extra energy consumption caused due to the encountered sea environment. In addition, many other important parameters related to the machinery system, environmental parameters (wind, wave, currents), and operational parameters (position, heading) are also collected in the system. A visualization platform is also integrated in the system to demonstrate a ship's performance in nearly real time. Meanwhile, some

low resolution of postprocessing data will be sent back to the ship's traffic management office through satellite.

In a ship's performance monitoring system, the most important component is to build the accurate baseline, which describes a ship's energy consumption in ideal conditions, i.e., sailing in calm water conditions with a newly maintenance hull and machinery system for various loading and operational (sailing speeds) conditions. Based on the baseline constructed in the system, the fleet management team could determine the ship's maintenance plan to increase the ship's energy efficiency. For example, when a ship is launched into the water, the microorganisms, plants, algae, or animals will continuously accumulate on the ship's wetted surfaces caused the so-called biofouling (Fig. 4), which can increase a ship's weight and thus the resistance significantly. If the biofouling occurs at the ship's propellers, it will reduce a ship's propulsion efficiency. Both can lead to large extra fuel consumption. In addition to every 5 years' dry dock maintenance, most shipping companies make other extra hull/propeller cleaning schedules to increase their ships' energy efficiency. The most optimal cleaning strategy is based on the performance analysis from a ship's monitoring systems.

In addition, the performance monitoring systems are often used to evaluate benefits of the implementation of easily plug-in energy efficiency solutions, such as the well-recognized weather routing and slow steaming. The weather

N



New Technologies in Auxiliary Propulsions, Fig. 4 Biofouling on a ship's hull (left) and on a ship's propellers (middle and right)

routing service is a computer-aided navigation system, which helps to plan a ship's sailing course and speed according to received weather forecast along the ship's sailing area. Often, due to large uncertainties of weather forecast especially after 2–3 days, the planned routes may not be very reliable. Furthermore, there are many weather routing services provided in the market. Therefore, the evaluation from a ship's performance monitoring system becomes an important source helping to select reliable energy efficiency technologies.

Renewable Energy Devices

The ocean contains huge amount of clean and renewable energies, such as the wind, tidal, and solar energy. Reliable technologies to harvest those renewable energy can significantly increase a ship's energy consumption. Through weather routing systems and training experiences, master mariners are making use of ocean currents (or avoid if sailing against ocean currents) to navigate their ships to save fuel burning. Furthermore, owing to the large deck space of merchant ships, such as tankers and bulk carriers, many innovative solutions have been proposed to install solar panels on board ships to harvest solar energy for providing electric power or even propulsion to ships. Concerning the whole sailing history, human beings have long history to make use of wind energy for ship propulsion. The wind power is also recognized as one of the most prominent renewable energy sources that can be realized by modern ships to reduce fossil fuel burning and air emission from shipping. In the following, we will focus on the description of wind propulsion concepts for ship navigation.

Wind Power as Auxiliary Propulsion in Ships

In the history of maritime industry, merchant ships using hybrid propulsions between steam engine and traditional sails have experienced about one century. Eventually, by the early 1900s, shipping companies migrated their ships to steamships. Gradually every transoceanic sailing, ship company went out of business. This is because of the

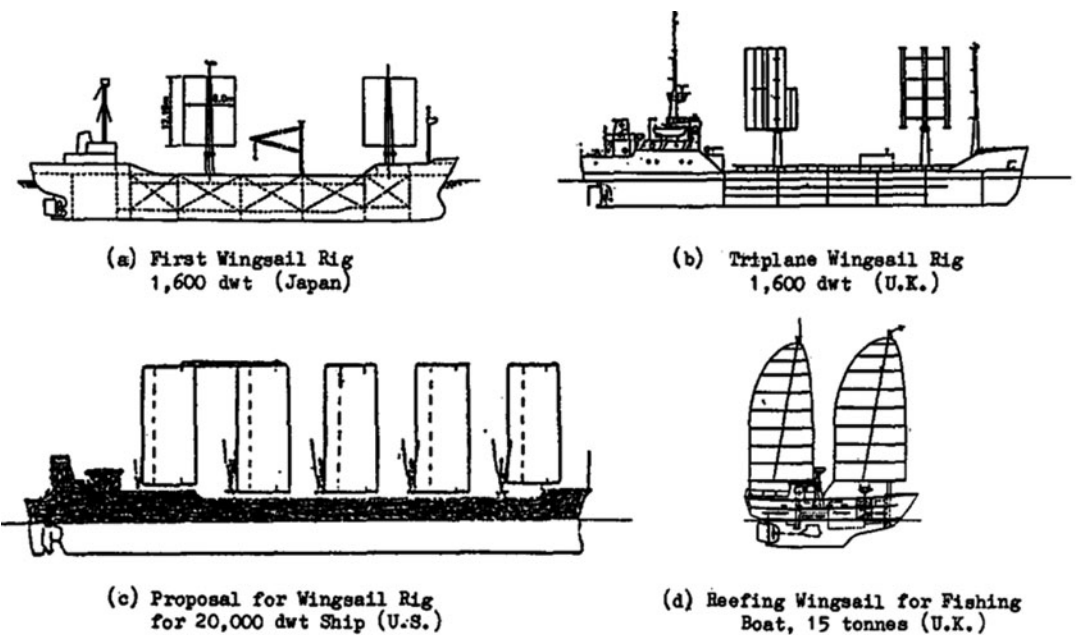
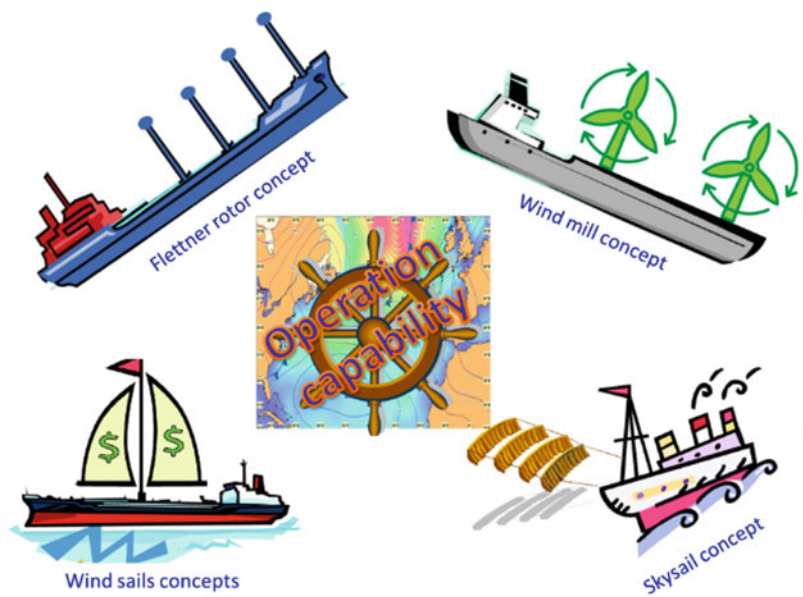
high energy efficiency of marine engine combined with modern propeller design and the low cost of energy sources. Furthermore, merchant ships with steam/marine engine are capable to sail across oceans faster than the wind and can be operated in a more regular and flexible manner. Figure 5 presents some of the most common discussed wind assistance ship using wind power as auxiliary propulsions. The state-of-the-art research and feasibility of these wind assistance ship concepts during the 1980s were extensively reviewed by, e.g., Nance (1985) and Bergeson and Greenwald (1985).

Wind Sail Concepts

Currently, hybrid ships using sails as auxiliary propulsion have attracted a lot of attention. Sails mounted on masts are often categorized as traditional soft sails and wingsails, which are airfoil-like structures similar to airplane wings. Traditional soft sails are still used for primary propulsion to many commercial fishing crafts in developing countries. These sails are also used on larger trading vessels in trade areas such as South Pacific routes with long distance and high fuel costs (Teeter and Cleary 2014). Soft sails are often designed as fore-and-aft sails or square sails, and one big advantage of using soft sails on merchant ships is their relative cheap investment for installation and maintenance. However, their drawbacks are also quite obvious, for example, lack of auto operation/control of the soft sails; it has to either rely on extra crew members to pilot sails or use very complex and expensive automate system for launch, recovery, and operation of the sails. Furthermore, the fuel savings using soft sails are usually very low in addition to the difficult handling problems (Schenzle 1985). Figure 6 lists a few wingsail design proposals in the early 1980s (Nance 1985). The most successfully implemented concept is the folding wingsail developed by the Japanese company NKK and fitted first to the *Shin Aitoku Maru*, a 1400 DWT tanker (Watanabe et al. 1982). Right after the successful installation of the system onboard the *Shin Aitoku Maru* tanker, it has also been retrofitted to about 10 ships of different sizes from 600 to 31,000 t, built in

New Technologies in Auxiliary Propulsions,

Fig. 5 Different auxiliary propulsion concepts using wind power (Mao et al. 2012)

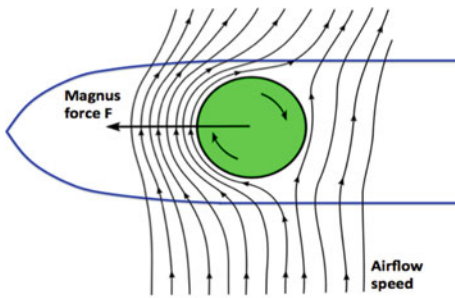


New Technologies in Auxiliary Propulsions, Fig. 6 Typical wingsail proposals developed in the early 1980s (Nance 1985)

6 different shipyards and operated by 6 different companies in Japan. This system has been demonstrated with a potential fuel saving up to 20–30%.

The Flettner Rotor Ship Concept

The Flettner rotor ship concept makes use of Magnus effect to generate propulsive force from the airflow around rotating cylinders. As is



New Technologies in Auxiliary Propulsions, Fig. 7 A sketch of Magnus effect for a Flettner rotor ship with vertical rotating cylinder

Fig. 7, assume that a ship is heading to the left, and the wind is blowing to the up direction. To generate propulsion force to the left, the cylinder must be rotated clockwise. The rotating cylinder will cause fractional drag force on the surrounding airflows, leading to that the speed of the airflow in front of the cylinder is greater than that behind. Based on the Bernoulli equation, it can cause lower air pressure in front of the cylinder and generate lift force on the cylinder to push the ship forward.

The first system to prove the Magnus effect concept that can provide auxiliary propulsion to ships was designed by German engineer Anton Flettner. The demonstration project was carried out on a schooner (the Buckau), which was outfitted with cylindrically shaped, rotating masts of 15 m tall and 3 m in diameter and crossed the North Sea from Danzig to Scotland in February 1925. In the past century, the Flettner rotor ship concept was never taken hold in the commercial shipping market due to its less competitive capabilities in comparison with steam and diesel engines as ship propulsions, even though the concepts were shortly revisited in the 1980s by maritime academia and research institutions driven by the concerns of the oil crisis (Nance 1985; Bergeson and Greenwald 1985). Flettner's cylindrical rotors were further with discs distributed along the cylinder. The performance of cylinder rotors with and without discs for ship auxiliary propulsion was experimentally studied by Clayton (1985). But the cylinder with discs distributed concept was abandoned by the inventor Thom since the power required to

spin the rotor with large discs became too expensive.

Due to the large weight and high central gravity of the rotors, a ship's structural integrity in some local details, as well as the vessel's hydrostatic and seakeeping capability, should be carefully planned and analyzed to ensure the safety. After 10 years' technical development, the first modern vessel, i.e., the 10, 000 dwt cargo vessel E-Ship 1 (ENERCON 2013) was completed and launched in 2010. She was equipped with four Flettner rotors of 27 m height and 4 m in diameter, demonstrating the design feasibility of the technology (Fig. 7).

Towing Kites Concepts

The concept of utilizing wind force from towing kites to provide propulsion forces to ships has been systematically invented since the 1980s (Duckworth 1985; Wellicome and Wilkinson 1984). Similar as other wind-assisted ship propulsion concepts, the incentive for the development of towing kite propulsions is also originated from the financial reasons and environmental concerns of bunker fuel consumption in shipping. Schlaak et al. (2009) investigated the worldwide shipping market for the implementation of wind towing kites as auxiliary ship propulsions. Based on the same result database from field trials with a kite into a study, it concludes that kite systems could be retrofitted to around 60,000 out of a total 90,000 ships. Significant fuel saving potential has been presented for a multipurpose freighter, between 1% and 21% for a ship speed of 15 knots



New Technologies in Auxiliary Propulsions, Fig. 8 A SkySails towing kite implemented in oceangoing commercial vessel (left), another kite propulsion concept developed by Kiteship (right). (Courtesy from the Internet)

and between 4% and 36% for a ship speed of 13 knots.

Before practical installation of kite systems in ships, two technical problems have to be solved. The first is a reliable prediction of mechanical performance of auxiliary wind propulsion using kites, and the other is the kite control/operation system, as well as the effect of ship structural and stability due to the installation and an optimum ship route plan system, which is required for all the wind-assisted propulsion concepts. The principle of ship propulsion using sails assumes that the aerodynamic forces are larger than kite masses, leading to the fact that the acceleration effects could be neglected. Thus, the whole kite system can be considered to be in equilibrium flight state. In addition, the kite operation system for auxiliary ship propulsion has also been investigated. For example, Grosan and Dinu (2010) studied the influence of a ship's stability and maneuverability with zero forward speed under certain wind conditions, Wrage (2007) patented his technical achievement for the kite launch and recovery systems, and Erhard and Struch (2012) presented their research efforts for an auto-pilot kite control, etc. These technical developments have enabled kite systems to be installed commercially for transoceanic voyages. It has been becoming the most popular wind-assisted ship propulsion concepts in the current shipping market due to its technical maturity. The industry partner SkySails GmbH is the market and technology leader in the field of marine towing kite

systems. SkySails kites are the key technology for capturing the vast potential of high-altitude winds, and SkySails is the first company in the world that has succeeded in developing towing kite technology into an industrial application. Figure 8 presents a prototype of SkySails towing kite implemented in oceangoing commercial vessel. The latest product generation developed by the company can replace up to 2 MW of the main engine's propulsion power. While another successful market player on the kite-assisted ship propulsion uses the system of different size and shape of kites developed by KiteShip (Kiteship 2018), which has a successful track record of using kites to provide propulsion force to pleasure yachts. They also built the world's largest sailing kite in 1997 and recently applied their systems to auxilarily power oceangoing tankers.

Other Wind Propulsion Concepts

Another more challenging concept is to install one or several wind mills onboard a ship. This concept can not only use the propulsion provided by windmills but also use the electrical power generated by windmills. Since wind mill propulsion generally provides high propulsive force only when the ship speed is less than about half the wind speed (Blackford 1985). This implies that for wind turbine propulsion to be the preferred for slow-steaming ships or sailing in high wind conditions. But the high wind conditions may induce extra challenges to a ship's structural safety and its stability in open sea operations. Ship motions

when operated in the open sea can significantly decrease the efficiency of the power generations from onboard windmills. Furthermore, to use the electrical power generated by the retrofitted windmills, large amount of investment must be put to revise marine engine system in ships. Moreover, maintenance cost may become too expensive during the operation period of windmills because of their working conditions under moist and salty sea environments. Hence, there are no merchant ships equipped with windmills to provide auxiliary ship propulsions (Böckmann and Steen 2011).

Cross-References

- [Energy Efficiency Regulations from IMO](#)
- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

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► [Pile Capacity](#)

NOMAD

► [Wave Measurement Buoy](#)

Nonexplosive Dismantling

► [Nonexplosive Removal](#)

Nonexplosive Removal

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Synonyms

[Nonexplosive dismantling](#)

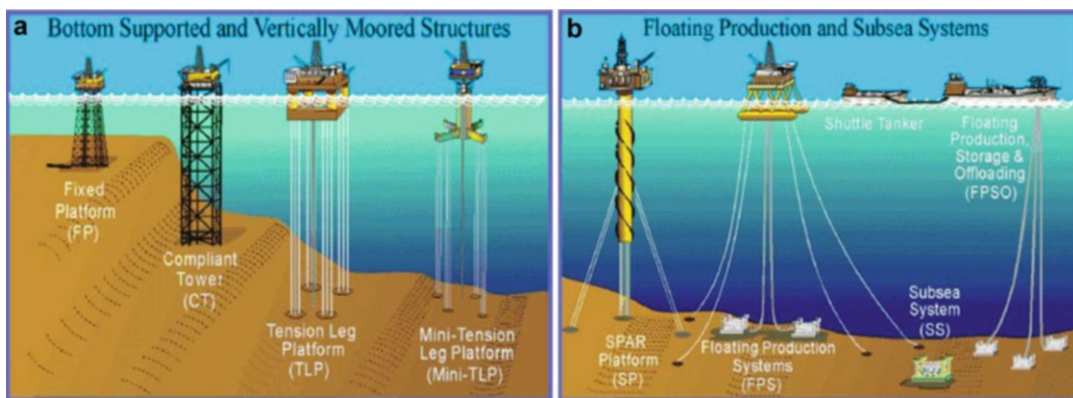
Scientific Fundamentals

The History of Nonexplosive Removal

Subsea systems are capable of producing hydrocarbons from reservoirs in all water depths and are frequently used to develop deepwater marginal fields. Deepwater subsea wells account for 164 of the 295 subsea wells in the Gulf of Mexico (GOM) (Baud et al. 2002). Compliant towers, spars, tensionless platforms, and floating production units (Fig. 1) are also employed in the deep water (Mark and Robert 2005).

The decommissioning of offshore structures, respecting the end of the production life, is a severing intensive operation (MMS 2004). Cutting is often required throughout the structure above and below the waterline and mudline on braces, pipelines, risers, umbilicals, manifolds, templates, guideposts, chains, deck equipment, and modules. More significant cutting operations are required on elements that are driven into the seafloor, such as multi-string conductors, piling, skirt piling, and stubs which need to be cut at a minimum of 15 below the mudline, pulled, and removed from the seabed.

A variety of technology exists to perform severance operations. These include abrasive water jet, diamond wire, diver torch, explosives, and mechanical and sand cutters. For severing operations that occur above the waterline, the cutting technique selected is usually dictated by the potential for an explosion. Cold cut methods are used when the potential for an explosion exists; otherwise hot cuts are employed. Cutting in the air zone is conventional since it involves methods which are regularly used for dismantling onshore industrial facilities. Below the waterline cutting is more specialized. In water depths that do not exceed 150 ft or so, divers perform cuts on simple elements such as braces and pipeline, and for shallow water structures such as caissons, torch cutting is sometimes the preferred severance method. Divers are also used to cut when other techniques produce an incomplete cut. In water depths exceeding 150 ft, remotely operated vehicles and automated diving systems are deployed with abrasive and diamond wire cutters and explosive charges. Major cutting operations



Nonexplosive Removal, Fig. 1 Deepwater development

on conductors, piling, and stubs normally employ mechanical, abrasive water jet, and explosive charges. Mechanical and explosive methods are primarily used for conductors with abrasive water jet and explosives predominately used for pile severance. The decision to use explosive and/or nonexplosive methods depends on the outcome of a risk-based comparative assessment involving cost, safety, technical, environmental, and operational considerations. The purpose of this report is to describe the factors involved in the decision to use explosive or nonexplosive methods: the business, market, and contract environment of nonexplosive technology and the environmental, physical, safety, and activity requirements associated with nonexplosive methods.

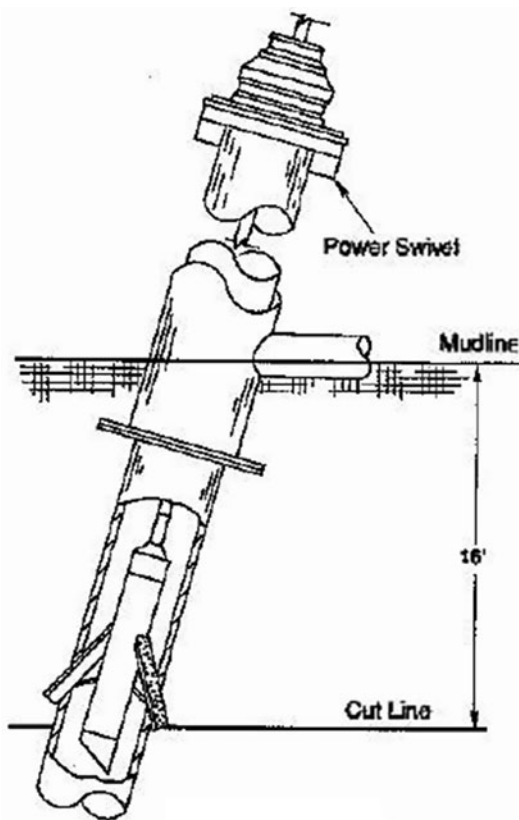
The Science and Technology of Nonexplosive Removal Methods

The science and technology of nonexplosive removal methods has changed remarkably, although significant progress has been made in diamond wire and AWJ (abrasive water jet) technology. For the most part, the resources required in decommissioning involve standard, readily available technology and tools which have been available for some time. There has been only incremental technological advancement associated with cutting technology over the past

decade, or indeed, over the past half century. Mechanical pipe cutters and diver torches are roughly the same today as they were 10 years ago, and while AWJ and diamond wire applications continue to see an increase in the frequency of application, a variety of factors continues to limit their application in practice. Nevertheless, while the routine application of diamond wire methods is still several years away, the technology has made significant strides. Further, the physical limitations associated with AWJ systems (e.g., water depth, cost, reliability) are significantly better than they were a decade ago, due to the steady influx of new contractors into the GOM and their product development.

Mechanical Methods

Cutting mechanisms that use hydraulically actuated, carbide-tipped tungsten blades to mill through tubular structures are called mechanical cutters. Figure 2 shows a schematic of this type of system. The mechanical casing cutter is perhaps the oldest method for cutting well conductors. The casing cutter is deployed on a drill pipe string, and the cutting tool has three blades which fold up against the drill pipe. The tool is lowered into an open pile or well using a drill rig. When hydraulic pressure is applied to the tool, the blades are forced outward as the tool is rotated by a power swivel. The carbide-tipped blades cut through the strings of the well, while centralizers on the tool keep it concentric inside the tubular



Nonexplosive Removal, Fig. 2 Schematic of a mechanical cutting system (MMS 2004)

member. Drillers watch the back pressure on the drill water to determine when the cut is complete, and cut verification can be made after the tool is recovered by the marks of penetration of the blades (Manago and Williamson 1997).

Mechanical cutters are frequently used for cutting shallow-water, small-diameter caissons with individual wells and well protector platforms with vertical piles (Fanguy 2001; Pulsipher 1996). Mechanical cutting is rarely used in conjunction with a derrick barge spread since the operation is time-consuming and inefficient and rig-up and rig-down time may be considerable. After wells are plugged and casing tubing cut and pulled, the contractor may run a mechanical cutting tool (or sand cutter) downhole to cut the conductors or, depending on the preference of the operator and configuration of the platform, may

subcontract for abrasive water jet or explosive severance methods.

To cut piling with mechanical cutters, the piling must be open at the surface to accommodate the power swivel. Thus, the deck of the platform must be removed prior to the operation using a derrick barge, and after the deck is removed, it is highly unlikely for a barge to stay on-site when the mechanical cutters operate. Remobilizing a derrick barge, however, is usually not an option and would represent a significant cost increase in the operation. Mechanical cutting is therefore rarely used for piling.

Abrasive Methods

Abrasive cutters are the mechanisms that inject cutting materials into a water jet and abrasively wear away steel/concrete. Abrasive technology has a long history of application in industrial and manufacturing processes and has been used in shipyards for many years. Several different systems of abrasive cutters exist.

Abrasive cutters can be classified as:

1. Low-pressure/high-volume systems (sand cutting)
2. High-pressure/low-volume systems (AWJ)

Cutters that use sand or slag mixed with water at low pressure (4000–10,000 psi) and high volume (80–100 gal/min) are called sand cutters, while cutters that use garnet injected at the nozzle at high pressure (50,000–70,000 psi) and low volume (50–80 gal/min) are commonly referred to as abrasive jet cutters (NRC 1986). The abrasive provides the force for cutting and is introduced at the cutting nozzle and sent down a hose with air pressure or through a water-based solution. Abrasive jet cutters produce a jet of water mixed with garnet under high pressure and directed through a diamond orifice. Abrasive jet cutters can cut both internal to the tubular member as well as external to the member, although internal cutting is the preferred method if below mudline access to the foundation pile can be achieved. The minimum inside diameter that can be accessed with abrasive cutters is approximately 7 in. and beyond 200–250 ft. Figure 3a and b

Nonexplosive Removal,

Fig. 3 Tool of abrasive cutters: (a) the abrasive water jet cutting tool for piles or caissons; (b) the cutting tool intended for use in conductors or small piles



shows the abrasive water jet cutting tool for piles or caissons and the cutting tool intended for use in conductors or small piles, respectively.

There also exists the problem of verifying that the cut has been made when using an internal abrasive cutter. Unlike explosives, the conductor or pile often does not drop, confirming that the cut was successful. With an abrasive tool, the width of the cut is small, and when combined with the soil friction, a visual response generally does not occur. To verify the cut, the conductor is pulled with either the platform crane or hydraulic jacks, and the lift force must overcome the conductor weight and the soil friction.

Diamond Wire Methods

A diamond wire cutting system uses a diamond embedded wire on a chain sawlike mechanism to cut steel, concrete, or composite material above and below the waterline. The wire is veered onto hydraulically driven pulleys resembling a band saw and mounted on a frame. The system can be configured to cut virtually any structural component, caissons, conductors, risers, and pipelines and is not limited by size or material as long as the cutting tool can be attached to the member. Figures 4 and 5 show diamond wire cutting tools for different applications. Figure 6 shows a close-up view of the cutting wire and industrial diamond beads.

The cutting machine is hydraulically clamped or manually strapped to the structure, and a surface-activated motor activates the tool. The diamond wire is driven at high speeds, and depending on the material and thickness, wire speeds are maintained to produce the cut. Even under large axial compressive loads, tubular members can be cut with diamond wire, and one of its strengths lies in its ability to cut large wall section thickness (Brandon et al. 2000). The operator monitors the progression of the cut and makes adjustments to improve the efficiency of the machine.

Diver Torch Methods

Underwater diver cutting is virtually the same as land-based cutting, but the torch used is somewhat different. In underwater arc cutting, an outside jet of oxygen and compressed air is needed to keep the water from the vicinity of the metal being cut. A tube around the torch tip uses air and gas pressure to create a gas pocket. This will induce an extremely high rate of heat at the work area since water dispels heat much faster than air. Underwater cutting can also be accomplished with an oxyhydrogen torch. Hydrogen is typically used instead of acetylene because of the greater pressure required in making cuts at increased depths.

In shallow water and for simple structures such as caissons, diving is sometimes the preferred

Nonexplosive Removal,
Fig. 4 Diamond wire
cutting system applied to a
72-in. deck leg on the
surface



Nonexplosive Removal,
Fig. 5 Diamond wire
cutting system deployed
from an ROV for use in
pipeline or small member
cutting



N

method. In deep water, because of the physical limitations associated with diving, an ROV or ADS system is commonly used in conjunction with a cutting torch. Diver cuts usually cost far more than other cutting technology, and the risk involved to the diver – especially in deep water – makes torch cutting generally less attractive than other removal options.

Key Applications

Environmental and Physical Impact

Energy is required to do work and all cutting operations require the expenditure of energy. As work is performed, energy is transferred and transformed which may have some impact on the ocean environment where these operations are



Nonexplosive Removal, Fig. 6 Local detail of the diamond wire, showing industrial diamond beads and spacers

performed. The power requirements of a cutting spread are approximately the same as a small offshore fishing vessel (less than 200 horsepower, or 150 kW). Unlike a typical sport fishing vessel, the typical cutting spread is fully self-contained with no marine discharges, other than the jet in the case of AWJ systems. Nonexplosive cutting methods are considered to be ecological and environmentally sensitive severance methods. For that reason, environmental and physical impact data of nonexplosive techniques are quite limited in the academic and trade literature. No record of negative environmental impacts for nonexplosive cutting methods has been found in the literature. In mechanical, AWJ, and diamond wire removals, diesel-fueled mechanical systems are employed in the operation which results in vibrations, the emissions of CO₂ and other gases to the atmosphere, and, potentially, low-frequency sound waves into the ocean environment. Abrasive water jet cutting involves using seawater and garnet or copper slag (grit). There is the question of the impact of the fluid and grit on the marine environment. Since the fluid involved in abrasive cutting is seawater and the grit is essentially inert, the environmental impact is believed to be inconsequential. Garnet is an inert rock material that poses no environmental consequences that have been reported. The level of available copper present in the slag is very low, and there are currently no restrictions on its use or reported environmental issues. The

noise level of the supersonic cutting jet is safe for divers and is not considered harmful to marine life. Mechanical, abrasive water jet and diamond wire methods are generally considered harmless to marine life and the environment. No adverse environmental impacts related to any of these cutting methods have been reported in the literature. The direct products of the processes are water, metal cuttings, and abrasive grit particles. Therefore, we conclude that there are no adverse environmental impacts associated with any of the nonexplosive cutting methods.

Nonexplosive Removal Market Potential

Mark and Robert (2005) did an estimation of the size and potential value of the nonexplosive severance market in the GOM. They assumed an average number of piles and conductors that need to be removed per structure type so that the total number of cuts can be estimated. Each caisson is assumed to be composed of one conductor, each well protector three piles and one conductor, and each fixed platform six piles and four conductors. In terms of the total number of tubular elements that are expected to be cut, the expected values $E(N_p)$ and $E(N_c)$ for the number of piles and conductors associated with each structure class were obtained. In their research, the value of the nonexplosive removal market in the Gulf of Mexico is estimated to range between \$1.5 M and \$2.5 M/year. The manner in which the nonexplosive removal market is “shared” among the subcontractors depends on the type of cuts that need to be made as well as the reputation, sales, performance, quality, and cost of the subcontractors providing the service. The final assumption decomposes the severance market according to mechanical/sand cutters, AWJ, and diver torch operations. The decomposition is a subjective assessment used to divide the market according to method of severance. Severing caissons with nonexplosive methods is assumed to be divided equally among mechanical/sand cutters, AWJ, and diving subcontractors. The split for well protectors and fixed platforms is assumed to be 60:40 and 80:20, respectively, relative to AWJ and mechanical cutting. In addition, changes in the

number of structure removals will impact the valuation estimate.

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Nonlinear Effects

- [Dynamic Analysis Method](#)

Nonuniform Distribution of Incident Waves

- [Station-Keeping System for VLFS](#)

Numerical Control (NC)

- [Forming Technology of Steel Hull Plates](#)
- [Piping Technology](#)
- [Ship Hull Lofting and Marking-Off](#)

Numerical Simulation Floe Ice–Sloping Structure Interactions

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Synonyms

[Computational fracture mechanics](#); [Floe ice](#); [Ice fracture](#); [Sloping structure](#)

Definition

*Sloping
Structure*

Offshore structure/ships with a sloping surface, which promotes the flexural failure of sea ice.

Broken Ice

Broken ice is loose ice, which consists of small floes that are broken up as a result of natural processes or active or passive intervention (ISO19906 [2019](#)).

Floe Ice

An ice field consists of ice floes of various sizes. The ice floes can be as large (in the order of kilometers) as being considered as level ice and as small (in the order of meters) as being considered as rubble ice (Lu [2014](#)). This is more general concept and includes

<i>Ice Fracture Simulation</i>	both the conventionally discussed “level ice” and “broken ice” conditions. From a numerical simulation point of view, this represents the simulation process to describe ice bodies changing from a continuum into fragments.
<i>Ice Rubble Transportation Simulation</i>	From a numerical simulation point of view, this is to simulate a collection of ice blocks interacting with each other and the structure; the ice blocks (ice rubble) normally slide along the ship hull or accumulate around the structure leading a significant amount of ice load on the structure.

Scientific Fundamentals

During the ice–structure interaction process, in most cases, the ice fractures first; then the ice fragments shall accumulate around the structure and need to be transported around the structure. Different interaction phases have been identified for this process for level ice–sloping structure interactions. These are:

- The ice breaking phase
- The ice rotating phase
- The ice rubble transport phase

For the ice breaking phase, there are many failure modes that can be identified depending on the ice feature, interaction velocity, and contact properties. The following theoretical simplifications can be made based on field observations:

- For level ice interacting with a sloping structure, it is mainly the local out-of-plane flexural type failures (Lu et al. 2015b) that are taking place.
- For floe ice, which is a more general term, the ice fails in both the local out-of-plane flexural

type failures and the in-plane splitting failure (Lu et al. 2015a); and the competition between these two types of failure should be accounted for (Lu et al. 2016a).

For the ice rotating phase and the ice rubble transport phase, we are essentially facing a multi-body dynamic simulation problem, that is, how an ensemble of ice blocks interacts with each other and the structure needs to be simulated.

Simulation Methods

The above two simulation tasks (i.e., ice fracture and ice rubble transport) can be solved with existing computational fracture mechanics and discrete element methods, respectively. Given the different physical natures of the processes that are to be simulated, different numerical simulation methods can be applied. Generally speaking, three categories of method can be identified. These are:

- Empirical formula-based simulation
- Hybrid approach
- Purely computational approach

Empirical Formula-Based Simulation

The empirical formula-based simulation is essentially a so-called “edge-tracking simulator” (Keijndener 2019). These simulations utilized empirical formula to calculate the ice fracturing (i.e., ice crushing and ice bending) and then utilize, for example, Lindqvist formula (Eq. 1), to calculate the ice rotating and rubble transport component.

$$\begin{aligned}
 R_s = & (\rho_w - \rho_i)ghB \left[T \frac{B+T}{B+2T} \right. \\
 & + \mu_{is} \left(0.7L - \frac{T}{\tan \phi} - 0.25 \frac{B}{\tan \alpha} \right. \\
 & \left. \left. + T \cos \psi \cos \phi \sqrt{\frac{1}{\sin^2 \phi} + \frac{1}{\tan^2 \alpha}} \right) \right] \quad (1)
 \end{aligned}$$

where

ρ_w and ρ_i	are the water and ice density.
h	is the ice thickness.
B, T, L	are the ship beam width, draft, and length.
$\psi = \arctan(\tan \phi / \sin \alpha)$	calculated from the stem angle ϕ and water entry angle α .
g	is the gravitational acceleration.

When applying the empirical formula, for example, to calculate the ice bending failure, the ice edge has to be constantly tracked and updated numerically, such that the ice–structure contact region can be identified.

When utilizing Eq. (1), it is implicitly assumed that 70% of the ship bottom shall be covered by ice rubbles (Lindqvist 1989). This leads to the coefficient 0.7 in the second term of the formula.

This method is rather simplified in terms of the ice fracturing and rubble accumulation terms; in addition, this method is mainly suitable for level ice–sloping structure interactions. The major challenge is to numerically track the ice edge to apply empirical ice breaking formulas. Extensive work has been carried out within this category. One can refer to work conducted in relevant literatures for detail (Su et al. 2010a, b, c, 2011; Tan et al. 2013; Zhou et al. 2011).

Hybrid Approaches

Computational Ice Fracture with Empirical Rubble Transport Formulations

Given the different interaction phases that are involved, one can choose different methods to simulate different phases and then combine them. For example, Valanto (2001) formulated the ice plate bending problem with consideration of the fluid beneath ice using potential theory. This is a rather computationally intensive approach to characterize the ice fracture part. However, when it comes to the rubble transport, Valanto (2001) simply adopted Eq. (1) to calculate the rubble transport. In this regard, extensive researches have been carried out on solving the ice fracturing part computationally; and in a

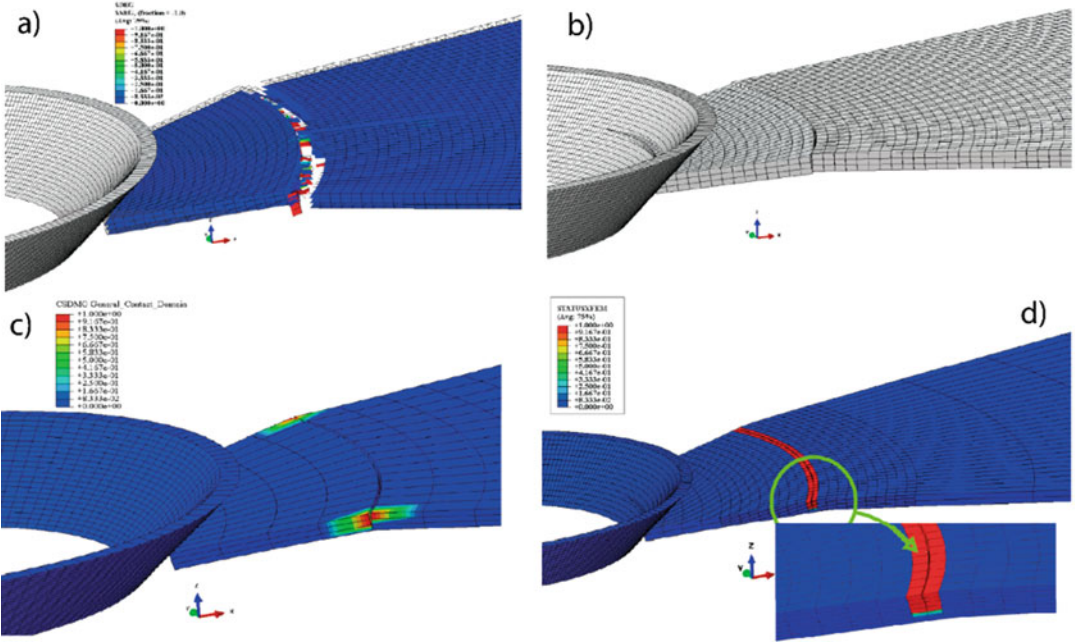
similar fashion, this load component can add up to the rubble transport part using the simplified Eq. (1) to calculate the total ice resistance. For example, Sawamura et al. (2008, 2010) numerically tabulated the results of an infinite ice wedge’s flexural failure, which were used for level ice–ship interaction simulations. The bending failure of an ice wedge is of major concern in this regard. There are many methods available to study the bending failure of an ice wedge; here we will present the cohesive zone method computational fracture mechanics.

The bending failure is controlled by circumferential crack initiation. Therefore, a strength-based theory would serve the purpose. However, it is possible to follow a fracture-mechanic approach to simulate the complete circumferential crack initiation and propagation process. There are many numerical methods available for this. Lu et al. (2012) presented the cohesive zone method based “element erosion method” (see Fig. 1a), “cohesive element method” (see Fig. 1b), “discrete element method” (see Fig. 1c), and “extended finite element method” (see Fig. 1d) to simulate a single flexural bending process. The simulated ice bending failure results can be tabulated and combine with existing rubble transport empirical formulas to calculate the total ice resistance.

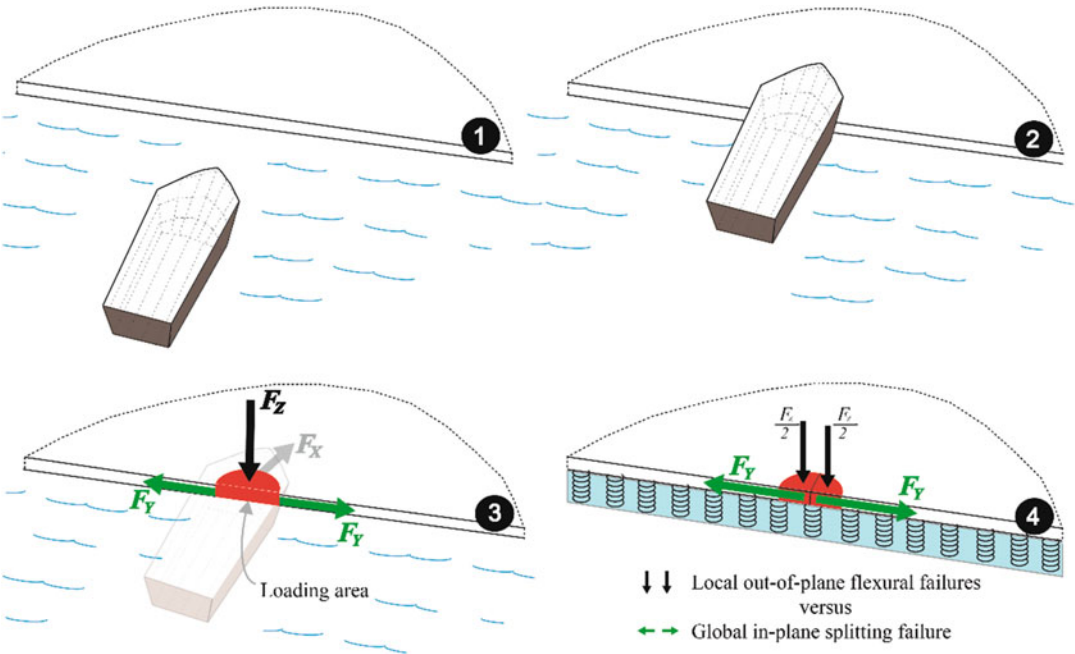
Analytical Ice Fracture with Computational Rubble Transport Formulations

Another hybrid approach is to calculate the ice fracture with analytical ice fracture solutions while simulate the rubble transport with Discrete Element Method. This work has been extensive developed and carried out in a series of recent studies (Lubbad et al. 2018a, b; Raza et al. 2019; Tsarau et al. 2018). The idea is to identify the most often observed failure modes of an ice floe and solve them by analytical solutions.

To achieve an analytical solution-based ice fracture calculation framework, the interaction mechanism between a floe ice and a floating structure can be simplified in Fig. 2 together with related contact forces. The two types of failure modes, that is, out-of-plane flexural failure by



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 1 Different numerical scheme to simulate the bending failure, by Lu et al. (2012)



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 2 Simplified interaction mechanism to with contact force components

the vertical force F_Z and the in-plane splitting failure force by the horizontal “wedge-out” force F_Y are considered separately in the following:

The vertical direction contact force F_Z , which leads to the out-of-plane bending failure, and the horizontal “wedge-out” force term F_Y , which leads to the global splitting failure, can both be formulated analytically (see the chapter on ► “Sloping Structure: Floe Ice Interactions”).

Combining the analytical solution in a DEM based simulation environment to simulate the rubble transport, the entire ice–structure interaction process can be simulated. The above analytical fracture based algorithms are implemented in the Simulator for Arctic Marine Structures (SAMS), which is a multibody dynamic based simulation tool (Lubbad et al. 2018a). A sample simulation snapshot is illustrated in Fig. 3. The top view in Fig. 3a shows the ice fracturing processes (both bending and splitting failure of ice floes) which are governed by analytical solutions; the underwater view in Fig. 3b shows how the broken ice rubble is sliding around the structure. The analytical fracture-based hybrid approach is rather efficient in terms of simulation. It can simulate an unprecedentedly large, special, and temporal domain, such that complicated Arctic operations, such as Ice management, which spans several kilometers and lasts for several hours can be

efficiently simulated by this hybrid approach (Lu et al. 2018b).

Purely Computational Approach

There exist many computational methods that can simulate both the fracture and upcoming fragmentation of ice floes in one goal. From a computational fracture mechanics point of view, Ingraffea and Wawrzynek (2004) made a classification and detailed review of all possible numerical methods. Here we will limit ourselves and selectively present the numerical methods that have been utilized in ice engineering; and the focus will be on ice–structure interactions.

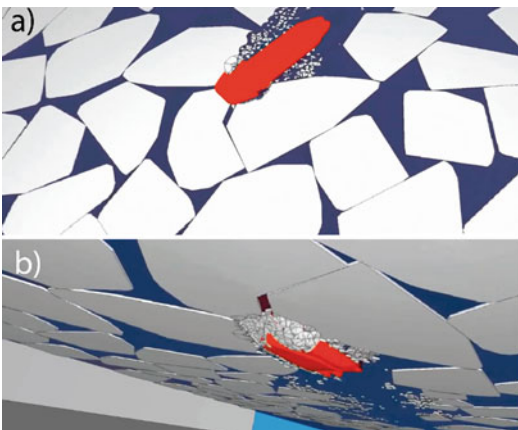
Combined Finite Element and Discrete Element Method

Before the ice fractures, finite element method is utilized to calculate the stress and displacement fields within the sea. When fracture takes place and results in ice fragments, the discrete element method is utilized to simulate the ice fragments. This method has been extensively studied and developed over the years by the research group in Finland (Paavilainen and Tuhkuri 2012, 2013; Paavilainen et al. 2006, 2009, 2010, 2011; Ranta et al. 2017; Tuhkuri and Polojärvi 2018). In addition, the research group in China has also carried out extensive research on using similar DEM method in simulating ice–structure interactions (e.g., Feng et al. 2015; Ji et al. 2013).

Cohesive Element Method

This method is available in most existing commercial software programs. It focuses more on the fracture modeling. The rubble transportation part is more an existing “default” feature that most FEM software have.

The cohesive element method is to characterize the material failure process following the so-called cohesive zone theory using “cohesive element.” Comparing to the conventional finite element, whose constitutive relationship is stress (unit: [Pa]) and strain (unit: [1]) based, the cohesive element is characterized by traction (unit: [N]) and separation (unit: [m]). The framework of this method has been described by Konuk et al. (2009a, b). It has been widely utilized in ice



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 3 Analytical ice fracture hybrid with DEM ice rubble transport simulation (simulated by SAMS (ArcIso 2020))

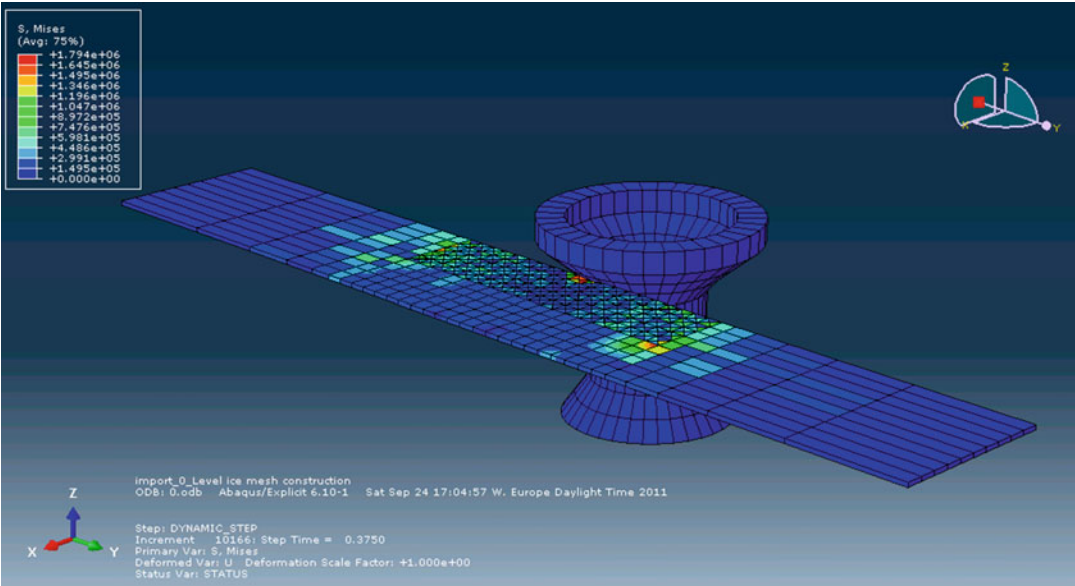
engineering since (e.g., Gürtner et al. 2008a, b, 2009; Konuk and Yu 2010; Feng et al. 2016; Kuutti et al. 2013; Wang et al. 2018, 2019; Lu et al. 2012, 2014).

Figures 4, 5, 6, and 7 show an example of such simulation concerning ice fracturing and ice

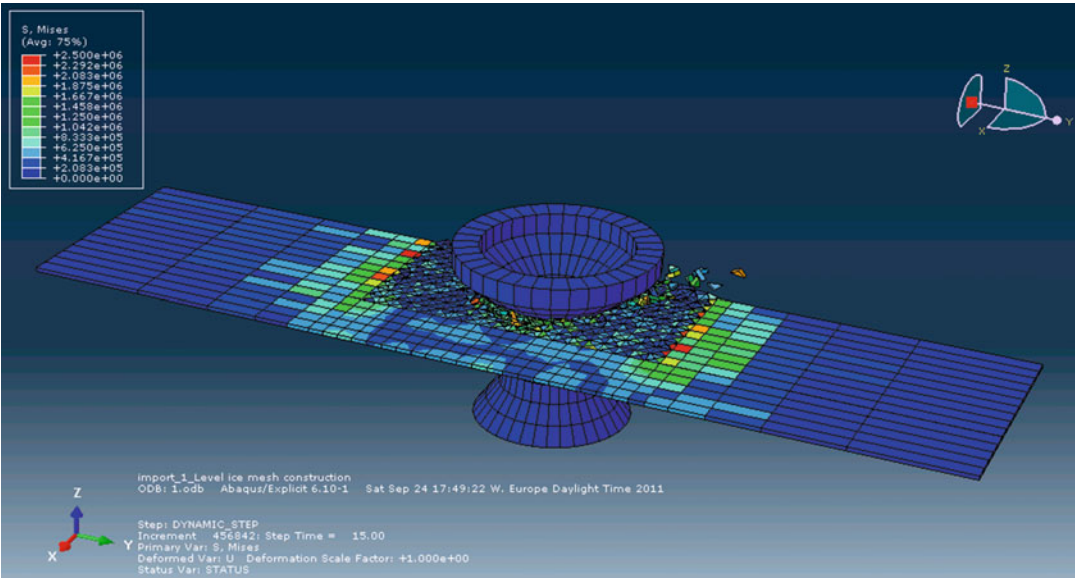
rubble’s transportation around the structure using inserted cohesive elements.

eXtended Finite Element Method (XFEM)

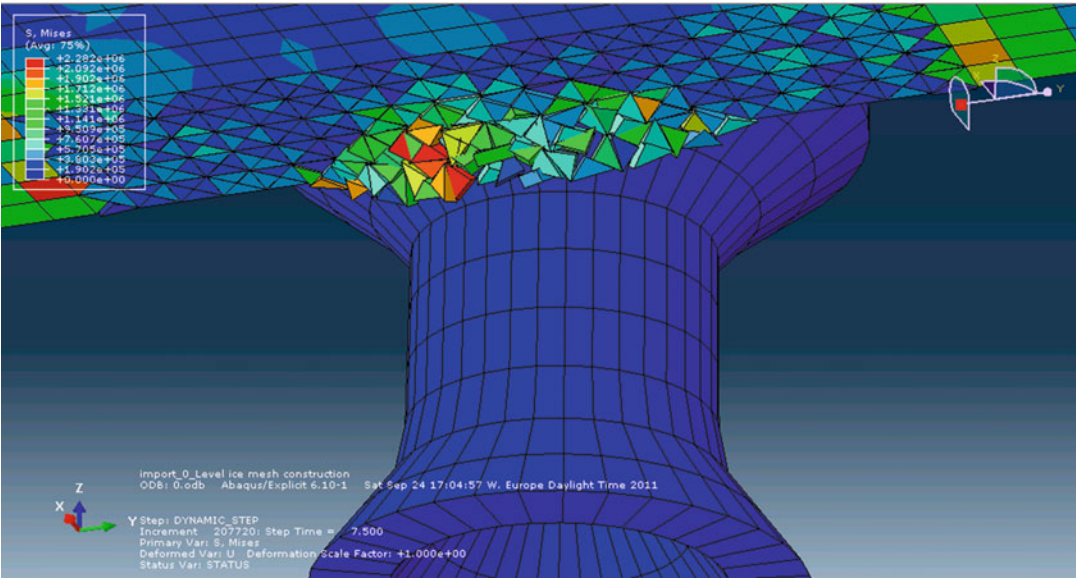
The XFEM can only simulate the fracturing process of ice, but not the fragmentation process. It is



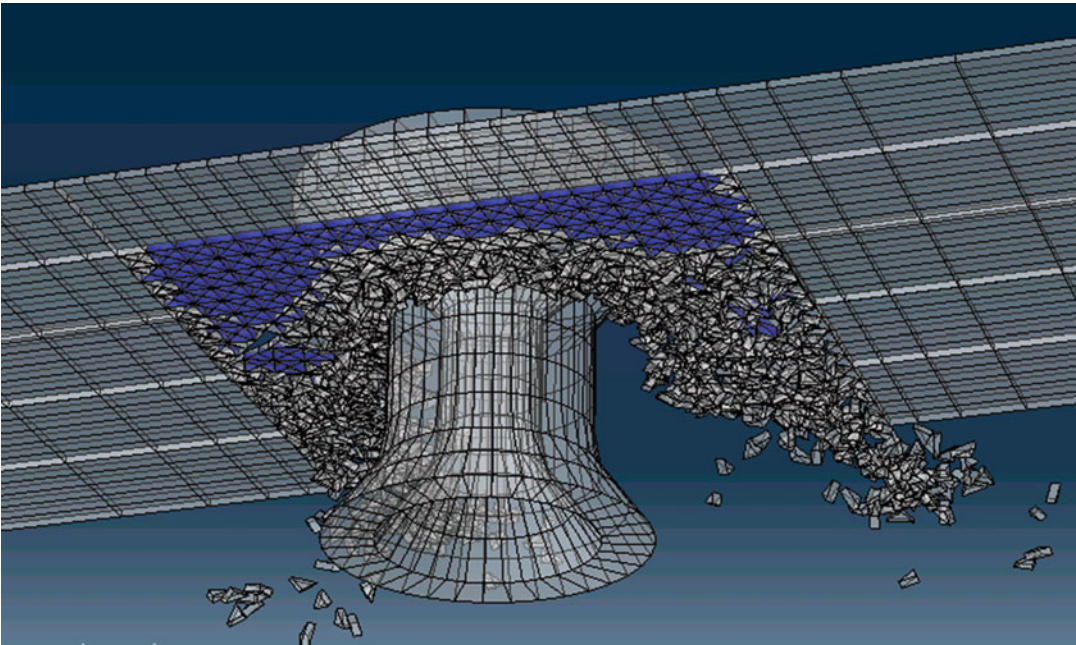
Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 4 A sloping structure enters an ice sheet



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 5 Ice fracture happens within the ice sheet



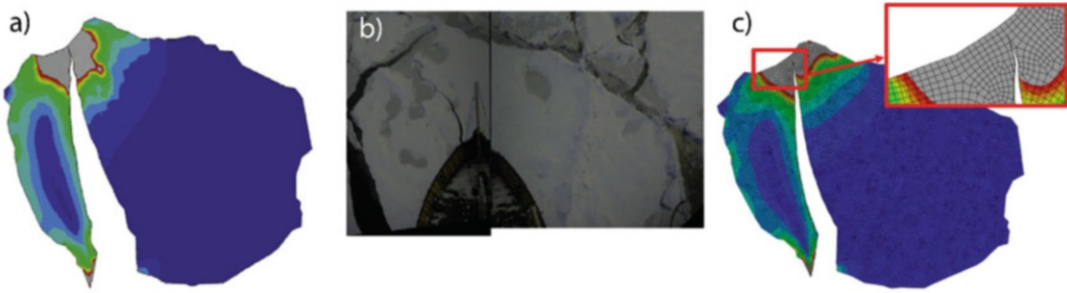
Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 6 Bottom view showing the generated ice fragments



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 7 A transparent view showing the not-yet-failed cohesive elements (in blue color) and rubble transport around the structure

a mesh independent method, which means the crack can propagate across the predefined mesh thanks to the Heaviside function to describe the

displacement field jump over the crack (Belytschko and Black 1999). It is useful to simulate large cracks within sea ice; but not the



Numerical Simulation Floe Ice–Sloping Structure Interactions, Fig. 8 Illustration of using XFEM simulating the splitting crack within an ice floe

fragmentation process. The fragmentation process should be simulated by a different numerical method, for example, DEM.

Figure 8 illustrates an example of using XFEM to simulate the splitting crack within an ice floe arbitrary geometry (Lu et al. 2018a). Fig. 8a shows the simulated crack path within the ice floe, which corresponds well with the actual interaction image (i.e., rectified image from an onboard camera system (Lu et al. 2016b)).

Lattice Model

Lattice model is a discrete approach and can be utilized simulate both the fracture and upcoming fragmentation of sea ice. Mass points (or bodies) are connected by lattice, either being spring, truss, or beam elements are utilized to formulate the ice body. If only the spring and dashpot type connector is used, the lattice model is rather similar to the conventional DEM method's formulation.

This method has been utilized to simulate the crushing failure of sea ice against vertical structures (Dorival et al. 2008; Jirásek and Bazant 1995); to the modeling level ice and ice ridges (Berg 2015); and has been evaluated by van Vliet (2018) for various ice failure modes and ice–structure interactions.

Peridynamics

The peridynamic method is also a discrete approach that gains popularity within ice engineering in recent years. Its formulation, implementation, and application in ice engineering can be found in related literatures (e.g., Oterkus et al. 2017; Vazic et al. 2019, 2020).

Applications

Numerical simulations of ice–structure interactions have seen increasing applications in recent years, partly due to the increase in our computational capacity and partly due to our ever-growing knowledge in simulating ice actions. The pertinent applications of each of the presented method can be found in the attached references. Here, we will stress one of the main advantages/applications of introducing numerical simulations in the concurrent Arctic offshore design loop. The current practice in designing an Arctic offshore structure, being an offshore platform or an icebreaker, involves model ice tank tests (Evers and Jochmann 1993; Timco 1984). However, model ice tank tests, despite its advantages of offering us well-controlled environmental conditions, suffer from: (1) slow preparation time; (2) limited variations in structural manufacturing due to resource limit; and (3) limited boundary conditions, comparing to numerical simulations. Additionally, as the ice conditions get more complicated, for example, the floe ice conditions, the ice tank tests' repeatability becomes questionable (van den Berg et al. 2020). Thus, introducing the above presented numerical simulation methods into the existing design loop is of great significance in the sense that it contributes to an essential complimentary component to the existing methods. This combination of “numerical ice tank” + “physical ice tank” can largely extend and optimize test scenarios, therefore leading to more efficient, more reliable, and less expensive Arctic structural designs.

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Numerical Simulation of Ice-Going Ships

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Synonyms

2D, two dimensional; 3DOF, three degrees of freedom; DEM, discrete-element method; DP, dynamic positioning; FE, finite element; FSI, fluid–structure interaction; GM, grid method; MIZ, marginal ice zone

Introduction

Good estimate of ice force is necessary to design ice-going ships because good hull shape design improves icebreaking and maneuverability and reduces the ice force acting on the hull, which reduce the required power. Currently, numerical models of ship–ice interactions have been developed and used for various ship and ice conditions. Four typical ice conditions must be considered when developing appropriate numerical models for ice-going vessel: level ice, pack ice, ridge ice, and icebergs. A main factor to assess ice-going ships is their performance in level ice. Early numerical modeling of level ice was conducted to calculate the icebreaking of continuous plate ice and to estimate the ice force (Valanto 1992, 2001). The drifting and impact of distinct ice floes should be considered for ship–ice interactions in pack ice (e.g., Hansen and Løset 1999; Wang and Derradji-Aouat 2010). Ship penetration in piles of rubble is a greatly challenging issue for the modeling of ships passing through an ice ridge (Gong et al. 2019). For ridge fields, estimations based on theoretical and empirical formulae are all that are available at present (e.g., Kuuliala et al. 2017; Myland 2014). The impact mechanism of ship–iceberg collision and material modeling of icebergs are difficult to construct in numerical models in ship–iceberg interaction (Liu and Amdahl 2010; Liu et al. 2011).

Normally, simulations calculate the global ice force (ice resistance) acting on the ship hull. Numerical results are used to estimate ship performance in the early design stage, such as the h – v curve and the required thrust (e.g., Su et al. 2010). The h – v curve represents the relation between ship speed and thickness under different ice conditions, which demonstrates icebreaking capability. Recently, numerical models are applied to more complicated sea ice conditions, such as dynamic positioning (DP) in ice (e.g., Metrikin 2014; Nguyen et al. 2009) and ice management in a marginal ice zone (MIZ) (e.g., Hamilton et al. 2011; Scibilia et al. 2014). Liu et al. (2008) and Lubbad and Løset (2011) developed real-time simulations for ship–ice interactions. These numerical models, if calibrated and validated properly, are useful tools not only for the early

design stage but also for crew training, for efficient operation in ice, and for onboard support systems in arctic operations.

This entry presents specific examinations of the numerical modeling used for most common ship–ice interactions: ship and level ice interaction. In terms of ship and level ice interaction, the failure of a flexural ice sheet and the motion of broken ice floes are considered. Numerical models for other ship–ice interactions can be developed using models both of the continuous plate ice and discrete broken ice.

Review of Numerical Models for Ships in Continuous Ice

Ship–ice interaction in the level ice is a repetitive phenomenon of ship–ice collisions and icebreaking of ice sheets. The first attempt to simulate the repetitive contacts and icebreaking is the numerical modeling of ice and conical structure interactions proposed by Izumiyama et al. (1992). In the numerical modeling, the icebreaking (crack) pattern and size are obtained using experimental data of the model tests in the ice basin. The icebreaking force is calculated using the ice failure model based on the plastic limit theory proposed by Ralston (1977). They mentioned that it is possible, in principle, to apply a similar method to prediction of ice resistance of ship.

Wang (2001) proposed numerical modeling for ice–cone interaction using a similar approach to that reported by Izumiyama et al. (1992). A hierarchy of discrete events that include crushing, bending, and rubble formation is adapted to simulate continuous ice–cone contact events. The ice–cone contact point is calculated using the grid method (GM), which represents the ice sheet as a regular grids. Contact occurs when the ice edge line is on the waterline of the structure. After the ice bending failure, a new ice edge appears by removal of the grids within the broken ice floe from the main ice sheet. The icebreaking force and the broken ice geometry are derived respectively from the empirical formulae proposed by Kashtelian (Kerr 1975) and results reported by Varsta (1983). This model calculates only repetitive peak force by the ice failure. It

neglects the rotation and sliding up of the broken ice floes. The developed model calculated the time history of ice force acting on the jacket structure.

Valanto (1992) developed a two-dimensional (2D) numerical model of icebreaking at the waterline of a ship advancing in level ice. The model specifically examines the ice sheet breaking cycle, incorporating the floating ice–fluid dynamics, ice edge crushing, floating ice failure, and broken ice rotation. The dynamic ice–fluid response and ice bending are assessed quantitatively using the assumption of potential flow with complicated boundary conditions on the ice sheet–fluid interface. The ice crushing and rotation of broken ice sheets were analyzed using geometrical formulations of ship–ice interaction. The 2D numerical model included effects of the compressive membrane stress in the ice sheet, which postpones the ice sheet failure (Valanto 1997). The numerical modeling has been extended to a three-dimensional (3D) problem, in which the underwater components of ice resistance were evaluated using the semiempirical formula presented by Lindqvist (1989) (Valanto 2001).

Many other researchers have adopted and modified the continuous contact between the structure and level ice derived by Wang (2001). Sawamura et al. (2009) proposed a numerical modeling of a ship advancing into level ice. The numerical model specifically examines icebreaking at the waterline and neglects broken ice rotating and sliding against the underwater hull. The repetitive ship–ice contact and plate ice failure are calculated using physically based modeling, which is often used in computer graphics, and plate bending obtained using finite element (FE) method. Nguyen et al. (2009) applied Wang's model to a DP system of ships in level ice. Su et al. (2010) developed a 2D numerical model using an empirical formula of icebreaking, crushing, and ice submergence. Ship motions are described by three degrees of freedom (3DOF) rigid motion equations in which the motions include surge, sway, and yaw. The ship–ice contact is calculated using more refined contact procedures based on GM method (Wang 2001). Numerical results (h – v curve and turning diameter) showed agreement with measured data obtained from a real trial of a ship turning in level ice. Su et al. (2011)

applied numerical methods to simulate a ship advancing into uniform and randomly varying ice conditions. This numerical model was extended to a 6DOF (surge, sway, heave, yaw, roll, and pitch) ship–ice interaction (Tan et al. 2013). The numerical models using a similar approach to that described by Su et al. (2010) were presented and used in different ship–ice interactions in level ice (e.g., Li et al. 2019; Zhou et al. 2016, 2017).

Lau et al. (2004) proposed real-time simulation of a ship maneuvering in level ice. The model realizes a short calculation time because analytical modeling of the icebreaking and buoyancy component is adopted. Liu et al. (2008) developed a software package of ship and level ice interaction using the numerical modeling of Lau et al. (2004). Lubbad and Løset (2011) developed real-time simulation of 3D ship–ice interactions. They combined the physics engine (3D rigid body simulation) with a self-developed icebreaking program based on a theoretical approach. In the simulation, the ice floes are represented as three-dimensional bodies with 6DOF. The ship motions and those of all ice floes (ship–ice and ice–ice interactions) are solved. The simulation was updated and validated using full-scale tests (Lubbad et al. 2018). Kubat et al. (2010) and Sayed and Kubat (2011) proposed a numerical simulation of ship transit through ice cover and simulated the pressure ice interaction. In the model, the governing equations account for the conservation of mass and the momentum of the ice cover. The ice cover is regarded as breaking in a rigid-plastic fashion. Many numerical models of ship and level ice interaction have been presented using a different approach (e.g., Aksnes 2010; Liu et al. 2018; Zhang et al. 2019). For the application of numerical simulation to the design for ice-going vessels, further improvement of ship–ice interaction is needed (Li et al. 2018).

Mechanism of Ship and Level Ice Interaction

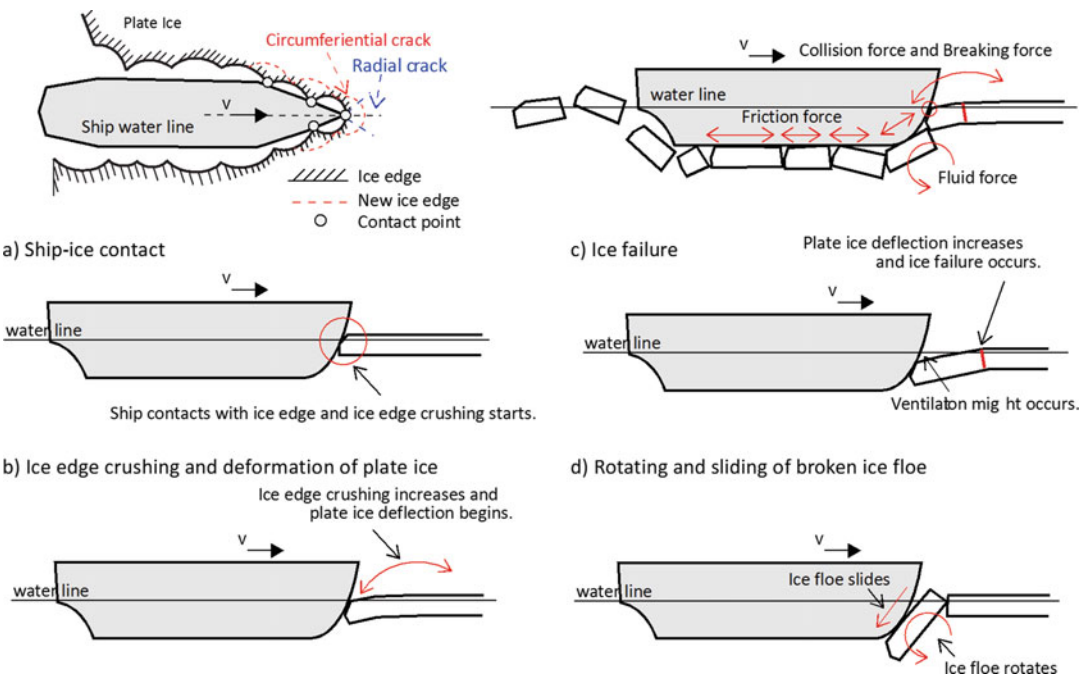
Ship–level ice interaction is a repetitive process with breaking, rotating, sliding, and clearing. A ship collides with level ice at the waterline

when it advances into the level ice. After a ship collides with a plate ice, ice edge crushing and plate ice deflection occur. When the ship advances further, the crushing area at the ice edge and the vertical deflection of the plate ice increase. At the same time, the stress in the plate ice induced by the plate ice deflection increases. When the stress in the plate ice exceeds the ice strength limit, ice failure occurs in different failure modes depending on the loading conditions e.g., bending, buckling, and splitting. After ice sheet failure, the broken ice floe is accelerated further and rotated by the ship hull until the ice floe is parallel to the hull. Subsequently, the broken ice floes slide along the underwater hull. Figure 1 portrays the icebreaking process.

The maximum ice force during the above ship and level ice interaction is induced by the ice failure. Therefore, icebreaking at the design waterline is the most relevant to evaluate the icebreaking performance. However, the ice force of the rotating and sliding of broken ice floes (ice submergence force) is smaller than the peak force

of the ice failure. The broken ice submergence engenders collision and friction force along the underwater hull, which makes a great contribution to the total ice resistance during the icebreaking cycle. Figure 2 presents idealization of the time history of ice force.

Many researchers divide ship-level ice interaction into icebreaking at the waterline and ice submergence underwater by the hull in their numerical model. Icebreaking is a simple physical phenomenon that can be represented by a quasi-2D model because the numerical simulation of 2D model is simulated with less computational time than those of the 3D model. Regarding ice submergence, interaction between the underwater hull and the broken ice floes is a complicated phenomenon. The ice submergence force is therefore often estimated using an empirical formula. The 3D numerical simulation of ship and level ice interaction including level icebreaking and ice submergence requires a powerful computer. The fluid force induced by ship-ice interactions is another difficulty hindering numerical modeling.

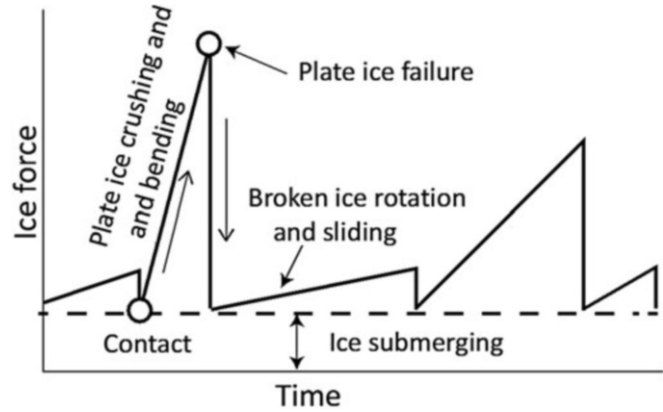


Numerical Simulation of Ice-Going Ships,
Fig. 1 Force components during ship and level ice interaction and icebreaking process: (a) ship-ice contact, (b) ice

edge crushing and deformation of the plate ice, (c) plate ice failure, and (d) rotating and sliding of broken ice floes

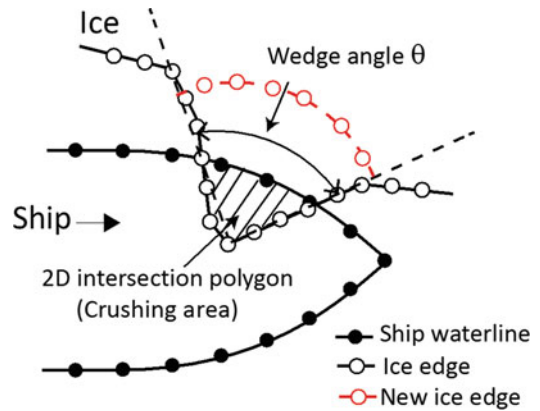
Numerical Simulation of Ice-Going Ships,

Fig. 2 Idealized time history of ice force



Ship–Ice Contact

When a ship advances into level ice, the ship collides with the ice edge at several locations at the ship waterline. The ice edges at the contact points are broken by crushing. The contact force at the ice edge increases concomitantly with increasing crushing area. For the numerical modeling, the precise contact points and the crushing (contact) area must be calculated. The 3D modeling requires a complicated algorithm and computational power. A 2D model at the waterline is therefore adopted in many numerical model (e.g., Lubbad and Løset 2011; Nguyen et al. 2009; Su et al. 2011). An example of modeling of the ship–ice contact point and the crushing area in the simulation is shown below (Fig. 3).



Numerical Simulation of Ice-Going Ships, Fig. 3 Contact detection and intersection area (crushing area)

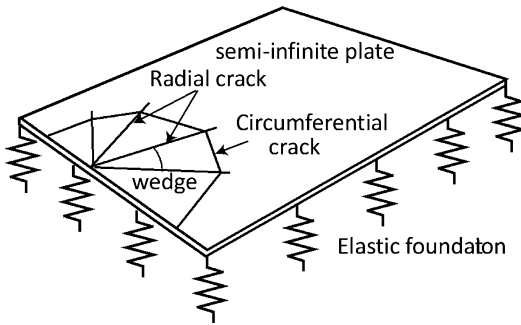
1. The ship waterline and ice edge are discretized as closed polygons.
2. Nodes and polygons that enter the polygon of both the ship and ice are identified.
3. The intersection area (crushing area) is defined at each time step.
4. Contact force is calculated based on the intersection area and the ice wedge angle.

The algorithm should be improved to maintain accuracy of the contact point and crushing area with less computation time. Zhou et al. (2016) adopted the point-in-polygon method. This algorithm specifically addresses on only the ice edges where ship–ice interaction might occur. Sawamura et al. (2009) adopted the circle contact detection algorithm to detect the ship–ice contact

points. The algorithm is often used in real-time animation of computer graphics. In his developed software package, Metrikin (2014) presented computational techniques of the ship and ice field modeling (mesh generation of the ice floes) and ship–ice interaction (collision detection and icebreaking).

Plate Ice Failure

In the ship–level ice interaction, the predominant failure mode is bending. First, the theory of an elastic plate resting on the elastic foundation, as a Winkler foundation (Fig. 4), was formulated to estimate plate ice bending behavior (e.g., Kerr 1975). Nevel (1961) calculated the deflection of the narrow infinite wedge, which considers radial and circumferential cracks when an ice sheet is



Numerical Simulation of Ice-Going Ships,
Fig. 4 Semi-finite plate resting on an elastic foundation
 with a radial and circumferential crack

broken by bending. Kashtelian (Kerr 1975) developed an expression for the bearing capacity of the wedge ice with edge force. These formulas deal with a floating plate with static load, but the actual state of the ship–ice interaction is dynamic load. However, static formulae are adopted by many numerical models because numerical implementation is simple (e.g., Lubbad and Løset 2011; Su et al. 2011).

Plate ice bending in ship–ice interaction is a rapid phenomenon. Therefore, dynamic effects of water underneath floating ice must be included in the bending behavior. Analytical formulations of the dynamic response attributable to an edge force on the simple shape ice sheet have been proposed (e.g., Fox 1993; Zhao and Dempsey 1996). These models are useful to estimate the bearing capacity by plate bending. However they cannot be used directly in the numerical modeling because the complicated wedge-shaped ice is generated by ship–ice interactions. Analytical solutions deal only with the simple ice shape. Sawamura et al. (2008) simulated bending behavior of wedge-shaped floating ice with a rapid edge force using fluid–structure interaction (FSI). They produced a database of the bending failure of the wedge-shaped ice and implemented this database into their numerical model (Sawamura et al. 2009). In addition to the vertical force (bending force), horizontal force (in-plane force) might also affect the ice failure mode. The in-plane stress by the horizontal force delays deflection of the sheet ice. Consequently, the icebreaking length and force

increase. Derradji-Aouat (2003) provided formulation for the 3D failure criterion of saline ice based on the multi-surface failure theory, which includes the effects of both vertical and horizontal force. Sawamura et al. (2010) investigated the effects of the horizontal force on the icebreaking during ship–level ice interactions using the FSI.

In ship–ice interactions, ice sheet failure patterns are characterized by radial cracking and circumferential cracking. According to the observations, radial cracks initiate from the ship–ice collision point. Subsequently, increasing force acting on the ice edge engenders circumferential cracks. Lubbad (2011) adopted both radial and circumferential crack failure criteria proposed by Nevel (1961) in his numerical simulation. Recently, fracture mechanics approaches have been applied by several researchers for crack initiation and crack propagation in floating ice sheets (e.g., Dempsey and Mu 2014; Lu et al. 2016). Nevertheless, ice failure based on observations from model-scale and real-scale experiments is adopted. For ice sheet crack pattern, a simple broken floe (e.g., circle ice floe) is assumed in most numerical models.

Ship in Broken Ice Floes

The ice force attributable to rotating and sliding of the broken ice floes is induced by buoyancy and ship–ice collision along the underwater hull. The ice submergence is a complicated phenomenon that entails many uncertainties. Actually, 3D simulation of the ice submergence requires many computational power. For this reason, most numerical models apply the empirical formulae to evaluate ice submergence force (e.g., Liu et al. 2008; Nguyen et al. 2009; Su et al. 2010).

To simulate ice submergence, an efficient 3D numerical model is required. Lubbad and Løset (2011) used NVIDIA PhysX, a real-time physics simulation engine including a rigid body dynamics, soft body dynamics, and particles and fluid simulation (NVIDIA Corp. 2018), to solve the equations of rigid body motions in 6DOF for all ice floes. They combined the self-developed icebreaking module with NVIDIA PhysX and simultaneously simulated both icebreaking and ice submergence. Konno and Mizuki (2006)

used the Open Dynamics Engine Library (Smith 2004), which is also an open-source physics library, to implement a physically based modeling of ship–ice floes interactions. They predicted ice-clearing resistance with the 3D numerical simulation of pre-sawn ice test. Konno (2009) applied the same physically based modeling to the ship in brush ice channel. Sawamura and Tachibana (2011) developed the numerical model to calculate the ice submergence force of the ship in pre-sawn ice using physically based modeling. The physics engine cannot support icebreaking: the physics engine is available for unbreakable ice floes.

Lau (2006) presented numerical modeling of ship maneuvering in level ice conditions using discrete-element method (DEM). The discrete elements in the ice channel are divided into the broken ice floes with the observed icebreaking pattern (pre-sawn ice). Each ice piece is connected by stiffness springs. The stiffness spring in most of the numerical model using DEM must be chosen using trial and error based on the experimental and measured data. DEM is an effective method to simulate the discretized bodies such as broken ice. Therefore, it has been adopted for numerical modeling of a ship in pack ice (e.g., Hansen and Løset 1999; Ji et al. 2013; Wang and Derradji-Aouat 2010; Zhan et al. 2010) and ridge ice (Gong et al. 2019). A reasonable sea ice model must be required in the DEM simulation because 3D simulation requires powerful computer resources and long computational times. The fluid modeling of ship–ice interactions is another challenging difficulty for numerical modeling. Ventilation phenomena are not considered in most numerical models.

Numerical Modeling of a Ship in Continuous Ice

Methodologies used for the simulation of the repetitive ship–ice collision and icebreaking at the waterline are introduced. In this section, the numerical modeling proposed by Su et al. (2010) is explained mainly, because their theories can be implemented easily into the numerical model and

many researchers have referred this modeling method (e.g., Nguyen et al. 2009; Sawamura et al. 2009; Zhou et al. 2016). The simulation evaluates repetitive ship–ice collision and icebreaking at the waterline. Ice submergence under the waterline is evaluated using the empirical formula.

Motion Equation

The ship motion is described by 3DOF rigid body equation in 2D model and 6DOF in 3D model as

$$(\mathbf{M} + \mathbf{A})\ddot{\mathbf{x}}(t) + \mathbf{B}\dot{\mathbf{x}}(t) + \mathbf{C}\mathbf{x}(t) = \mathbf{F}(t), \quad (1)$$

where $\mathbf{M}$, $\mathbf{A}$, $\mathbf{B}$, and $\mathbf{C}$, respectively, denote the mass, added mass, damping, and restoring force matrices. Also, $\mathbf{F}$ represents forces and the moment vector acting on the ship; $\mathbf{x}$ denotes the ship displacement vector. The force and moment vector can be decomposed into the following components:

$$\mathbf{F} = \mathbf{F}_H + \mathbf{F}_P + \mathbf{F}_R + \mathbf{F}_{Ice}. \quad (2)$$

Therein, subscripts H, P, R, and Ice, respectively, denote forces of hydrodynamics, propeller, rudder, and ice. For the motion equations for the ice-going vessels, the specific component is only the ice-induced force $\mathbf{F}_{Ice}$. Furthermore, ice force can be written as

$$\mathbf{F}_{Ice} = \mathbf{F}_{break} + \mathbf{F}_{sub} + \mathbf{F}_{others}, \quad (3)$$

where $\mathbf{F}_{break}$ signifies the icebreaking force, $\mathbf{F}_{sub}$ denotes the ice submergence force, and $\mathbf{F}_{others}$ stands for other force component induced by ship–ice interaction. At each time step, Eq. (1) is solvable by the numerical integration such as Newmark method and Runge–Kutta method.

Ice Contact Force

At the ship–ice contact, ice crushing occurs at the ice edge. The contact force increases proportionally with increasing crushing area. When the ice pressure on the contact surface is assumed to be constant, the contact force at the contact surface between the ship hull and ice edge is given as

$$F_n = f_{ic} A_c \sigma_c, \quad (4)$$

where F_n stands for the contact force normal to the crushing area, f_{ic} expresses the contact coefficient, A_c denotes the crushing area, and σ_c signifies the compressive strength of ice. Contact coefficient f_{ic} accounts for irregularity of the contact area. Valanto (2001) used the value of 0.5 for the contact coefficient for the ship hull–full-scale level ice contact in his numerical model. Crushing area A_c increases with the ship penetration distance into an ice edge. The plate ice deflection is very slight compared to the characteristic length of ice. It is assumed to be zero before ice bending failure in the numerical modeling because of simplicity of the calculation. Crushing area A_c is calculated numerically or geometrically using the relation between the ship hull and ice edge, as described in section “Ship–Ice Contact.”

The kinematic friction force is presented on the crushing surface. Coulomb type friction is assumed, which direction of frictional force is opposite to the relative velocity. The total ice force attributable to ship–ice interaction is calculated as the sum of the contact force and the friction force. The total ice force is divided into vertical and horizontal components at the contact surface, as presented in Fig. 5. It is derived as the following equations.

$$\begin{aligned} F_Z &= f_{ic} A_c \sigma_c (\cos \varphi - \mu \sin \varphi) \\ F_H &= f_{ic} A_c \sigma_c (\sin \varphi - \mu \cos \varphi). \end{aligned} \quad (5)$$

Therein, F_Z and F_H , respectively, denote vertical and horizontal components of the ice force, φ

signifies the slope angle, and μ stands for friction coefficient between the hull and contact surface. The ice edge is subjected to a sudden downward force F_Z , which deflects the ice sheet by plate bending. The horizontal component F_H might affect the floating ice bending. However, it is often neglected in the ice failure calculation. Ice crushing takes place at the ice edge until the crushing area increases the downward force sufficiently to initiate the flexural failure. Otherwise, only the ice edge is crushed without bending failure.

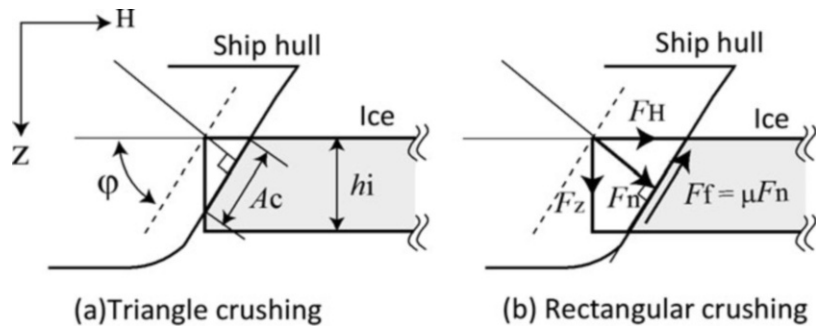
Ice Failure and Breaking Force (F_{brek})

The floating ice is deflected by the downward force. It is eventually broken by bending failure. The ice failure criterion with the bearing force of floating ice is used. Ice bending failure occurs when downward force F_Z exceeds the bearing force of the floating ice. The bearing force of the floating ice with a quasi-static edge force is calculated according to Kashtelian (Kerr 1975).

$$P_f = C_f \left(\frac{\theta}{\pi} \right)^2 \sigma_f h_i^2 \quad (6)$$

Therein, P_f stands for the failure force of the floating ice, θ represents the ice wedge angle, σ_f denotes the flexural strength of ice, h_i expresses the ice thickness, and C_f is an empirical coefficient. Kashtelian suggested a small value (approx. 1.0) for C_f , whereas Su et al. (2010) adopted a value of 3.1 by comparison of ice resistance with the empirical formula reported by Lindqvist (1989).

Numerical Simulation of Ice-Going Ships,
Fig. 5 Ice force between the ship and ice edge



After bending failure, a circular ice cusp is broken off the plate ice. The broken cusp length is determined using the formula presented in Wang (2001), which is based on methods described by Enkvist (1972) and Varsta (1983).

$$r_f = C_l l (1 + C_v v_n^{\text{ref}}) \quad (7)$$

Therein, r_f denotes the circumference crack length, v_n^{ref} stands for the velocity component normal to the contact surface, C_l and C_v express empirical formulae, where C_l has positive value and C_v has negative value, and l represents characteristic length of ice given as

$$l = \left(\frac{E h_i^3}{12(1 - \nu^2) \rho_w g} \right)^{\frac{1}{4}} \quad (8)$$

where E represents Young's modulus, ν stands for Poisson's ratio, g signifies the gravitational constant, and ρ_w denotes the density of seawater.

Ice Submergence Force (F_{sub})

The ice resistance attributable to ice submergence is estimated using empirical formulae. Su et al. (2010) adopted the submersion resistance formula proposed by Lindqvist (1989).

$$\begin{aligned} R_{\text{sub}} &= R_s \left(1 + 9.4\nu / \sqrt{gL} \right) \\ R_s &= \Delta_p g h_i B \left\{ \frac{T(B+T)}{B+2T} + \mu (A_b + \cos \phi \cos \psi A_f) \right\} \\ \psi &= \arctan (\tan \phi / \sin \alpha) \end{aligned} \quad (9)$$

Therein, R_{sub} and R_s , respectively, denote the ice submergence resistance with and without speed dependence, ν denotes the ship speed, L , B , and T , respectively, represents the ship length, breadth, and draft, Δ_p is the density difference between the water and the ice, α , ϕ , and ψ respectively, denotes the waterline entrance angle, stem angle, and angle between the normal of the hull and a vertical vector, and A_b and A_f , respectively, represent the area of flat bottom and the bow of the ship. Zhou et al. (2016) used the model proposed by Spencer and Jones (2001) in which

submergence resistance is divided into buoyancy and ice-clearing components. Lubbad and Løset (2011) numerically evaluated the ice submergence force using the 3D rigid body simulation of the physics engine.

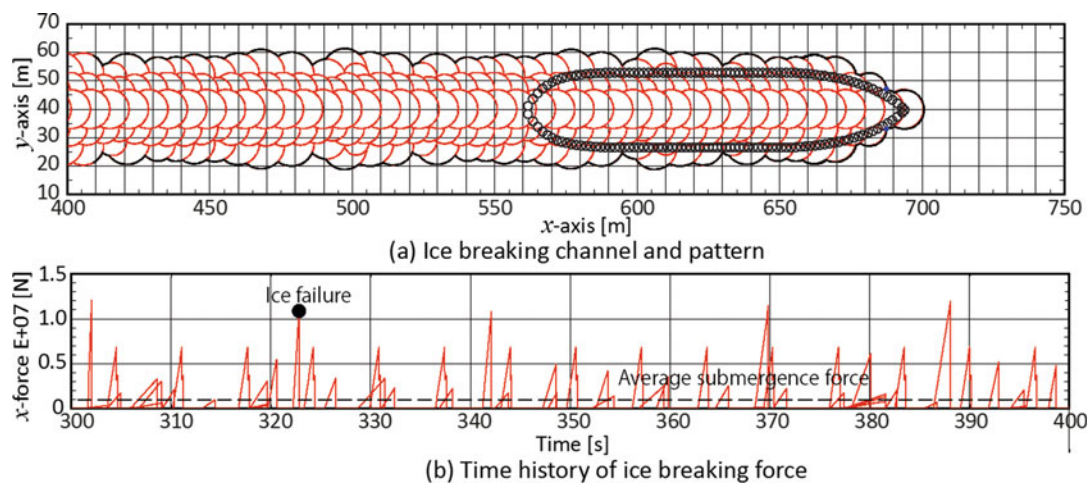
Simulation of Ship–Ice Interaction

The numerical model mainly divides the ship–ice interaction into four components: ship motion, ship–ice contact, icebreaking along the waterline, and ice submergence under the waterline.

1. Ship motion is calculated using rigid body motion equations using the icebreaking and submergence force at the previous time step (motion equation).
2. Ship–ice contact along the waterline is calculated using contact detection technique. Subsequently, the ship–ice contact conditions (ice crushing, ice contact force) are updated (ice contact force).
3. Ice failure is judged using the ice failure criterion. If bending failure occurs, then the ice cusp is broken off the plate ice (ice failure and breaking force).
4. Ice submergence force and the positions of all broken ice floes (if necessary) are calculated (ice submergence).

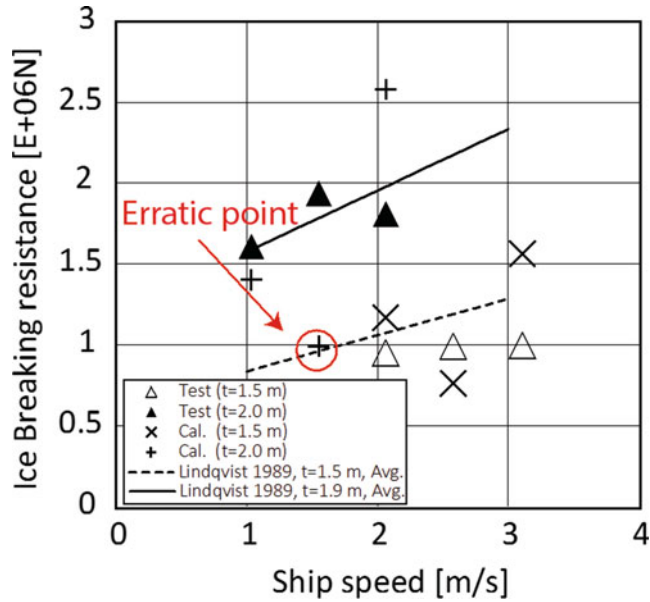
The processes presented above are calculated separately and are summed to complete one numerical cycle in each time step.

Figure 6 presents the computed ice channel, breaking pattern, and time history of icebreaking force calculated using the 2D numerical model based on an earlier report of the literature (Sawamura et al. 2017). The principal dimensions of the ship are 136 m length, 29 m width, 10 m draft, and 20° bow angle. The ice sheet is 2 m thick. Young's modulus of the plate ice is 254 MPa. The ice flexural strength is 575 kPa. The compressive strength of the ice is 820 kPa. The friction coefficient between the ship and ice is 0.075. The icebreaker advances with constant speed of 1 m/s. The ship motion equation is neglected. The total ice force is obtained as the sum of the ice submergence force to the icebreaking force as depicted in Fig. 6.



Numerical Simulation of Ice-Going Ships, Fig. 6 Numerical results of ice channel, breaking pattern, and time history of icebreaking force

Numerical Simulation of Ice-Going Ships, Fig. 7 Relation between ice resistance and ship speed and comparison with the model test data and ice resistance formula



In Fig. 7, the numerical results are compared with the ice resistance of model test data and Lindqvist (1989). The numerical resistance shows good correlation with the model test and the empirical formula. The erratic point is shown, however. This irregularity of the ice resistance is apparent also in the numerical results reported by Su et al. (2010) and by Liu et al. (2008). These results demonstrate that the numerical results are sensitive to the ship and ice conditions.

Future Directions

Numerical simulation can evaluate the icebreaking process of a ship in simple ice conditions (level ice and uniform pack ice field). Further improvement of ship–ice interaction is necessary when the numerical simulation is applied to the design for ice-going ships. Recently, numerical models apply more complicated ship–ice interactions (DP and ice

management operation in MIZ). The numerical model of a ship in an MIZ should incorporate consideration of ice–wave interactions because the ship is affected by small ice floes and waves in the MIZ. The ship in a ridge field and ship ramming operations remain as challenging topics for numerical modeling. Verification of the numerical model must be done through comparison with real ship trials. However, many difficulties might arise with collection of data in the actual ice and ship conditions. The model- and real-scaled measurement technique should also be improved to acquire and accumulate verification data. The fluid model during ship–ice interactions and the failure criteria of bending, crushing, and other failure modes of floating ice plates should be investigated to improve the numerical model accuracy.

Cross-References

- [Ice Breaking Vessel](#)
- [Numerical Simulation Floe Ice–Sloping Structure Interactions](#)
- [Sloping Structure](#)

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Numerical Tank

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Synonyms

Digital tank

Antonym

Physical tank

Definition

Numerical tank is a set of highly efficient computing software which is programmed systematically by using advanced hydrodynamic model and numerical algorithm as well as professional experience. Combined with advanced computer and Internet service, Numerical Tank can perform highly accurate virtual tests on hydrodynamic performance for marine structures and hence meets the requirements for research, design, and application in the field of naval architecture and ocean engineering ([Numerical tank](#)).

The research and development of the Numerical Tank Application System is supposed to have high virtual test accuracy standards and rely on

high efficient hydrodynamic model; it also needs the verification support of physical tests data. In addition, the Numerical Tank Virtual Test System, in its process of development, made integrated packaging of experts' experience in geometrical modeling, mesh discretization, and computing parameters so to acquire stable and reliable virtual test results (Zhao et al. [2014](#)).

Compared with the physical tank in terms of test objects, test environment, and test content, the numerical tank not only “tests” the overall hydrodynamic force and motion response but also “tests” the fluid information such as fluid velocity and pressure with good accuracy more easily. The test results satisfy various requirements of ship hydrodynamic performance evaluation and optimization. In the present trend of development, it is expected that the numerical tank will go beyond physical tank and be able to perform the full-scale simulation of the motions and loads for the marine structures, which will contribute to the design of more optimal ships and offshore platforms and serve the base of intelligent marine operation. The numerical tank technology brings the revolutionary change to the design methods of marine engineering equipment, so it carries great significance in engineering.

Scientific Fundamentals

The ship hydrodynamic performance measurement is mainly carried out through physical tank test for a long time. There are various types of physical tanks around the world, which include towing tank, ship seakeeping tank, ship maneuvering tank, cavitation water tunnel, etc. These physical tanks can perform various tests on ship hydrodynamic performance including ship forward speed, ship motions in waves, ship maneuverability, etc. It can also provide guidance in the selection of optimal ship. Currently, the ship hydrodynamic performance test in the physical tanks is still the most important and reliable way to test the ship's hydrodynamic performance.

However, the traditional physical tank test has the following disadvantages:

- (a) It usually requires high costs of installation and long operating time.

- (b) New unforeseeable requirements for the test are sometimes difficult to be satisfied by the physical tanks.
- (c) It is impossible to simulate the hydrodynamic loads for full-scale ships in the physical tank due to the limited physical tank dimensions.
- (d) The interruption of the space and time in the ship design cycle, during the physical test, remains the dominant bottleneck for the intelligent design.

Faced with the requirements from the new regulations regarding the low emission and safety on the ship design, and the challenges of the harsh weather conditions on the design of offshore equipment in deep water, the traditional physical tank cannot adapt to the development of the modern technologies, which has the limiting characteristics of big investment, long operating time, and limited dimensions. With the fast development in computer hardware technology, as well as the continued improvement in computational fluid dynamics (CFD) technology and the software functions, CFD technology has found its wide application in optimizing of hydrodynamic performance for marine structures. In order to adapt to the change from the physical tank test to the virtual test in the research and development for the marine structures, the idea of the numerical tank technology is hereby proposed, which combines the CFD technology and its application experiences during the hydrodynamic performance simulation.

The core basis of the numerical tank is the CFD technology, which mainly includes the potential flow and viscous flow numerical models. The practical effectiveness of the numerical tank lies in the obtained simulation experience based on the fluid characteristics of the ships and other marine structures. In addition, the high efficiency application of the numerical tank also involves the parallel computing techniques, visual technology, Internet technology, etc.

Key Applications

The numerical tank can perform the ship resistance and propulsion virtual test, propeller cavitation-induced exciting force virtual test,

ship seakeeping virtual test, ship maneuvering virtual test, ocean wave environment virtual test, motion and loads virtual test for offshore platform, and VIV (vortex-induced vibration) and VIM (vortex-induced motion) virtual test. It is expected that the numerical tank will be able to perform full-scale ship virtual test in real sea environment and full-scale offshore platform motion and loads virtual test in real sea environment. The numerical tank not only supports the digitization and intelligent design for the marine equipment but also forms an important foundation for intelligent ocean exploration and navigation.

Based on our research in China Numerical Tank, some applications are presented about the numerical tank virtual test for ships and offshore structures.

Wave Added Resistance Virtual Test Numerical Tank

In order to serve in the estimation of EEDI whether factor or minimum propulsion power ([Guidelines on the method of calculation of the attained energy efficiency design index \(EEDI\)](#)), the wave added ship resistance virtual test is developed in China Numerical Tank (CNT). The virtual test is developed based on three-dimensional potential flow theory with forward speed in time domain and is solved using innovative Taylor Expansion Boundary Element Method (Duan et al. 2020; Duan et al. 2015a, b), which is therefore named as CNT-TEBEM WAR. The virtual test is integrated with intelligent parameters setting including the panel mesh generalization in free surface and control volume boundary. Its main functions are as follows:

- The wave-induced ship motions and wave added ship resistance in regular/irregular waves.
- The virtual test is suitable for primary large commercial ship types such as container ships, bulk carriers, and oil tankers.

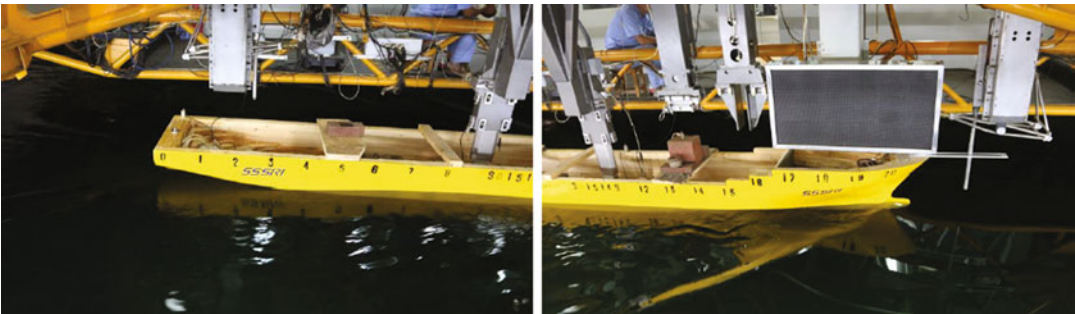
The wave added ship resistance virtual test numerical tank can be operated in popular PC with Windows operating system or High-Performance Computing Cluster. The computational efficiency is quite high, and it takes about only 5 min to finish the wave added resistance computation under one

incident regular wave using popular PC, which is much faster than the viscous flow model with the same accuracy.

In the following, the wave added resistance virtual test is validated with model test and another commercial software viscous flow ISIS-CFD software (Queutey and Visonneau 2007; Theoretical Manual 2011). The model test is performed at Shanghai Ship and Shipping Research Institute (SSSRI) (Figs. 1, 2, 3, 4, 5).

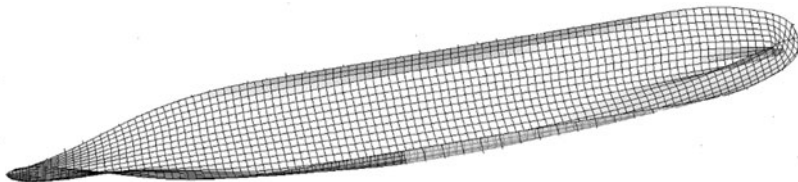
Ship Motions Virtual Test Numerical Tank

In the traditional ship seakeeping physical model test, the wave-induced ship motions are often tested. In the China Numerical Tank, the wave-induced ship motions virtual test is developed. The virtual test is based on the three-dimensional time domain potential flow theory using impulse response function formulation. The matched boundary integral equations are used to solve the fluid potential, and the virtual test is developed by the Taylor Expansion

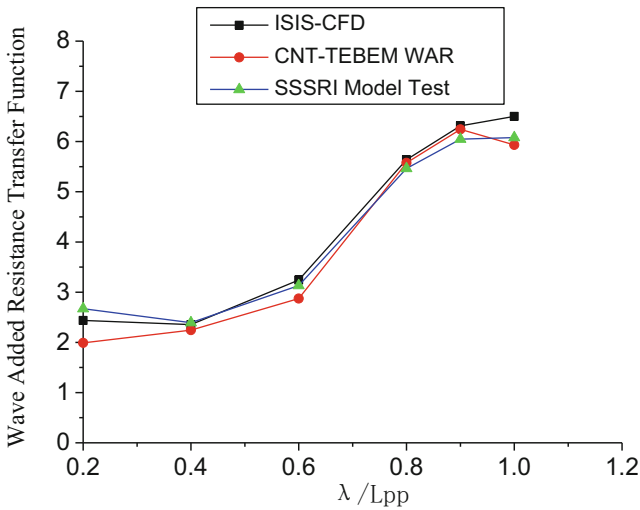


Numerical Tank, Fig. 1 The physical model test of wave added resistance for a 14000TEU container ship performed at SSSRI

Numerical Tank, Fig. 2 The hull panel for a 14000TEU containership

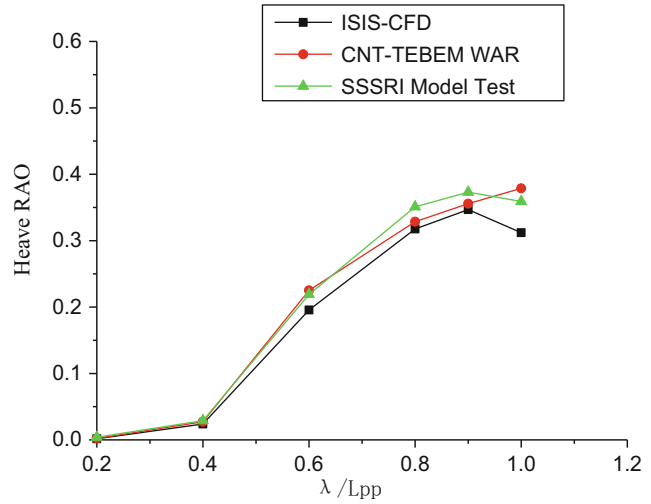


Numerical Tank, Fig. 3 The wave added resistance from China Numerical Tank compared with viscous flow commercial software and physical model test at regular head wave ($F_n = 0.158$)



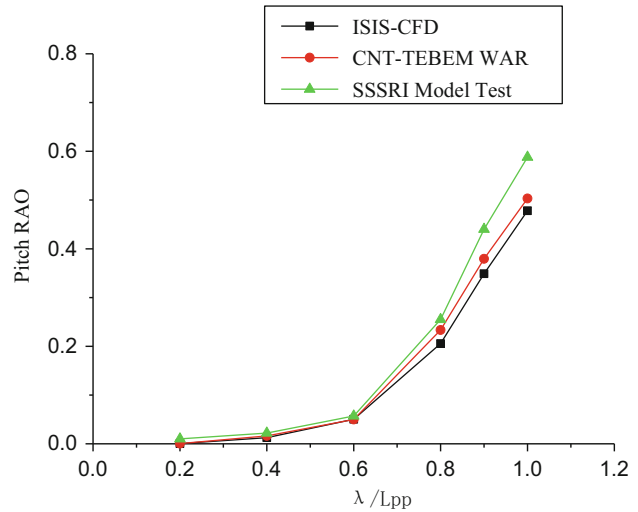
Numerical Tank,

Fig. 4 The heave motion RAO from China Numerical Tank compared with viscous flow commercial software and physical model test at regular head wave ($F_n = 0.158$)



Numerical Tank,

Fig. 5 The pitch motion RAO from China Numerical Tank compared with viscous flow commercial software and physical model test at regular head wave ($F_n = 0.158$)



Boundary Element Method, which is therefore named as CNT-TEBEM WIM. Its main functions are as follows:

- Wave-induced ship motions virtual test at various wave headings in regular waves
- Wave-induced ship motions virtual test at various wave headings in irregular waves
- The probability test of green water, slamming, and propeller emergence virtual test in irregular waves

The computational efficiency is quite high, and it takes only about 30 min to finish one RAO

curve for one wave heading and forward speed using popular PC. In the following, the accuracy of the wave-induced ship motions virtual test is verified with the model test for YUPENG ship, where the model test is held at China Ship Science Research Center (CSSRC). The model test is performed for a self-propulsion ship model at various wave headings (Figs. 6, 7, 8, 9).

Ship Roll Damping Virtual Test

In the traditional ship seakeeping model test, the calm water roll damping test is often tested in order to capture the viscous ship roll damping coefficients in the ship roll motions predictions

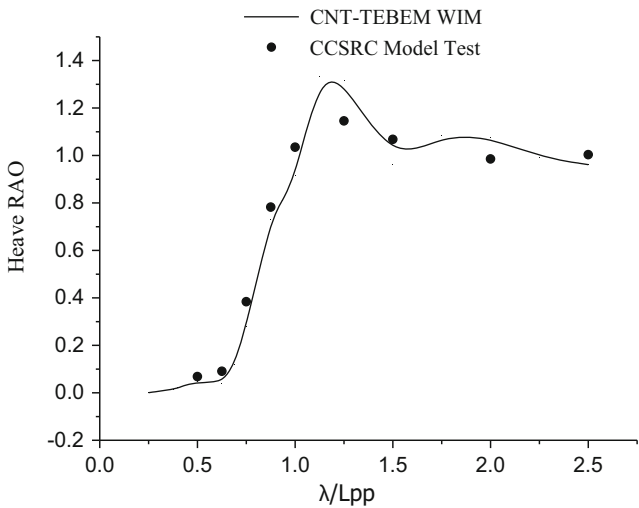
Numerical Tank,

Fig. 6 Ship motions
physical model test in
regular oblique wave at
CSSRC



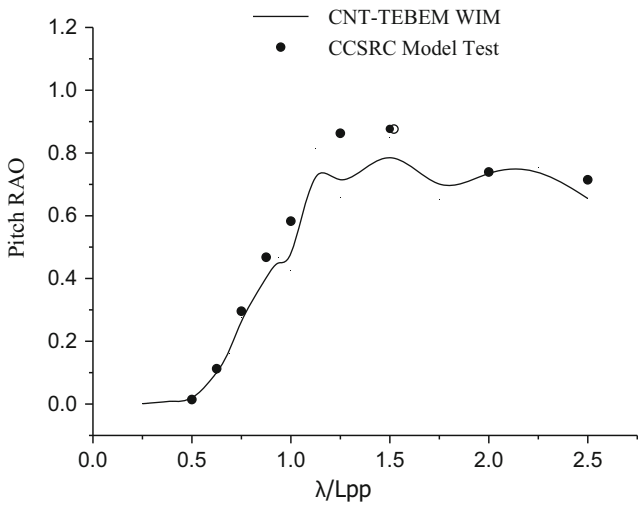
Numerical Tank,

Fig. 7 Heave motion
virtual test compared with
physical model test at
regular oblique 135° wave
heading ($F_n = 0.213$)



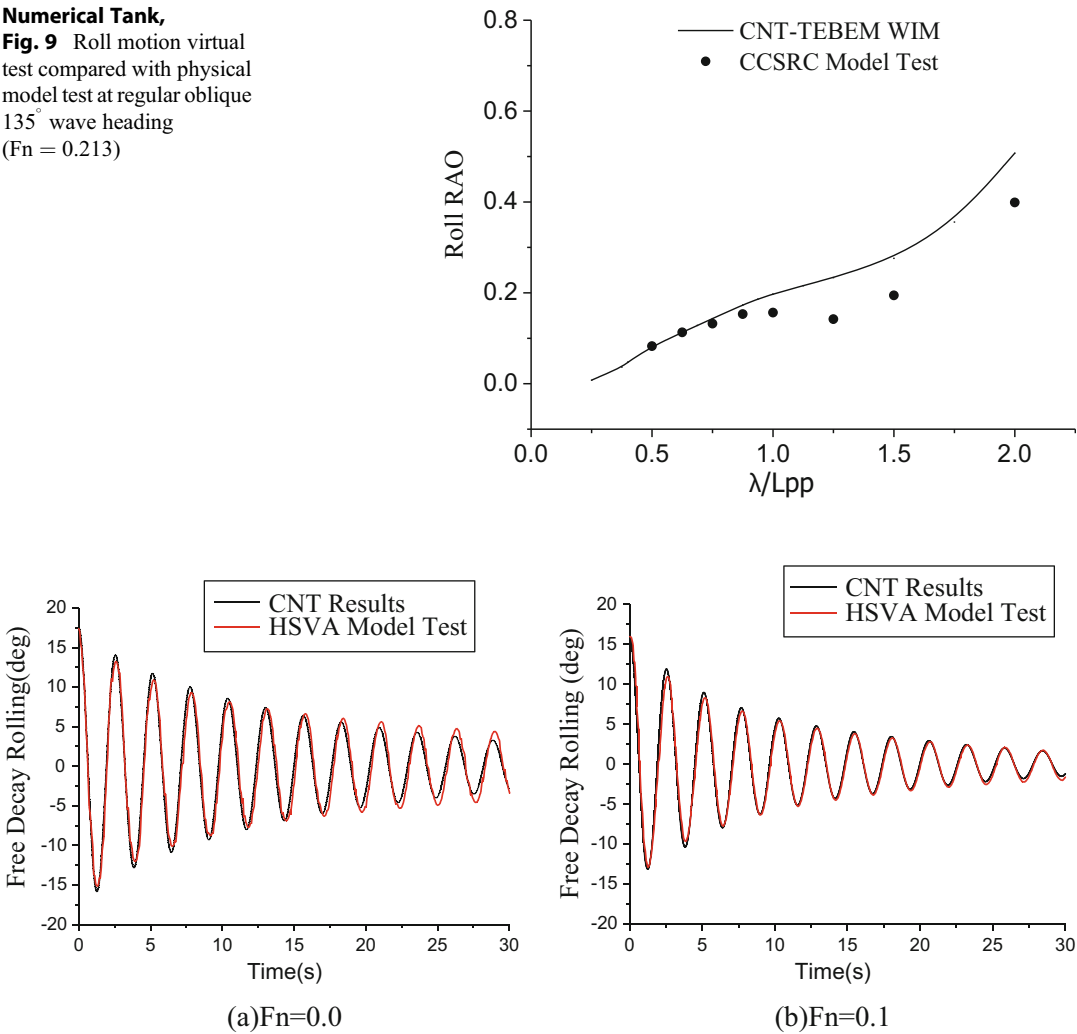
Numerical Tank,

Fig. 8 Pitch motion virtual
test compared with physical
model test at regular oblique
135° wave heading
($F_n = 0.213$)



Numerical Tank,

Fig. 9 Roll motion virtual test compared with physical model test at regular oblique 135° wave heading ($F_n = 0.213$)



Numerical Tank, Fig. 10 Validation of free decay rolling for DTC container ship

using potential flow theory. In the China Numerical Tank, the ship roll damping free decay virtual test is developed.

The virtual test is developed by unstructured body-fitted mesh viscous flow model, applicable for the vortex simulation of ship rolling. The parallel computing algorithm based on MPI is used; the parallel computing can be achieved at high-performance clusters. The automatic generation of the fluid mesh and integration of the intelligent setting for the computational parameters are performed in the development of the virtual test. Its main function is as follows:

- Ship roll damping virtual test by free decay

The virtual test accuracy is validated with the model test results. In the following, the free decay roll virtual test is performed for a Duisburg Test Case (DTC) container ship, and the test results are verified with the physical model test from Hamburg Ship Model Basin (HSVA) (Moctar et al. 2012) (Fig. 10).

Ship Green Water Slamming Virtual Test

In severe sea states, the large-amplitude wave-induced ship motions and slamming are of great

concern for the ship safety. In the China Numerical Tank, the ship green water slamming virtual test is developed and can be used to test the nonlinear ship motions and hydrodynamic slamming loads in bad weather.

The virtual test is developed using large-eddy simulation model and Cartesian grid algorithm (CNT-CGCFD) (Liao et al. 2018). The immersed boundary method is used to model the dynamic boundary on the ship surface. The parallel computing algorithm based on MPI is used; the parallel computing using 100 million grids can be achieved in supercomputing platform. The violent flow and slamming pressure during green water can be well captured.

Its main functions are as follows,

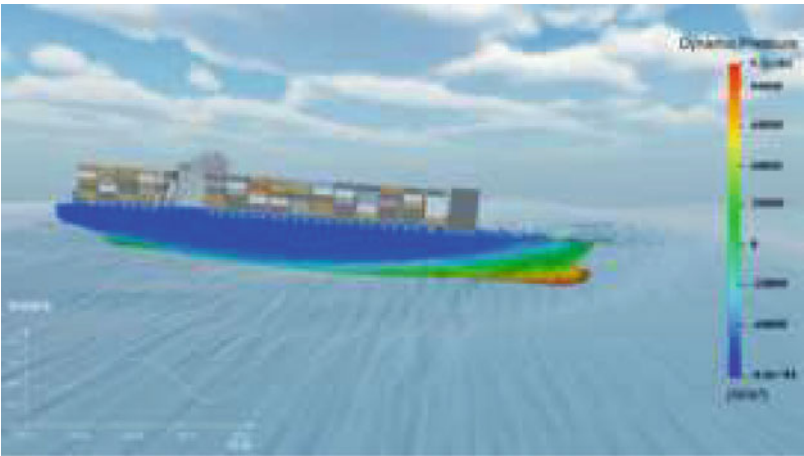
- Large-amplitude ship motions virtual test in regular/irregular waves
- Ship green water slamming virtual test

The virtual test accuracy is validated with the model test results. In the following, the green water virtual test for a 14000 TEU container ship is performed, and the virtual test results are compared with the model test. The model test is performed at Shanghai Ship and Shipping Research Institute (SSSRI) (Figs. 11, 12).

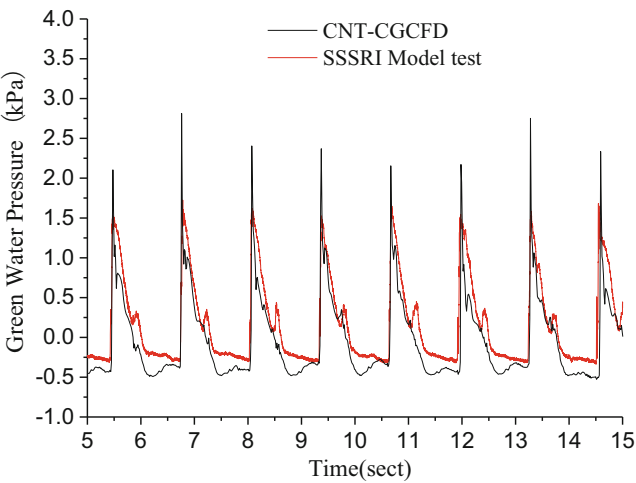
Wave-Induced Offshore Platform Motions Virtual Test

In order to obtain the wave-induced motions and operational limit in different sea states for offshore

Numerical Tank,
Fig. 11 Ship green water and slamming virtual test for a 14,000 TEU container ship

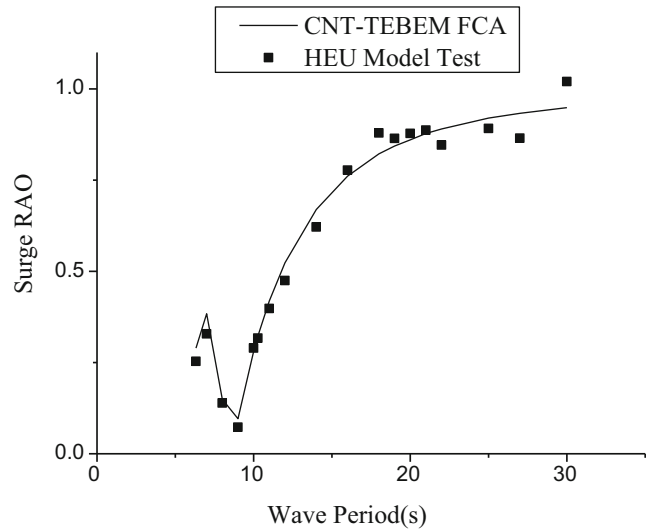


Numerical Tank,
Fig. 12 Validation of dynamic green water slamming loads at one measured point on the deck for a 14,000 TEU container ship model (23 knots, heading seas, $\lambda/L = 1.2$)



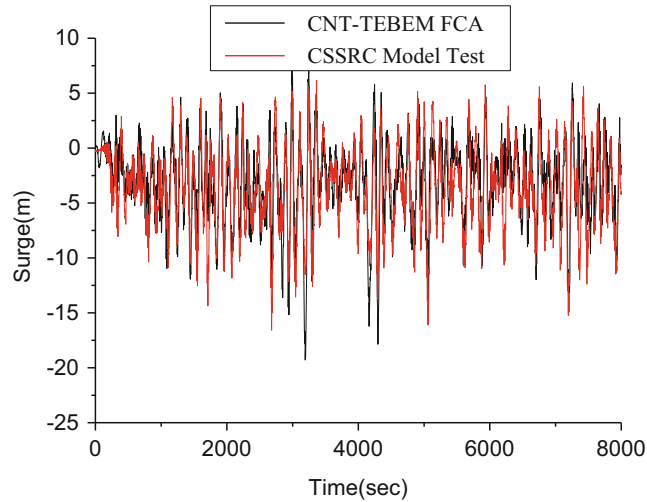
Numerical Tank,

Fig. 13 Validation of wave-induced motion virtual test for a semisubmersible at head regular waves



Numerical Tank,

Fig. 14 Validation of low-frequency moored surge motion virtual test for a semisubmersible at head seas ($H_s = 7.5$ m, $T_p = 11.7$ s)



platforms, the physical model test for the platform in waves is needed to be performed. In the China Numerical Tank, the offshore platform motions virtual test is developed and can be used to test the wave-induced motions for the semisubmersible platform in waves.

The virtual test is developed using three-dimensional potential flow theory. The solver is developed by the Taylor Expansion Boundary Element Method (Chen et al. 2018). Once the hydrodynamic loads have been calculated using three-dimensional potential flow theory, the dynamic coupled analysis for the platform/

moorings/risers can be performed in wind, wave, and current (Ma et al. 2015). In order to consider the viscous load contribution, semiempirical Morison's formula is used in the estimation of the hydrodynamic loads from the slender structural members in the platform. The virtual test is named as CNT-TEBEM FCA.

Its main functions are as follows:

- First-order wave-induced motions RAO test
- Low-frequency horizontal motions and mooring line tensions for the moored platform in combined wind, wave, and current

The virtual test accuracy is validated with the model test results. In the following, the physical model test for the wave-induced motions RAO and the low-frequency moored surge motion from a semisubmersible is performed at Harbin Engineering University (HEU) (Fig. 13) and China Ship Science Research Center (CSSRC) (Fig. 14).

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Numerically Controlled Oscillator (NCO)

► Integrated Navigation

O

Obstacle Avoidance Technology for Underwater Vehicle

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Synonyms

Alarm pheromone-assisted ant colony system (AP-ACS); Ant colony system (ACS); Constructive solid geometry (CSG); Cooperative coevolutionary artificial bee colony algorithm and Tabu search (CCTS-ABCA); Differential evolution (DE); Doppler velocity log (DVL); Dynamic segment-based predominant learning swarm optimizer (DSPLSO); Inertial navigation sensors (INS); Long-baseline system (LBL); Obstacle avoidance technology, Obstacle avoidance method, Obstacle avoidance system; Super-short-baseline system (SSBL); Underwater vehicle, Underwater robot, UUV, ROV, HOV

Definition

Obstacle avoidance technology is a control strategy or method that the underwater vehicle system must have. It enables the underwater vehicle to search and confirm static or dynamic obstacles on its forward path and can actively take avoidance

measures to avoid collisions with obstacles and finally restore to the set path to continue to perform the specified task.

Scientific Fundamentals

With the advancement of technology, underwater vehicles have gradually become one of the most important tools for understanding and developing the ocean, which has greatly promoted the development of marine economy and related technologies; underwater vehicles are widely used in seabed topography in complex environments. At the same time, they provide effective technical support and guarantees for research in the fields of underwater ecology, underwater archaeology, and underwater geoscience. In order to enable the underwater vehicle to adapt to the complex and changeable marine environment and complete underwater detection tasks quickly, efficiently, and autonomously, it is required that the underwater vehicle must have dynamic route planning and autonomous obstacle avoidance functions. In the specified operating area, the underwater vehicles can seek a continuous path through all reachable points from the starting point of the operation to the target end, on the premise of effectively avoiding unknown obstacles.

General Underwater Vehicle

The main missions of the underwater vehicles are geological and biological surveys of the seabed

and water column, including benthic and pelagic missions which could last up to a specified period of time. The need for an automated piloting system, or at least an aided piloting system, is clear. The cruising speed for long working underwater vehicles is around 1 km, and their movements are measured by several onboard sensors feeding the obstacle avoidance system with the position, speed, and orientation of the vehicle in world coordinates. The positioning system will consist of a long-baseline system (LBL), a super-short-baseline system (SSBL), and of various inertial navigation sensors (INS), depending on the vehicle such as compasses, gyro-meters, and Doppler velocity log (DVL). Despite the different depths at which underwater vehicles work, the baselines systems are likely to be imprecise, and INS sensors will drift with time; the obstacle avoidance module does not require long-term accuracy as it is creating a local map of the environment. The drift of the INS sensors should be negligible on short periods and ensure the successful creation of a precise local map. Due to the particularity of the underwater environment, sonars have become the most practical detection equipment used by underwater vehicles.

Obstacle Detection Method

When addressing the problem of obstacle avoidance and path planning, the main problem encountered is the extraction of information from the sonar images to create a representation of the environment and can be interpreted in terms that are suitable for computation.

In the underwater vehicle case, the first aim is to detect and avoid obstacles. Therefore, the segmentation of the scene is the first and crucial task. It is also useful to know how these obstacles are moving with respect to the vehicle in order to take movement into account in the obstacle avoidance/path planning process. Once the obstacles' static and dynamic characteristics have been computed from the input data flow, a workspace of obstacles surrounding the vehicle is created. This workspace uses constructive solid geometry (CSG) to create a convex representation of the obstacles, which eases convergence of the path planning algorithm. The map of obstacles

surrounding the vehicle takes into account the obstacles currently in the field of view of the vehicle but also obstacles that have gone out of the field of view, which have been tracked in the near past, and whose position is still critical to the definition of a safe path for the vehicle (Petillot et al. 2001).

The obstacle avoidance system for underwater vehicles that can be designed (see Fig. 1) is modular in nature. Modularity is seen as a way to handle different needs within the same framework.

Segmentation Module

The purpose of this module is to identify the interesting regions of the image-containing obstacles. Considering the very nature of multibeam sonar images, the certainty grid approach is discarded, and the object-oriented description of the workspace is focused on. As the vehicle is moving close to the seabed, high backscatter seabed returns are expected. This backscatter must be estimated and removed when possible in the segmentation process.

Feature Extraction Module

After segmenting the image, potential obstacles and their characteristics (location, size, and area) will be calculated. These features are used to determine whether there is an obstacle in front of the underwater vehicle, and to identify whether the obstacle has been detected before. These functions will later be used to discard false alarms and track obstacles and other vehicles.

Tracking Module

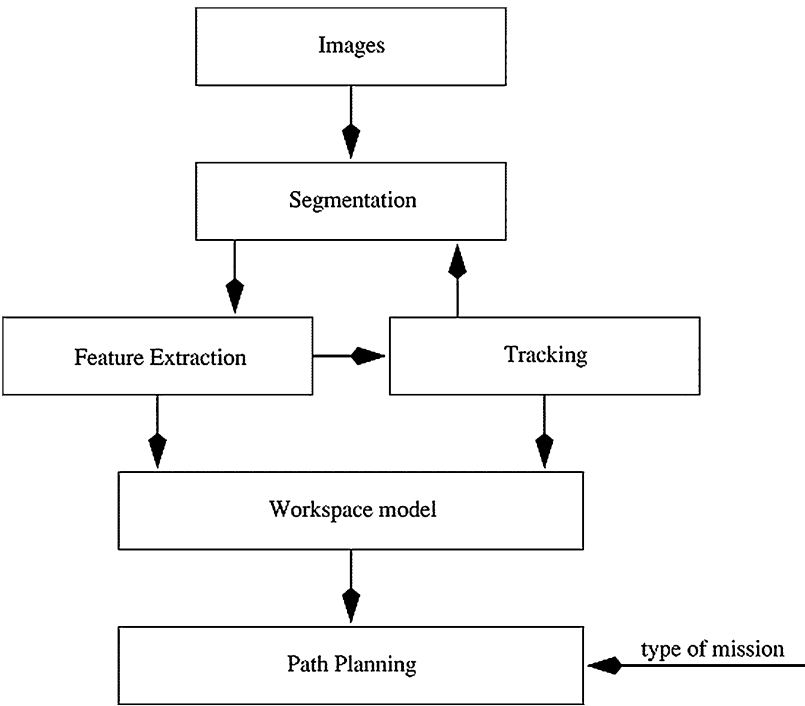
Obstacles can be divided into two types, namely static obstacles and dynamic obstacles. This division method is mainly to facilitate the use of different obstacle avoidance strategies later. The tracking module will provide dynamic characteristics and models of obstacles according to the requirements of the obstacle avoidance system.

Workspace Representation

From obstacles and features extracted from the current image, a symbolic representation of the

Obstacle Avoidance Technology for Underwater Vehicle,

Fig. 1 Architecture of sonar-based, real-time obstacle avoidance and path planning. (Petillot et al. 2001)



vehicle’s surroundings is built outright. Combining this representation with previous instances of the vehicle’s environment, a dynamic workspace is built and constantly updated. It forms the basis for the path planning algorithm. In this workspace, each object is represented by its current position, shape, and estimated velocity. The object’s shape is assumed to be elliptic, and real objects are represented as ellipses. This workspace can be seen as a local map of symbolic objects with their associated estimated static (shape, position) and dynamic (velocity) properties.

Path Planning Module

Various methods based on the CSG representation of obstacles are used for path planning. Each obstacle in the workspace is expressed as a constraint that must be met in the search space (that is, the path must not pass through the obstacle) while minimizing the Euclidean distance from the start point to the target point. The specific method will be introduced in the next section.

Obstacle Avoidance and Path Planning Method

Obstacle avoidance and path planning methods can be divided into two types: global path planning methods and dynamic path planning methods. The global path planning method is suitable for the situation where the global map is known and the obstacles are usually static obstacles. Therefore, from the perspective of practical significance, the dynamic path planning method is more practical.

Global Path Planning Method

Global path planning is a classic optimization problem that can be solved by many optimization algorithms. The complexity of three-dimensional (3D) path planning for autonomous underwater vehicles requires the optimization algorithm to have a quick convergence speed.

Liu et al. (2015) provide a new 3D path planning method for AUV using a modified firefly algorithm. In order to solve the problem of slow convergence of the basic firefly algorithm, an improved method was proposed. In the modified

firefly algorithm, the parameters of the algorithm and the random movement steps can be adjusted according to the operating process. At the same time, an autonomous flight strategy is introduced to avoid instances of invalid flight. An excluding operator was used to improve the effect of obstacle avoidance, and a contracting operator was used to enhance the convergence speed and the smoothness of the path.

Zhang (2006) proposed a genetic algorithm-based hierarchical global path planning method for Autonomous Underwater Vehicles in ocean. The method bases on cell decomposition approach, but it only decomposes the cells which are used in the path planning, so the method can save the memory effectively for the global path planning for AUV in ocean. The genetic algorithm is applied to find the path at each level. Results of the computer simulation are given to show the feasibility of the proposed algorithm.

Dynamic Path Planning Method

The dynamic path planning and obstacle avoidance control algorithm are based on the established ocean map model, and searches for the best reachable path between the planned start point and the target end point under the constraints of sailing time, smooth path, and lowest energy consumption. In the past few years, AUV path planning research has achieved fruitful results, which are mainly divided into four categories: geometric model search algorithm, probability sampling algorithm, artificial potential field algorithm, and artificial intelligence algorithm.

The geometric model search algorithm is a classic path search algorithm, which belongs to the category of discrete optimization planning. The realization process is simple, and the model establishment is strict. It mainly includes Dijkstra algorithm (Solka et al. 1995), A* algorithm (Duchon et al. 2014), D* algorithm (Ferguson and Stentz 2005), D*Lite algorithm (Al-Mutib et al. 2011), FM algorithm (Petres et al. 2005), etc. Dijkstra algorithm uses a greedy strategy, using weighted directed graphs or undirected graphs to solve the single-source shortest path problem. A* algorithm is an efficient search algorithm for the shortest path in a static map. D*

algorithm is a dynamic A* algorithm that can be effective that deals with the path search problem in dynamic environment. FM algorithm has good reliability and convergence and can meet real-time requirements.

Probability sampling algorithms mainly include PRM method (Bohlin and Kavraki 2000) and RRT method (Karaman et al. 2011). The core idea of PRM method is pose space sampling and path search, which can effectively avoid obstacles while achieving accurate pose space modeling. The advantage of RRT algorithm lies in its efficient spatial search ability. The expected distance between the random state point and the tree can be quickly shortened.

The artificial potential field method (Chen et al. 2016) is widely used in the field of real-time obstacle avoidance. It has low requirements for environmental complexity, small calculation volume, and fast search speed, but it needs to integrate other algorithms to enhance the global optimization capability.

The artificial intelligence algorithm is a current research hotspot in path planning. Researchers have evolved a variety of algorithm forms from different perspectives of intelligence (Gigras and Gupta 2012), including particle swarm optimization, ant colony optimization, wolf swarm algorithm, simulated annealing algorithm, genetic algorithm, differential evolution algorithm, artificial nerve Network, etc. The artificial intelligence algorithms play an important role in promoting the solution of AUV path planning in complex and dynamic environments. Compared with other algorithms, they have shown great potential, although they need to be further improved in terms of real-time, robustness, and rigorous theoretical proof. It is still the mainstream direction of future research in this field.

Key Applications

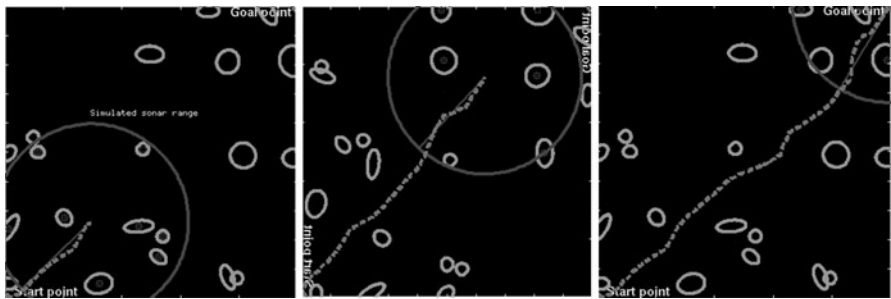
Petillot et al. (2001) have presented here a general framework for performing 2-D obstacle avoidance and path planning for underwater vehicles based on a multibeam forward-looking sonar sensor. The ability of the system has been

demonstrated on real sonar data. The sequence used corresponds to a real trial, and the ability of the system to perform obstacle avoidance is demonstrated. The path planning system generates very smooth paths, can handle complex and changing workspaces, and presents no local minima as it uses a convex representation for the obstacles.

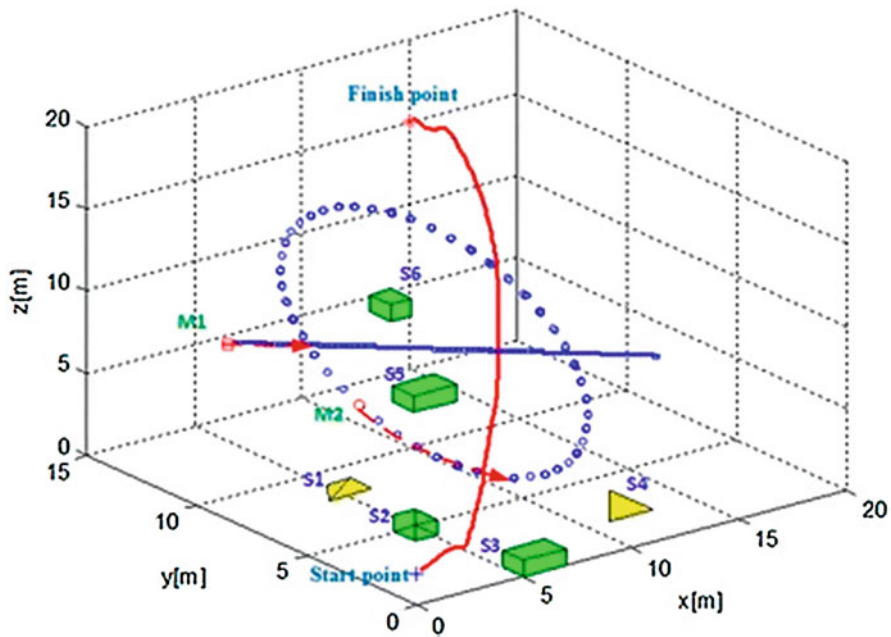
Figure 2 shows the simulation results showing the performance of their proposed algorithm. The starting point is the bottom left corner while the goal point is the top right corner. The vehicle

speed is limited to 5 m/s. The simulated sonar's field of view is defined by the circle. The objects in the field of view and currently used for the path planning process are denoted with crosses (+ sign) while the ones that will be used in the next iteration of the path planning algorithm are denoted with squares.

Sun et al. (2018) propose an optimized fuzzy control algorithm for three-dimensional AUV path planning. Based on the sonar model, it can be used for path planning in the complex underwater environment. First, the path planning is



Obstacle Avoidance Technology for Underwater Vehicle, Fig. 2 Example of a generated path on a simulated workspace. (Petillot et al. 2001)

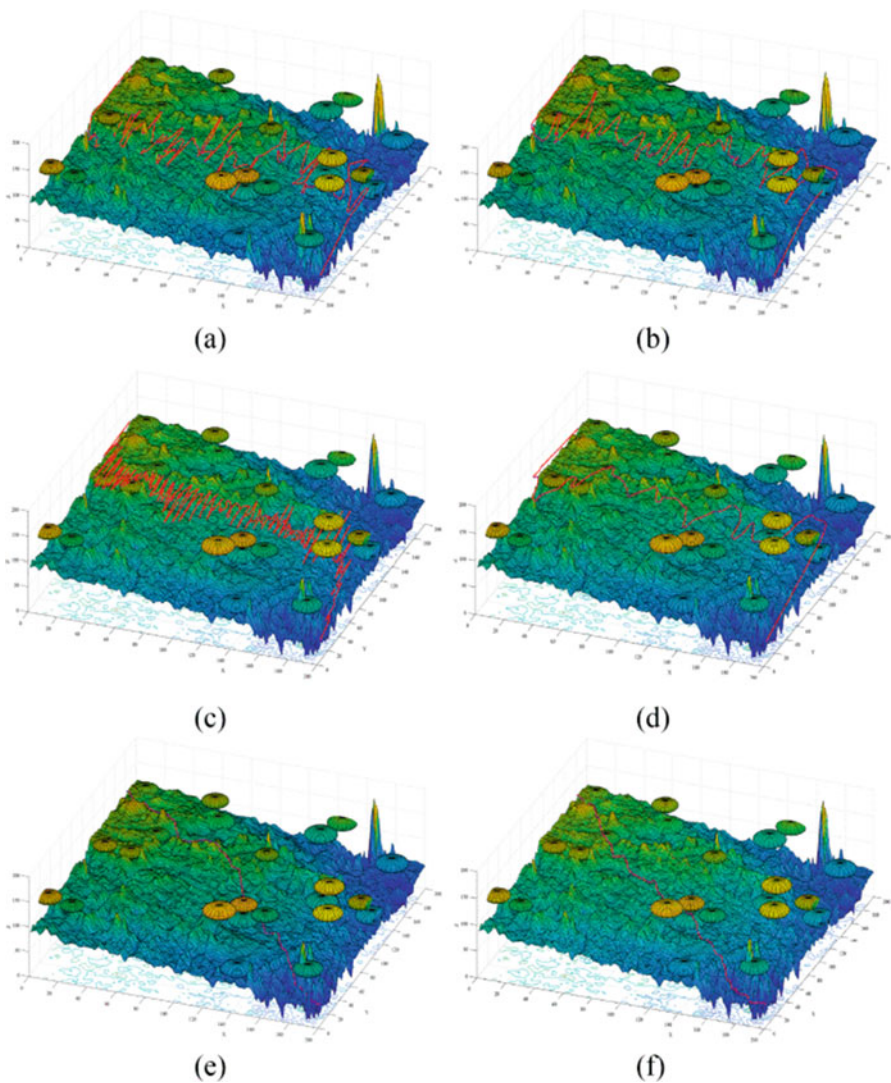


Obstacle Avoidance Technology for Underwater Vehicle, Fig. 3 3D dynamic fuzzy path planning without optimization. (Sun et al. 2018)

established based on the two sonars arranged on the horizontal plane and vertical plane. On the environment information collected by sonars, the virtual acceleration and velocity of AUV in the three-dimensional space can be obtained through the fuzzy system with the accelerate/brake module which enables AUV to avoid dynamic obstacles automatically. Considering the fuzzy boundary choice has great subjectivity, the generated path cannot guarantee to be optimal. Therefore, in order to deal with the optimal path

problem, two optimized methods are compared to do the optimization of the fuzzy set. Simulation results (shown as Fig. 3) indicate that the proposed method can generate an optimal 3D path under the three-dimensional underwater environment (Sun et al. 2018).

Ma et al. (2018) improve the path planning of AUVs in terms of both the path planning model and the optimization algorithm. The proposed model is comprehensive, which aggregates the length, energy consumption, and collision risk



Obstacle Avoidance Technology for Underwater Vehicle, Fig. 4 Visualization of the paths produced by the six evolutionary algorithms in the East China Sea. (a)

The original DE, (b) Self-adaptive DE, (c) CCTS-ABCA, (d) DSPLSO, (e) The original ACS, (f) AP-ACS. (Ma et al. 2018)

into the objective function and incorporates the steering window constraint. Based on the model, they develop a nature-inspired ant colony optimization algorithm to search the optimal path. Our algorithm is named alarm pheromone-assisted ant colony system (AP-ACS) since it incorporates the alarm pheromone in addition to the traditional guiding pheromone. The alarm pheromone alerts the ants to infeasible areas, which saves invalid search efforts and, thus, improves the search efficiency. Meanwhile, three heuristic measures are specifically designed to provide additional knowledge to the ants for path planning. In Fig. 4, the comparison between the AP-ACS and the simulation results of the other five path planning methods is shown. The experimental results show that AP-ACS can effectively handle the constraints and outperform the other algorithms in terms of accuracy, efficiency, and stability (Ma et al. 2018).

Cross-References

- [Obstacle Avoidance Technology for Underwater Vehicle](#)

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Obstacle Avoidance Technology, Obstacle Avoidance Method, Obstacle Avoidance System

- [Obstacle Avoidance Technology for Underwater Vehicle](#)

Ocean Data Acquisition System

- [Wave Measurement Buoy](#)

Ocean Mining

- [Deep Sea Mining](#)

Ocean Thermal Energy Conversion

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Definition

Ocean thermal energy conversion (OTEC) is a technology that uses the temperature difference between cooler deep water and warmer shallow or surface seawater to run a heat engine to produce energy, typically electricity.

Ocean Thermal Energy Resource

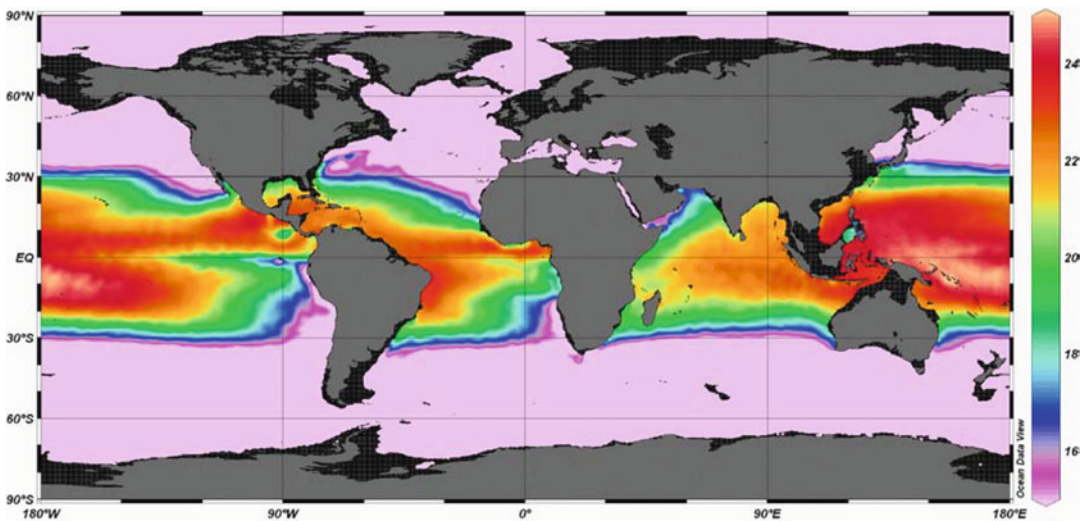
The ocean thermal energy resource depends strongly on the temperature difference in the ocean and the volume of the seawater in the circulation of the OTEC device for electricity generation. Fig. 1 shows the global distribution of the seawater temperature difference near the surface (20 m below the surface) and at the bottom (1000 m below the surface) of the oceans

(Nihous 2010). As we can see, areas with large temperature difference, in the order of 20–25 °C, are in the tropics around the equator and have the potential to develop OTEC devices. In such areas, the surface water temperature can exceed 25 °C, while the sea temperature at a depth of 1000 m is in the range of 5–10 °C. A minimum 20 °C of temperature difference is needed to run an OTEC cost-effectively (Lewis et al. 2011). The ocean thermal energy resource is estimated to be about 88,000 TWh/year, which is much larger than other ocean energy forms (Pelc and Fujita 2002).

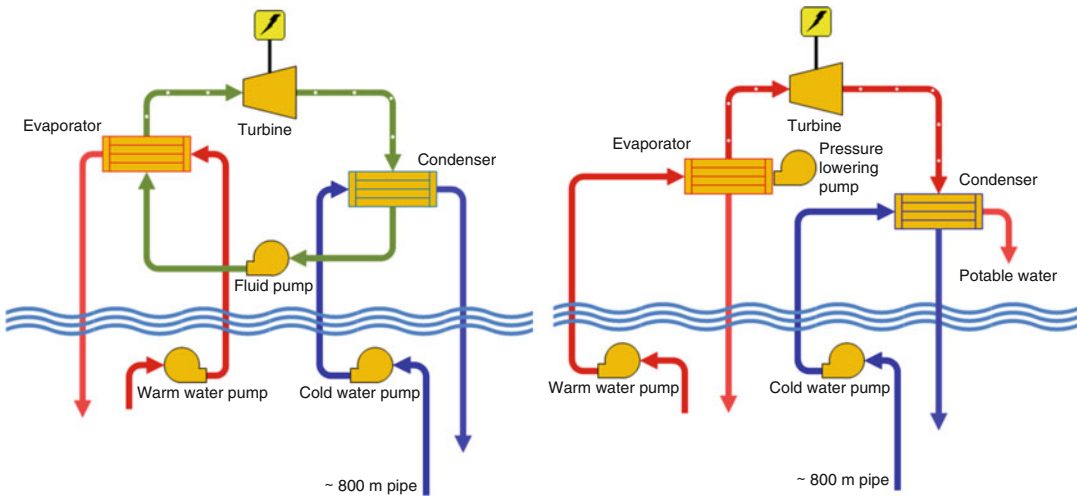
Ocean Thermal Energy Conversion Technology

As compared to other offshore renewable energy technology, such as offshore wind, the OTEC technology is still in the early phase of development. Basically, there are two types of OTEC devices, as shown in Fig. 2 (OTEC Africa 2019). In some situation, a hybrid plant can be used (Charlier and Justus 1993).

In a closed-cycle OTEC plant, the warm seawater is pumped through a heat exchanger (evaporator) to evaporate a secondary working



Ocean Thermal Energy Conversion, Fig. 1 Global distribution of average seawater temperature differences (between 20 m and 1000 m water depths) from World Ocean Atlas 2005 data (Nihous 2010)



Ocean Thermal Energy Conversion, Fig. 2 Illustration of a closed-cycle (left) and an open-cycle (right) OTEC plant (OTEC Africa 2019)

fluid (such as ammonia, propane) in a high-pressure condition to drive a turbine to generate electricity. After that, the vapor is cooled down by the pumped cold seawater through another heat exchanger (condenser) to complete a cycle of fluid-vapor-fluid transform. Since the turbine working fluid is under high pressure, a high turbine efficiency for energy conversion can be achieved. The originally warm and cold water are then discharged back to the sea. In an open-cycle OTEC plant, 1% of the warm seawater is evaporated directly in a vacuum chamber, and the vapor will drive a turbine for electricity generation. After that, the vapor is cooled down by the cold seawater as in a closed-cycle OTEC plant, but it will not return to the circulation. In fact, the vaporized warm seawater is also desalinated due to this process and can be used as fresh water. The originally cold water and most of the warm water will be discharged back to the sea.

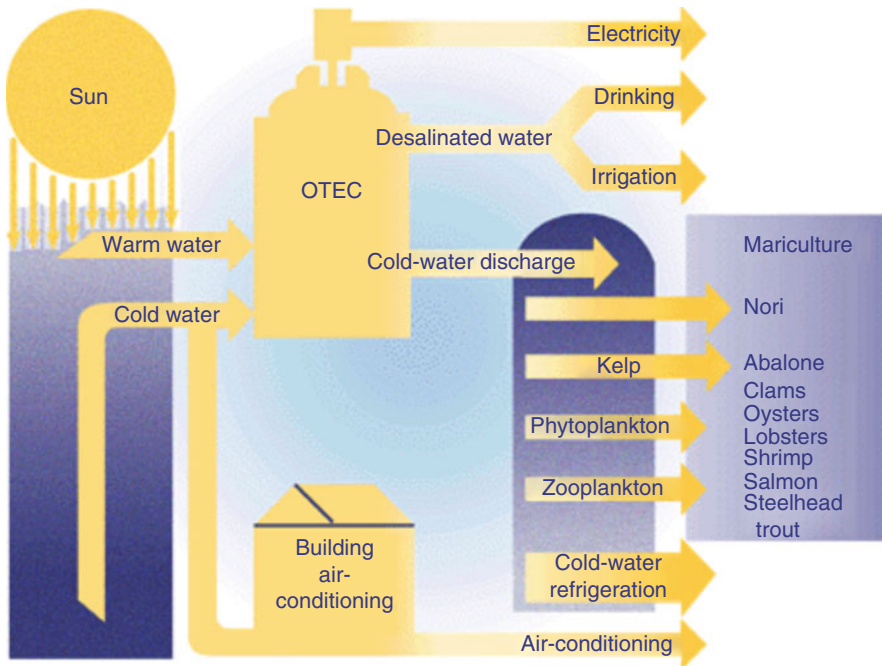
It should be noted that the change in temperature of the warm and the cold water in an OTEC plant is not significant, in the order of a few degrees Celsius, and therefore a large amount of water needs to be pumped in order to realize enough electricity generation. The whole process of an OTEC, as a heat engine,

has a thermodynamic efficiency of 1–3% (Wikipedia 2019).

Technological Development and Challenges

There are no commercial OTEC plants for electricity generation yet. A few prototypes have been tested (Kobayashi et al. 2004). Japan, followed by the USA and India, has been the main contributor to the development of OTEC technology since 1970s. A closed-cycle OTEC plant became operational in 1981 on the island of Nauru in Japan, producing 120 kW of electricity, with 90 kW used to power the plant and 30 kW to power one school and other places. In 1993, a 200-kW OTEC plant was completed in Hawaii. In 2002, India tested a 1-MW floating OTEC prototype but had a failure in the deep-sea cold-water pipe. From 2006 to 2013, Makai Ocean Engineering developed an OTEC Heat Exchanger Test Facility with a 100-kW turbine at the Natural Energy Laboratory of Hawaii.

In addition to the issues related to maintenance of the vacuum and biofouling and corrosion of heat exchangers, the big challenge for an OTEC is related to the cold-water pipe (Lewis



Ocean Thermal Energy Conversion, Fig. 3 Principle of an OTEC plant and its by-products (Wikipedia 2019)

et al. 2011). OTEC plant can be shore-based or floating at the sea. In all cases, a long pipe is needed to reach the cold water in deep sea, in the order of 1000 m deep. The long pipe will be subjected to current and wave loads, which makes cost higher and design more challenging, especially for a floating OTEC plant.

As of today, it is still too expensive to use an OTEC for commercial electricity generation. However, there are many by-products that an OTEC can produce, including desalinated water for drinking or irrigation and cold water for mariculture, refrigeration, and air-conditioning, as shown in Fig. 3 (Wikipedia 2019). An OTEC plan might be used to produce hydrogen, lithium, and other rare elements. These can improve the economic feasibility of OTEC plant.

Cross-References

- [Economic Assessment](#)
- [Power Take-Off System](#)
- [Resource Assessment](#)

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OCTM – Ore Collecting Test Machine

- Collector and Crusher

ODAS

- Wave Measurement Buoy

Offshore Engineering Vessel

- Offshore Vessel

Offshore Fish Farm

- Traditional Aquaculture Structures

Offshore Floating Module

- Drag Anchors

Offshore In Situ Penetrometers

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Definition

The offshore in-situ penetrometer is a specially designed device suitable for geotechnical site investigation. It is commonly used in the in-situ testing such as penetration and shearing testing, aiming to characterize the soil properties, thus

capturing the engineering parameters for practice design.

Classification According to Probe Geometry

Cone Penetrometers

The cone penetrometer (Fig. 1a), particularly piezocone, is now regarded the most performed offshore test (occupying more than 95%) during a geotechnical investigation (Lunne 2001). The standard piezocone is a cylindrical penetrometer with a 35.7 mm diameter (projected area of 1000 mm²) 60° conical tip. However, for the seabed site investigation, cone penetrometer with a 43.8 mm conical diameter (projected area of 1500 mm²) is increasingly used. In the area where the properties of shallow sediments are important, cone penetrometers with an 11.3 mm conical diameter (projected area of 100 mm²) are also deployed to provide a sensitive measurement of the upper few meters of the seabed (Lunne 2001). A load cell is attached to the conical tip, measuring the penetration resistance, which is normally termed as q_c (kPa). A sleeve is mounted above the tip and a load cell attached to this sleeve, measuring the side friction termed as f_s (kPa). The importance of penetration pore pressure measurement for the interpretation of cone penetration test (CPTU) has been increasingly acknowledged, and therefore almost all offshore cone penetrometers allow the simultaneous measurement of pore pressure, cone resistance, and sleeve friction. This means the pore pressure transducers are mounted at least at one position: face of the cone (so-called u_1 position), on the shoulder (just above the tip – u_2) or above the sleeve (u_3).

The cone penetrometer provides a continuous measurement of soil profile with depth, and therefore is extremely useful in detecting the fine change in soil stratigraphy and soil behavioral classification (Robertson 1990). A number of correlations of the soil parameters and measurement have also been established for different types of soil (e.g., the normalized cone response with the relative density D_r for silica sands; the net cone



Offshore In Situ Penetrometers, Fig. 1 Typical penetrometers used offshore

resistance with the shear strength profile by means of a cone factor N_{kt} for fine-grained sediments).

Due to the high ambient water pressure, a high capacity load cell is often required. However, this affects the measurement sensitivity in detecting small changes in resistance in soft sediments

from CPTU. Also, the measurement accuracy is hindered due to the required measurement corrections on the cone resistance for the unequal area effect and overburden stress. These limitations have led to the introduction of the full-flow penetrometers to provide reliable soft soil characterization at deep water sites.

Full-Flow T-Bar and Ball Penetrometers

The T-bar penetrometer (Fig. 1b) introduced at the University of Western Australia was the first full-flow penetrometer, which consists of a cylindrical bar attached to the penetrometer rods. It was originally designed to improve the accuracy of strength profiling in centrifuge modeling (Stewart and Randolph 1994), and was later scaled up to be used offshore in the late 1990s (Randolph et al. 1998). For field use, a standard T-bar dimensions of 40 mm diameter by 250 mm long is recommended. And the cylinder should have a minimum length to diameter ratio of 4, and the connecting shaft should be no more than 15% of the projected area of the T-bar. With the large protrusion of the cylindrical bar compared to the shaft, the T-bar penetrometer is more susceptible to eccentric loading. The piezoball (Fig. 1c) penetrometer has been therefore proposed. The piezoball penetrometer normally includes a ball with a diameter varying between 50 and 120 mm, with the shaft (no more than 15% of the projected area of the ball) mounted just behind the ball. Pore pressure sensors are mounted either at the mid-depth or at discrete points on the lower half of the ball.

The main advantages of the full-flow penetrometers over the cone penetrometer include (Randolph et al. 2007): (1) the load cell measures a net pressure on the bar, and therefore no correction to the effect of overburden stress is required; (2) the projected area of the full-flow penetrometer is normally 10 times of the cone, and this reduces the sensitivity to any load cell drift if the same load cell is used; (3) the exact plasticity solution has been presented to correlate the measured net pressure with the shear strength; (4) the remolded shear strength can be assessed using cyclic penetration and extraction test over short depth intervals (typically less than 0.5 m).

Vane Shear

The vane shear is conventional in situ test, as to measure the peak and residual shear strength for soft sediments. Generally, three different sizes of vane are used offshore, all with a height to width ratio of two and with the height ranging between

80 mm and 130 mm (NORSOK Standard 2004). The size of vane is chosen according to the localized strength of the soil to be investigated. In offshore condition, the measurement to the residual shear strength using vane shear is less attempting than that under onshore condition, as the offshore downhole tool or seabed frame limits the upper rotation speed and angle, which affects the robustness of the measurement.

Specialized Penetrometers

Lunne (2001) summarized a number of more specialized tests that may be undertaken, which includes:

- Seismic piezocone tests, to obtain profiles of small strain shear modulus G_0
- Cone pressuremeter tests, with a cylindrical expansion pressuremeter situated just behind the cone, with the primary aim of measuring unload–reload shear modulus
- Electrical resistivity (in sand) and nuclear density probes, both of which provide an estimate of the soil density
- Piezometer probe (or piezoprobe), to determine the ambient pore pressure

More specific tests are also undertaken aimed at a particular foundation or pipeline option in order to validate design parameters or a construction procedure, which include:

- Plate load tests, generally conducted at the seabed
- Hydraulic fracture tests
- Open hole stability tests
- Grouted section tests
- SMARTPIPE facility
- Toroid and hemi-ball penetrometers

Installation and Testing Technology

Work Platform

The platform is most expensive component in the field of site investigation. The selection of the appropriate platform relies on the careful weighing up between the economy efficiency

and the working ability. The recommendation in the different water depths at the site include:

1. Diver operated from a support vessel or barge (water depth < 20 m)
2. Jack-up rig (20 m < water depth < 200 m)
3. Remote equipment from a support vessel (all water depths, depending on vessel draft)
4. Anchored drilling vessel (20 m < water depth < 200 m)
5. Specialized dynamically positioned (DP) vessels (for deepwater locations)
6. Remotely operated equipment, supported by construction vessels or small DP vessels

Installation Type

The geotechnical site investigation is performed by a specialized penetrometer from a purpose-built system (a downhole system and alternatively, a seabed system).

Both systems can be used for CPT, T-bar, ball penetrometers, and vane shear. But the seabed system intrudes less investigation depth than the downhole system and is also disadvantaged when a hard layer is present. Therefore, the seabed system is more appropriate for the deepwater site investigation, where the sediments within the depth up to 20–50 m is most of interest.

Testing Procedure

Conducting cone penetration tests in the offshore condition follows the International Reference Test Procedure (IRTP) published by the International Society for Soil Mechanics and Geotechnical Engineering (ISSMGE) (1999) and various national and international standards such as NORSOK Standard (2004), ASTM D5778-07 (2007), and ENISO 22476-1 (2007). The standard rate of penetration is 20 mm/s, or approximately half a diameter per second. Lunne et al. (1997) provides a detailed introduction about the cone penetration testing, including both equipment and interpretation.

Standard practice for full-flow penetrometers in the site investigation has not been published, but the following principles are generally agreed:

The penetration and extraction rate should be in the range of 0.2–0.5 diameters per second, and at least one cyclic penetration and extraction test is

recommended within each penetration test in order to correct the load cell offset if necessary, and assess the penetration resistance for remolded conditions. Specific tests may also be involved, using different rates of penetration in order to evaluate strain-rate and consolidation characteristics of penetration resistance.

The vane is thrust into the soil by a minimum distance of 0.5 m and rotated at a speed of 0.1 or 0.2 °/s, and additional tests may be required by pushing the vane deeper recommended by at least 0.5 m. Under the onshore condition, the remolded shear strength can be achieved by rotating the vane rapidly for ten revolutions, and then reverted to the original test speed to obtain a remolded strength. However, the offshore equipment involved a down-hole tool or a seabed frame, thus limiting the upper speed to less than 1°/s. A test of 20 min is typical to obtain the peak strength and remolded strength (Peuchen and Mayne 2007). The vane is competent in giving a reliable assessment of the localised shear strength. However, it is unable to present a continuous profiling of the site, or providing information about soil classification. It is therefore less attempting to use vane shear test to assess the resistance compared to the other penetrometers.

Assessment of Soil Properties

Soil Type Characterization

Measurement of the friction from the cone penetrometer along the sleeve and the pore pressure close to the tip provides information from which the type of soil may be assessed through correlations. Normalized parameters are used for such classification, principally:

- Normalized cone tip resistance, $Q = q_{\text{net}}/\sigma'_{v0}$
- Excess pore pressure ratio, $B_q = \Delta u_2/q_{\text{net}}$
- Friction ratio, $F_r = f_s/q_{\text{net}}$

Generally, the classification charts show that stronger soils (dense sands or cemented sediments) correspond to higher normalized cone resistance, Q , and low friction ratio, F_r , compared with clays (Robertson 1990). The data may be distorted due to the effects of increasing over-

consolidation ratio or increasing degree of drainage for silty soils, which both lead to increasing Q , but decreasing B_q . And by considering the excess pore pressure ratio, $\Delta u_2/\sigma'_{v0}$, the modified charts are able to separate the trends again (Schneider et al. 2008).

Interpretation in Sands (Free Drainage Sediments)

For the silica sands, the major correlations are for the relative density D_r and the state parameter ψ (Been and Jefferies 1985). The correlation of relative density may be correlated with normalized cone resistance $Q = q_{net}/p'_0$ where p'_0 is the in situ mean effective stress as:

$$D_r = \frac{1}{C_2} \lambda \ln \left(\frac{1}{C_0} \frac{q_c}{p'_0} \left(\frac{p'_0}{p_a} \right)^{1-C_1} \right) \quad (1)$$

where p_a is atmospheric pressure, and the various parameters are usually taken as, $C_0 = 25$, $C_1 = 0.46$, $C_2 = 2.96$.

Note for the more compressible (still free draining) sediments, such as carbonate sands, much lower normalized cone resistances are given unless cemented.

The cone penetration tests are increasingly preferred as a design method for the driven piles in sand by the American Petroleum Institute (API) and International Standards Organisation (ISO). And cone penetration test is also regarded as a model test for the shallow foundation design (Randolph et al. 2004).

Interpretation in Clays

Undrained Cone Penetration

The cone penetration in fine-grained sediments can be regarded as undrained behaviours, and the interpretation of cone test mainly focused on assessing the shear strength profile by cone factor N_{kt} ,

$$s_u = \frac{q_t - \sigma_{v0}}{N_{kt}} = \frac{q_{net}}{N_{kt}} \quad (2)$$

As the cone factors range widely (Lunne et al. 1997) depending on the soil characteristics, industry practice recommends calibrating the cone

resistance at each site against laboratory strength measurements.

For the ease of use, theoretical solutions relate N_{kt} with the rigidity index $I_r = G/s_u$, a stress anisotropy factor $\Delta = (\sigma'_{v0} - \sigma'_{h0})/2s_u$, and friction ratio on the cone face $\alpha_c = \tau_f/s_u$:

$$N_{kt} \approx C_1 + C_2 \ln(I_r) - C_3 \Delta + C_4 \alpha_c \quad (3)$$

And the values for the various constants are given by Teh and Houlsby (1991).

Undrained T-Bar/Ball Penetration

Strictly, the correction to the net resistance is also required for the full-flow penetrometers. However, it is often regarded sufficient to ignore such correction, due to the large projected area of the penetrometer relative to the shaft.

Similar to the cone penetration test, the shear strength profile may be estimated from a T-bar or ball penetration by means of a penetrometer factor $N_{k,Tbar}$ or $N_{k,ball}$. The factor $N_{k,Tbar}$ is originally recommended taken as 10.5, based on a plasticity solution for a cylinder moving laterally through the soil (Randolph and Houlsby 1984; Martin and Randolph 2006). The refined analyses consider strain rate hardening, remolding, and the soil conditions of offshore sites, which leads to minor adjusted recommended values. Theoretical resistance factors for the ball penetrometers are around 20–25% greater than that for the cylindrical T-bar (Randolph et al. 2000), but the field and laboratory data indicates much closer resistances (Chung and Randolph 2004).

Cyclic penetration and extraction tests allow assessment of the remolded shear strength, and this often requires a load cell zero correction (Randolph et al. 2007).

Vane Shear Test

Interpretation of the vane shear test follows the classical relationship between torque and undrained shear strength by:

$$s_u = \frac{T}{\frac{\pi d^3}{6} (1 + 3 \frac{h}{d})} \quad (4)$$

The measured strength in a vane shear test is sensitive to the precise testing procedure,

especially the delay between insertion and testing, and the rotation rate (Chandler 1988). However, the current practice recommends no correction factors to offshore vane strength, as the different factors compensate each other to some extent (Quirós and Young 1988).

Pore Pressure Measurement and Consolidation Characteristics

Cone Test

The filter positioned at the shoulder of the cone provides the pore pressure measurement during a cone test. The measured dissipation response may be fit to the theoretical solutions (e.g., Teh and Houlsby 1991), as to derive a consolidation coefficient. It is argued that the soil close to the cone may undergo consolidating, but the major part of the surrounding soil undergo swelling, and therefore the derived coefficient reflects both the horizontal flow paths and the swelling nature of the strain path (Levadoux and Baligh 1986). The dissipation tests in clay may require 1–3 h for the T_{50} to be reached, and this is very expensive in the offshore condition. In siltier sediments, the test requires much shorter duration. The derived consolidation coefficient also helps to determine whether the cone penetration itself is an undrained process and the cone penetration resistance is affected by the partial consolidation (Randolph and Hope 2004).

T-Bar and Ball Penetrometers

The pore pressure sensors may also be built into both T-bar and ball penetrometers (Kelleher and Randolph 2005; Low et al. 2007). And the piezoball penetrometer is a more efficient device compared to the piezocone penetrometer to detect the consolidation coefficient in the field condition, if the same diameter is adopted for both penetrometers.

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Offshore Mooring System

- ▶ [Mooring Anchor](#)
- ▶ [Mooring Connector](#)
- ▶ [Single-Point Mooring](#)
- ▶ [Spread Mooring System](#)

Offshore Pile Driving

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Synonyms

[API \(American Petroleum Institute\)](#); [CPT \(Cone Penetration Test\)](#); [IFR \(Incremental Filling](#)

[Ratio\)](#); [LDPPs \(long large diameter pipe piles\)](#); [PLR \(The plug length ratio\)](#); [SPI \(The soil plugging index\)](#); [SRD \(Soil resistance during the driving\)](#)

Definition

Pile driving is the process of installing a pile into the seabed in order to support or anchor super structures.

Introductions

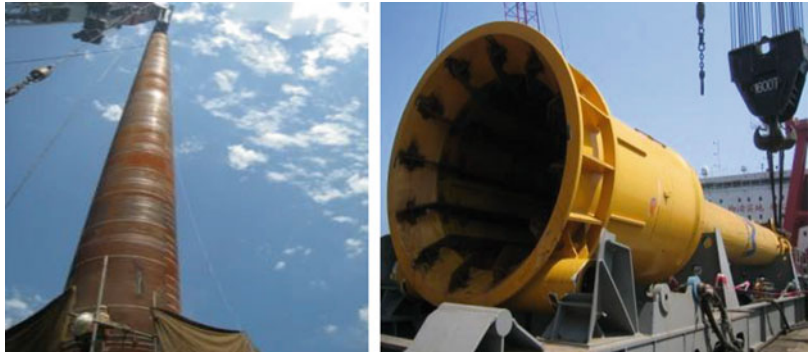
Pile driving system includes the hydraulic hammer hanging on a crane hook and the pile (shown in Fig. 1). Open-ended piles are widely used in many engineering applications, especially for offshore and near-shore engineering, such as jacket platforms, mooring systems, offshore wind turbines (Fig. 2). The advantages of onshore applications of piles are well recognized, in particular because of the short installation time and low installation cost.

The driveability of the open-ended piles is influenced significantly by the soil resistance during the driving (SRD), which is generally attributed to the outer skin resistance, inner skin resistance, and bearing resistance. The soil plug may influence the SRD. During the driving, the penetration velocity will also influence the SRD. There are also many issues during the pile installation, including premature pile refusal and pile running during the driving process (Yan et al. 2012; Guo and Shao-Lin 2011; Randolph 2003a; Dover and Davidson 2007). The work is to summarize the study outcomes on pile driving and calculation method, including pile driving formula, the penetration resistance, and the pile running and premature refusal analysis.

Pile Driving Formula

The vibratory driving technique such as jacking and impact driving is widely used in offshore and near-shore engineering. Pile driving formula is

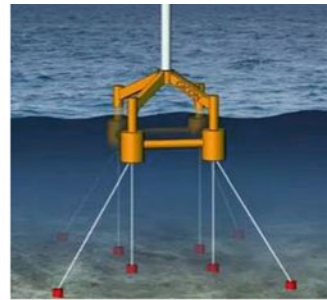
Offshore Pile Driving,
Fig. 1 Pipe pile and
hammer



(a) Jacket platform



(b) offshore wind turbines



(c) mooring piles

Offshore Pile Driving, Fig. 2 Piles used in Jacket platform, offshore wind turbines, and mooring systems. (a) Jacket platform. (b) offshore wind turbines. (c) mooring piles

used for designing the driving and determining the pile capacity. Pile driving formulas can be written mathematically as (Olson and Flaate 1967):

$$e_h W_H h = Q_u (s + s_c) \quad (1)$$

where W_H is the hammer (ram) weight; h is the hammer drop height; e_h is the hammer efficiency; Q_u is the ultimate pile capacity; s is the observed pile set; and s_c is an empirical constant expressing the aforementioned energy losses and the energy stored temporarily inside the pile due to elastic compression during the blow process.

There are a large variety of pile driving formulas available in the literature; they are summarized in Table 1.

Hiley Formula (1925): Q_u = the ultimate load, W_H = Weight of Hammer, W_P = weight of pile, C_1 = temporary compression in packing helmet, C_2 = temporary compression of pile, C_3 = temporary compression of ground at toe (often

referred as quake), s = pile penetration for last blow, also called “set,” e = coefficient of restitution.

The pile driving formulas summarized above usually used in specific deposit that verified through the experiment.

Penetration Resistance During Pile Driving

Soil Plug During Pile Driving

The contribution of the plug capacity to the SRD is very significant if the pile is plugged during driving, either partially or fully. When the pile is plugged during driving, the end bearing is not limited to the frictional resistance developed by the soil plug for SRD calculations (Randolph et al. 1991). To study the effect of plugging on the SRD, it is necessary to measure the movement of the soil plug. Therefore, it is important to quantify the

Offshore Pile Driving, Table 1 Summary of pile driving formulas

Formula	Equations	Notes
Danish Formula(Olson and Flaate 1967)	$Q_u = \frac{e_h E_h}{s + \sqrt{(e_h E_h L_{2AE})}}$	e_h = hammer efficiency E_h = the rated hammer energy E = modulus of elasticity of pile materials L = length of piles A = area of pile cross section
Hiley Formula(Hiley 1925)	$Q_u = \frac{\eta e_h E_h}{s + \frac{C}{2}} + W_H + P$	$C = C_1 + C_2 + C_3$ $\eta = \frac{(W_H + e^2 W_P)}{(W_H + W_P)}$
Janbu Formula (Janbu 1953)	$Q_u = \frac{e_h E_h}{K_u s}$	$K_u = C_d \left[1 + \left(1 + \frac{\lambda_e}{C_d} \right)^{0.5} \right]$ $C_d = 0.75 + 0.15 \frac{W_P}{W_H}$ $\lambda_e = \frac{e_h E_h L}{AE s^2}$
Dutch Formula (Whitaker 1976)	$Q_u = \frac{e_h E_h}{s + a W_P / W_H}$	$a = s$ for drop hammers and 0.1 for steam hammers
Gates Formula (Gates 1957)	$Q_u = a \sqrt{(e_h E_h)} (b - \log s)$	$e_h = 0.75$ for drop hammer s = rated hammer energy $a = 104.5, b = 2.4$
Modified ENR (ENR 1965)	$Q_u = \left(\frac{1.25 e_h E_h}{s + C} \right) \left(\frac{W_H + n^2 W_P}{W_H + W_P} \right)$	$C = 0.0025 \text{ m}$ $n = 0.5$ for steel-on-steel anvil on steel or concrete piles
PCUBC formula (Hiley 1925)	$Q_u = \left(\frac{e_h E_h W_H + K W_P}{s + \frac{Q_H L}{AE}} \right)$	$K = 0.25$ for steel piles and 0.1 for all other piles

degree of plugging which can be done by measuring the Incremental Filling Ratio (IFR) during driving. The degree of plugging can be quantified by IFR defined as,

$$IFR = \frac{\Delta L_s}{\Delta D} \times 100\% \quad (2)$$

where ΔL_s is the unit increment of the soil plug length corresponding to a unit increment of pile penetration depth (ΔD). The fully plugged and fully coring modes are then represented as IFR = 0 and 100%, respectively. A value of IFR between 0% and 100% means that the pile is partially plugged. (Paik and Salgado 2003a) IFR is linked to the hammer impact characteristics such as input energy and pile diameter. (Brucy et al. 1991).

The degree of soil plugging can be quantified using the plug length ratio (PLR) and IFR. The PLR is defined as:

$$PLR = \frac{L_s}{H} \quad (3)$$

where L_s = the soil plug length (m), H = penetration depth of pile (m).

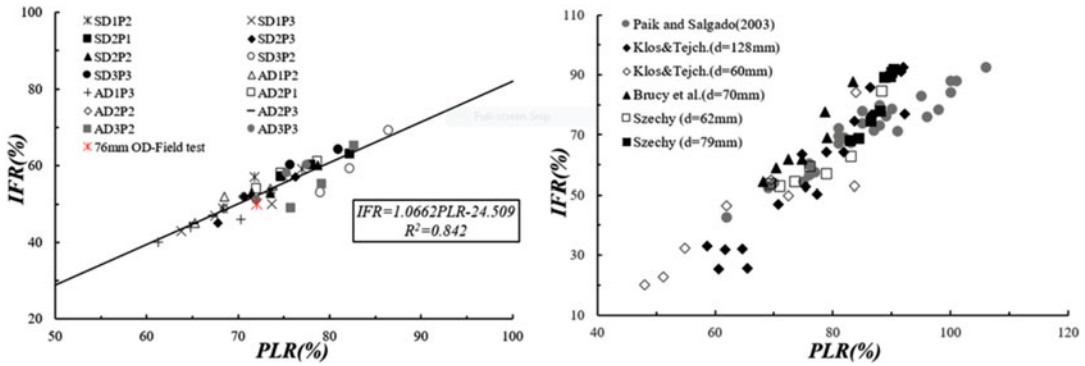
The relation between IFR and PLR can be defined as $IFR(\%) = a \times PLR + b$ (Jeong et al. 2015). Many researchers studied the relationship between IFR and PLR (Jeong et al. 2015; Murthya et al. 2018; Paik and Salgado 2003b; Xie et al. 2005; Fattah et al. 2016), a and b are also established though model test (Szechy 1959; Klos and Tejchman 1977), shown in Fig. 3.

Penetration Resistance Calculation

The penetration of an open-ended pile consists of three components during driving, including tip resistance, outer skin resistance, and plug resistance (Kumara et al. 2016) which is can be defined as:

$$Q_u = Q_{an} + Q_{out} + Q_{plug} \quad (4)$$

$$Q_{plug} = \min(Q_{in}, Q_b) \quad (5)$$



Offshore Pile Driving, Fig. 3 Relationship between IFR and PLR

where Q_u is ultimate bearing capacity, Q_{an} is annulus resistance, Q_{out} is outer frictional resistance, Q_{plug} is plug resistance, Q_{in} is inner frictional resistance, and Q_b is base resistance.

In the calculation of ultimate bearing capacity, the inner friction resistance is more complex than the end resistance and lateral friction resistance due to the internal earth plug.

According to Paikowsky (1990) arching occurs within a soil plug under static loading, which leads to high internal friction.

The soil plugging index (SPI) is defined as:

$$SPI = \frac{L_{is}}{L_s} \times 100 \quad (6)$$

where L_s is the soil plug length and L_{is} is the length of the developing inner skin friction. According to Sangseom Jeong et al. (2015) the soil plugging index (SPI) based on the combined test results is expressed as $SPI = -0.03D_i + 43.2$, where D_i is the inner pile diameter (mm).

The proposed SPI is a function of the inner pile diameter. The length of mobilizing inner skin friction is obtained as follows:

$$L_{is} = \frac{L_s \times SPI}{100} \quad (7)$$

The actually mobilizing inner skin friction can be obtained by using the obtained L_{is} .

$$Q_{si} = f_{si} \times A_{si} = f_{si} \times \pi D_i L_{is} \quad (8)$$

where Q_{si} is the actually mobilizing inner skin friction, f_{si} is the unit inner skin friction, A_{si} is the area of the mobilizing inner skin friction, and D_i is the inner pile diameter (Jeong et al. 2015).

In the practice, the open-ended pile will have a completely occlusion effect and partially occlusion effect, and the bearing capacity formula of the completely plugged pile can be calculated as (Yan et al. 2011):

$$Q_u = \eta q_u A_0$$

$$\eta = \begin{cases} 0.490.65 & \text{Clay} \\ 0.49 & \text{Silt} \\ 0.49 & \text{Sand} \end{cases} \quad (9)$$

$$q_u = 1.3cN_c + \gamma_1 DN_q + 0.6\gamma RN_r$$

$$\gamma_1 = \gamma + 2 \frac{f_s + n\tau}{(n^2 - 1)R}$$

where η is Dynamic effect coefficients, R is the inner radius of the pile, and γ_1 is the equivalent unit weight of soil above the pile tip. γ is the unit weight of the soil above the pile tip, f_s is the ultimate raft friction, and τ is the shear strength of the soil above the pile tip.

By the survey data of CPT method, the calculation method of the bearing capacity of pipe piles with complete and incompletely occlusion was summarized (Lehane et al. 2005) and the calculation of pile tip resistance in the case of complete occlusion is as follows:

$$\begin{aligned} Q_b &= q_{b0.1} \frac{\pi}{4} D^2 \\ \frac{q_{b0.1}}{q_c} &= 0.6 \end{aligned} \quad (10)$$

where q_c is average CPT end resistance, $q_{b0.1}$ is the pile end bearing resistance at a pile base movement of 10% of the pile diameter, and D is the pile diameters.

Under the condition of partially occlusion, the pile end bearing is calculated as follows:

$$\begin{aligned} Q_b &= q_{b0.1} \frac{\pi}{4} D^2 \\ \frac{q_{b0.1}}{q_c} &= 0.15 + 0.45 \times A_{rb}^* \\ A_{rb}^* &= 1 - FFR \left(\frac{D_i^2}{D^2} \right) \\ FFR &= \min \left\{ 1, \left[\frac{D_i(m)}{1.5m} \right]^{0.2} \right\} \end{aligned} \quad (11)$$

where D_i is Inner Diameter, A_{rb}^* is the effective area ratio, the other parameters as same as 10 equation. And the skin resistance is calculated as follows:

$$\begin{aligned} Q_s &= \pi D \int \tau_f dz \\ \tau_f &= \sigma'_{rf} \tan \delta_{cv} = \frac{f}{f_c} (\sigma'_{rc} + \Delta \sigma'_{rd}) \tan \delta_{cv} \\ \sigma'_{rc} &= 0.03 \cdot q_c (A_{rs}^*)^{0.3} \left[\max \left(\frac{h}{D}, 2 \right) \right]^{-0.5} \\ A_{rs}^* &= 1 - IFR_{\text{mean}} \left(\frac{D_i^2}{D^2} \right) \\ IFR_{\text{mean}} &= \max \left[1, \left(\frac{D_i(m)}{1.5m} \right)^{0.2} \right] \\ \Delta \sigma'_{rd} &= 4G \frac{\Delta r}{D} \text{ with } G = q_c \cdot 185 \cdot q_{c1N}^{-0.7} \text{ and} \\ q_{c1N} &= (q_c / p_{atm}) / (\sigma'_{vo} / p_{atm})^{0.5} \end{aligned} \quad (12)$$

where δ_{cv} is constant volume interface friction angle, σ'_{rf} is radial effective stress at failure, σ'_{rc} is radial effective stress after installation and equalization, $\Delta \sigma'_{rd}$ is increase in radial stress due to loading stress path (dilation), f/f_c is 1 for compression loading and 0.75 in tension, IFR_{mean} is

the average IFR measured over the final 20 diameters of penetrations, G is operational shear modulus (assumed equal to the in-situ very small strain value), Δr is interface dilation (assumed = 0.02 mm for the database), and p_{atm} is a reference stress = 100 kPa.

Velocity-Dependent Penetration Resistance

The strength behavior of soils can be significantly affected by the applied rate of loading, which is very important for the design of pile bearing capacity. Some new methods such as Cone Penetration Test (CPT) have been in popular use considering the speed of pile penetration into the ground and the relative velocity of the pile with respect to the surrounding soil.

A parameter R_f was adopted to describe the effect of velocity on soil strength (Chow et al. 2015), which can be expressed as:

$$R_f = \left[n \frac{v/d}{(v/d)_{ref}} \right]^\beta \quad (13)$$

where v_{ref} is a reference velocity equal to the penetration rate adopted in the measurement of $s_{u(CPT)}$, d is the FFP diameter, n is introduced to account for the greater rate effects reported for shaft resistance compared to tip resistance, and β is a strain rate parameter.

The bearing and frictional resistances were scaled by a rate function R_f and assumed that at any given location the operational strain rate may be approximated by the normalized velocity (O'Loughlin et al. 2013). R_f can be expressed as:

$$\begin{aligned} R_f &= 1 + \lambda \log \left[\frac{v/d}{(v/d)_{ref}} \right] \quad \text{or} \quad R_f \\ &= \left[\frac{v/d}{(v/d)_{ref}} \right]^\beta \end{aligned} \quad (14)$$

where λ and β are strain-rate parameters and v is velocity of penetration in static penetration tests.

The effect of shear rate on measured vane shear strength was studied (Sharifounnasab and Ullrich 1985) and the relevant formula is:

$$\frac{s_v}{s_{v0}} = \alpha \left(\frac{\omega}{\omega_0} \right)^\beta \quad (15)$$

in which s_v is vertical shear strength; s_{v0} is vertical shear strength, measured at the standard shear rate; ω is shear rate; ω_0 is standard shear rate; and α and β are coefficients which depend on the properties of the soil.

The majority of dynamic pile tests are analyzed using pile–soil interaction models based on Smith (1960), (Randolph 2003b) with the soil resistance expressed as:

$$\tau_d = \text{Min} \left(1, \frac{w_p}{Q} \right)^\beta (1 + Jv) \tau_s \quad (16)$$

where w_p is the local pile displacement, τ_d and τ_s are the dynamic and static shaft friction values, Q is the displacement, or “quake,” to mobilize the limiting static resistance, and J is a damping parameter multiplying the local pile velocity, v .

American Petroleum Institute (API) does not provide guidelines for the effects of cyclic loading and rate of loading on the axial capacity of piles. (Iskander 2010) There is evidence that cyclic tensile loading, in particular, may significantly reduce pile capacity (Kraftjr 1990).

Thus, it can be seen the velocity do have a significant effect on the properties of soil which cannot be ignored and more research needs to be further developed.

Refusal and Pile Running During the Driving

Pile is an important part of offshore platform structure. The materials and operations of construction have a significant impact on the cost of offshore projects. It has proved that there are certain risks in the construction of pile and may cause some serious problems during the operation, such as pile driving refusal and pile running. With the development of offshore platforms into deep water and large scale, the impact of such kind of risks is even more prominent. Therefore, it is necessary to have a good understanding of the

causes of these risks from the technical perspective and present the preventive methods and measures in the design and construction.

Pile Driving Refusal

Super long piles with large diameters usually have to be manufactured by segments before installation because of transportation and hoisting difficulties. The segments of the pile will be assembled by welding to the formerly penetrated one. The assembling work often takes 1 day or even a longer time. In additional, the construction process may pause due to other various problems which will take a long time to restart the driving (Yan et al. 2012).

During this period of time, the excess pore pressure in the soil below and around the pile, which is built up during pile driving, will dissipate to some extent, increase the strength of the soil, and make restarting the successive penetration very difficult, sometimes even refusal may take place. This matter has been encountered quite often in many engineering cases and puzzled the designers for years. As for the pile driving refusal, many experts have conducted extensive research and made a series of achievements.

Based on the platform WHPE in Bohai Gulf, Yan et al. (2012) proposed several measures to reduce the refusal risk for restarting a segmented pile:

1. Cutting down the pile extension time is the most effective measure to minimize the refusal risk.
2. To keep the pile tip in a clay layer instead of in the sand layer when extending the pile segment.
3. When the stick-up stability was satisfied, increasing the length of the last segment of the pile and assembling the pile segments when the pile tip penetrates into a clay layer could greatly ease the difficulty of driving the pile.

The reasons of piling refusal in the restarting process were analyzed and presented a numerical method for judging the risk of refusal by estimating the blow counts after pile extension, in which

the regain of soil strength is considered (Yan et al. 2015a). A fatigue factor β , which can be expressed as the following formula, is consistently introduced to cover all reasons and associated with the degree of consolidation.

$$\beta = \begin{cases} \left(\frac{X}{Y-5}\right)^2, & X \leq Y - 5 \\ 1, & Y - 5 < X \leq Y \end{cases} \quad (17)$$

where Y is the designed penetration and X is the specified pile tip penetration. In order to predict if refusal may occur at any restart penetration after pile extension, blow counts needed to penetrate the pile for further 30 cm are estimated. If the estimated blow counts less than an acceptable value, refusal is considered not to take place at this penetration.

A case of a practical offshore pile installation which experienced refusal is analyzed (Sa et al. 2013). Empirical equations are provided to predict soil resistance during driving and after setup. According to the research results of Skov and Denver (1988), the empirical relationship to estimate the time effect on the resistance which helps to analyze bearing capacity can be expressed as:

$$\frac{Q_t}{Q_0} = 1 + A \left[\log \left(\frac{t}{t_0} \right) \right] \quad (18)$$

where the soil resistance corresponding to finishing one pile section is denoted as Q_0 while Q_t represents the soil resistance after a suspension of time t . A and t_0 are two empirical parameters depending on the soil properties.

Characteristics of local soil around the pile are summarized though comparing the resistance changes between local geologic data and pile driving record data (Pu 2012). Result shows that the soil resistance within a period of time after driving is much higher than that in the process of pile driving. Based on the driving records, the back-figured method is applied to estimate the pile bearing capacity after refusal taking place. The required equipment, key technology, and main technological processes in the installation process of the shallow water jacket after pile refusing were described (Hui and Wang 2014).

The researches relevant to the driving refusal mainly focused on the mechanism and some measures based on the practical engineering or simulation software, which can provide the references and analytical analysis to the design and construction of practical project. Some more flexible and economic measures need further study.

Pile Running

Offshore platforms in deep seas require extremely long large diameter pipe piles (LDPPs) to support the loads generated from the structure itself, wind and waves. If the LDPPs are in two or more segments, in situ pile splicing is required, which requires additional construction time and leads to premature refusal. One solution to this problem is to use a single segment of LDPP and drive it continuously into the seabed. As the LDPP could be very heavy (700 t or more), pile running may occur (Randolph 2003a; Dover and Davidson 2007; Olson and Flaate 1967). The unexpected pile running may break the steel wire or even cause the hammer to be lost in the sea.

The researches relevant to the pile running are mainly engaged in the predicting procedure based on the energy conservation method or critical equilibrium conditions.

Yan et al. (2015b) proposed a procedure for predicting the pile-running zones based on the limit equilibrium of pile weight and soil resistance. The total soil resistance can be expressed as:

$$Q_d = Q_f + Q_p = \bar{f} A_s q_u A \quad (19)$$

where Q_d is the total soil resistance, Q_f is shaft resistance, Q_p is the bearing capacity of the pile sand, $\bar{f}$ is the average frictional force per unit area, q_u is the bearing capacity of the pile end, and A is the area of the annular pile end. Pile-running may occur if the following conditions are satisfied:

$$Q_d \leq W_P \quad \text{or} \quad Q_d \leq W_P + W_H \quad (20)$$

In which W_H is the hammer weight and W_P is the weight of the pile. The equilibrium conditions above belong to different states. Yan et al. (2016) also established a predicting procedure based on

the energy conservation method of the running pile. The key theoretical formula can be expressed as:

$$E_G = W_r + E_s \quad (21)$$

where E_G is the potential energy of pile hammer, W_r is the work done by lateral friction resistance and the tip resistance, and E_s is the consume energy.

Sun et al. (2016) proposed an analytical method to predict the occurrence of pile running and established the framework for determining the calculation parameters from a laboratory or field test. The process can be separated into two phases: (1) the energy transfers from the hammer to the pilehammer system and (2) the pile and hammer sink together into the soil embedment, which can be, respectively, expressed as the following formulas:

$$\eta_b m_h g h_p = \frac{1}{2} (m_h + m_p) v_0^2 \quad (22)$$

$$\begin{aligned} \frac{1}{2} (m_h + m_p) (v_j^2 - v_{j-1}^2) \\ = (W - F_s - F_t - F_b) \Delta S \end{aligned} \quad (23)$$

where m_h is the mass of the ram (unit in kg); g is the gravitational constant; h_p is the stroke of the hammer; m_p is the total mass of the LDPP, anvil, helmet, and follower with units in kg; v_0 is the instantaneous velocity of the hammer and pile at the time immediately after the impact; and η_b is the efficiency of the blow. Sun et al. (2017) also proposed a simplified theoretical method to explain the mechanisms of the pile running and proposed a factor of friction degradation to calculate the dynamic skin friction from the static ultimate skin friction of surrounding soil to predict the pile running condition. Based on the force equilibrium on the vertical direction, the resultant force F could be derived as:

$$F = W - Q_f - Q_p - F_b \quad (24)$$

where W is the effective self-weight of the LDPP and hydraulic hammer, Q_f is the side skin friction,

Q_p is the end bearing resistance, and the F_b is Buoyant weight of the displaced soil. The LDPP driven can be analyzed by this formula.

Conclusions

The driveability of the open-ended piles is influenced significantly by the soil resistance during the driving (SRD); SRD consists of the outer skin resistance, inner skin resistance, and bearing resistance. The soil plug has a great influence to the SRD. The soil strength is dependent on the penetration velocity; during the driving, the penetration velocity will also influence the SRD. Based on the influence factor, the calculation method of the penetration resistant is summarized, the judgment approach of the pile running and premature pile refusal are also summarized.

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Offshore Project Development

► Field Development

Offshore Ship

► Offshore Vessel

Offshore Structure

► Offshore Structure Design Under Ice Loads

Offshore Structure Design Under Ice Loads

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Synonyms

Accidental loads; Ice loads; Offshore structure;
Plastic design; Ultimate strength

Definition

Offshore structure is an installation of structures in a marine environment with relative far distance to nearest shore, usually for the production and transmission of electricity, oil, gas, and other resources. Typical offshore structures are subsea oil and gas developments, offshore platforms (fixed platforms, semi-submersibles, spars, tension leg platforms, floating production storage, and offloading), offshore wind turbines, and subsea pipelines. Depending on the loading conditions, offshore structures can be made of steel, concrete, or composite materials. **Ice loads** are referring to the loads exerted to *offshore structures* from various ice types, such as first-year level ice, multi-year level ice, ice ridge, brash ice, and iceberg. The main focus of this section is the design methodology used for the design of *offshore structure under ice loads*.

Scientific Fundamentals

Limit State Design Principles

The offshore structures shall be designed in a way that it can sustain all possible loads during their lifetime without causing human loss and significant economic or environmental pollutions. The designers are encouraged to check the conditions that may lead to structure failure, which are referred as limit states. Following limit states are

defined for offshore structure design under ice actions (ISO19906 2010):

- Ultimate limit state (ULS) that generally corresponds to resistance to extreme-level ice event (ELIE), both local and global actions shall be considered.
- Serviceability limit state (SLS) that corresponds to criteria governing normal functional use under serviceability-level ice event (SLIE).
- Fatigue limit state (FLS) that corresponds to the accumulated effect of repetitive actions.
- Accidental limit state (ALS) that corresponds to accidental events and abnormal-level ice event (ALIE), both local and global actions shall be considered.

Generally, the offshore structure shall be designed to comply the criteria corresponding to all of the above defined limit states. However, it is challenging to predict the ice loads with high precision. The stochastic characteristics of ice loads increase the complexity of structure design. Permanent plate deflections due to compressive ice loads may happen due to large exposed ice loads (Hänninen 2010). An extreme case is the famous Titanic accident in maritime history back to 1912. Obviously, the cruise ship is not considered to take the iceberg impact loads at that time.

Limiting Mechanisms for Ice Loads

In order to identify the actual ice loads, three limiting mechanisms are defined. They are namely termed as limit stress, limit force, and limit momentum.

(a) Limit stress

The ice fails against the structure and the strength of ice determines the ice loads level. This normally gives highest ice loads. The unconfined strength of ice is commonly used to represent the ice strength.

(b) Limit force

The driving force, such as winds, currents, and the surrounding pack ice, are pushing the ice features against wide structures with thinner pack ice failing behind. The load on the structure is given by the wind and current drag on the ice feature leaning against the structure

plus the pack ice driving pushing on the back of the ice feature.

(c) Limit momentum

This mechanism corresponds to the ice mass with certain velocity collides with a structure and the momentum of ice mass is insufficient to envelop the whole structure. The contact force builds up as the initial momentum changes.

Types of Offshore Structure

The design of offshore structure under ice loads largely depends on the loads level of ice events. Meanwhile, the structure types also have significant impacts to the ice loads. The interaction between structure and ice is a complicated process. The physical understandings are still limited. The characteristics of structural geometry shape shall be considered when evaluating the ice loads. Typical offshore structure types are as follows:

- Floating production, storage, and offloading unit (FPSO)
- Semisubmersible floater
- Gravity-based structure (GBS)
- Jackup rig
- Icebreaker
- Offshore supply vessel (OSV)
- Offshore oil tanker
- Liquefied natural gas carrier (LNG)
- Offshore wind turbine

Based on the main dimensions of the offshore structures, they can be categorized as shown in

Fig. 1. Figure 2 shows some of the main concepts used for offshore structures (not necessarily for ice loads).

Types of Interaction Scenario

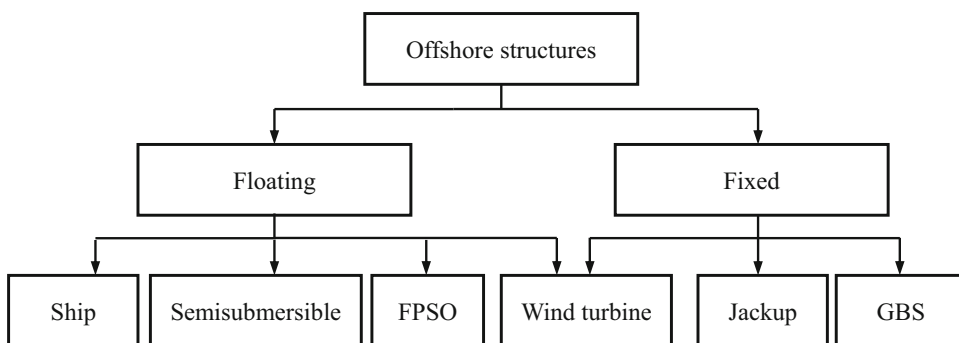
The interaction scenarios play a crucial role on deriving ice-related loads. For an optimized structure design, the identification of the interaction scenario is of great importance. The typical interaction scenarios are summarized as follows:

(a) Vertical-sided structure

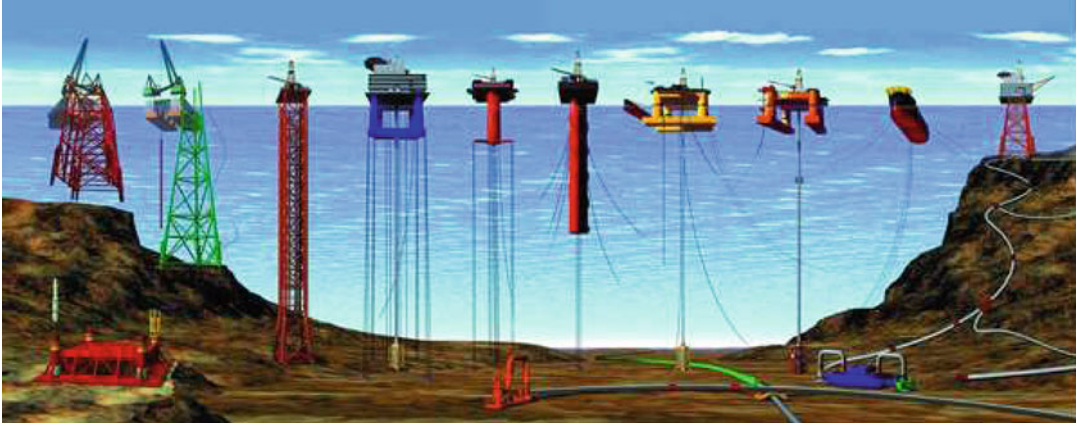
The ratio between the width of structure and the ice thickness determines the likelihood of possible ice failure modes. The crushing of ice against the vertical wall may cause significant local high pressure, as explained by Jordaan (2001). This type of interaction is normally for fixed structures under relative light ice conditions. Figure 3 shows an illustration of this interaction scenario.

(b) Multi-leg

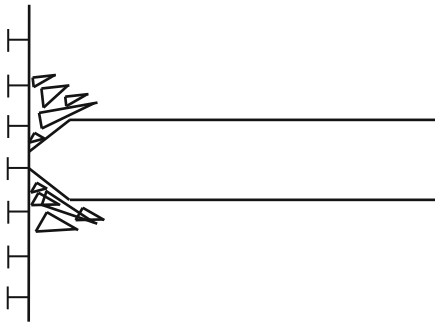
This type of interaction has decreased exposed contact area, and consequently, it may have better performance under harsh wave loads. It is normally used for environment where ice is not severe. The presence of several legs interacting with ice in close vicinity to each other has mutual influences, see the discussions by Karulin and Karlina (2014). The sheltering effect causes unsymmetrical ice loads to legs, which may increase the yawing



Offshore Structure Design Under Ice Loads, Fig. 1 Main offshore structure under ice condition



Offshore Structure Design Under Ice Loads, Fig. 2 Some typical offshore structural concepts with permission of NOAA (2010)



Offshore Structure Design Under Ice Loads, Fig. 3 An illustration of ice sheet crushing with vertical side structure

moment. Figure 4 shows scenarios for a multi-leg platform operating in ice conditions with different heading angles. The potential jamming effect of broken ice could be another hazard to the multi-leg structures.

(c) Sloping-sided structure

Sloping-sided structures break ice upwards or downwards. The corresponding ice failure mode is bending failure and the bending failure results generally lower ice loads. This type of structure has been widely applied to various offshore structures, such as jackets in Bohai sea of China (Yue and Xiangjun 2000), wind turbines in ice covered waters (Barker et al. 2005), and the confidential bridge in Canada (Aïtcin et al. 2016). The main loads are

from ice-breaking failure and ice clearance (Fig. 5).

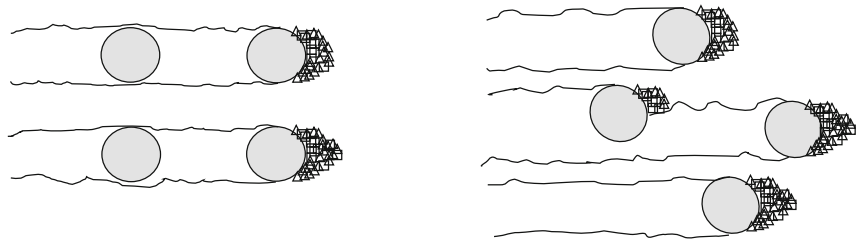
p-A Relationship for Ice Crushing

The ice-structure interaction is a complicated process. The understanding of it is still limited. For simplification, ice crushing failure pressure could be determined by the contact area (Sanderson 1988). The pressure and area relationship is widely used for designing structures under ice actions, see Fig. 6 for an example. It is defined as follows:

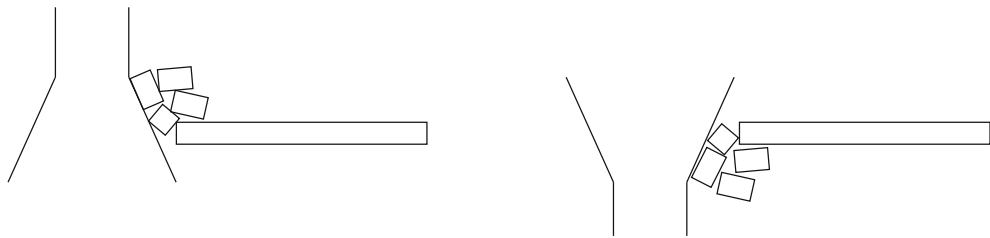
$$p = C_p \cdot A^{D_p}$$

where p is pressure, C_p and D_p are both constants. Constants C_p and D_p are depending on various factors, such as the confinement effects, indentation geometry, area of loading, aspect ratio etc., see the discussion by Johnston et al. (1998). The understandings to the background of the pressure-area relationship are crucial for selecting the right combinations.

The pressure p can be either global nominal pressure or local contact pressure, and the constants shall be selected accordingly. The improper selection of constants may cause significant discrepancies on predicting the ice loads. The recommended constants for local pressure and area relationship from ISO 19906 are listed in Table 1.

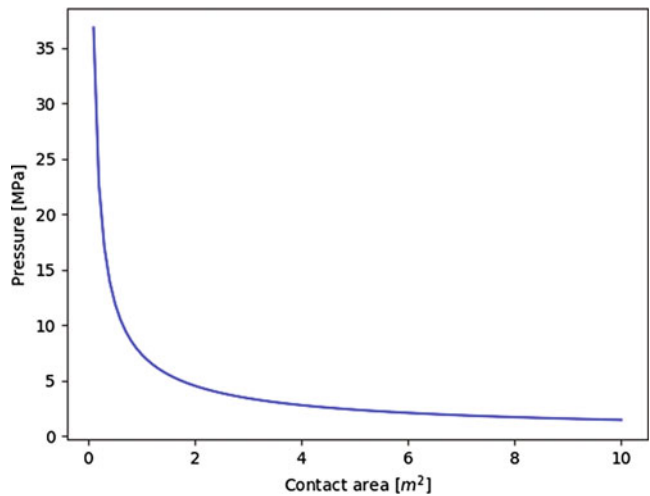


Offshore Structure Design Under Ice Loads, Fig. 4 An illustration of multi-leg interaction with ice sheet under different heading angles



Offshore Structure Design Under Ice Loads, Fig. 5 Typical ice failure on upward and downward cone

Offshore Structure Design Under Ice Loads, Fig. 6 A typical pressure area relationship plot



Local Structure Design Under Ice Crushing

In this section, the main principles of the local structure design are presented. The related regulations, the pressure area relationship and the theory behind the plate design equations are shown.

Related Regulations

The design of local structural scanting shall follow the rules that classification society has given. The main relevant ice regulations are as follows:

- The Finnish-Swedish Ice Class Rules, FSICR (Finish-Swedish 2017)

- The Russian Maritime Register of Shipping ice rules (RMRS 2005)
- The unified Polar Class (PC) rules of the International Association of Classification Societies (IACS 2016)
- The DNV rules for classification of ships, Part 5 Chapter 1 (DNV 2003)

A design patch load (uniform distributed pressure load) shall be established prior to the structure design. Figure 7 shows an example of effective contact area for structure exposed to ice sheet. The detailed procedures on how to derive the effective contact area can be found in related regulations, such as the DNV ice rule (DNV 2003).

Plastic Plate Strip Analogy

The regulations mentioned above are using either elastic or plastic analogy to derive the design

Offshore Structure Design Under Ice Loads, Table 1 Summary of constants for local pressure and area relationship, from ISO 19906 (ISO19906 2010)

Interaction scenario	Pressure type	C_p	D_p
First-year level ice	Local	2.35	-0.50
Thick, massive ice	Local	7.40	-0.70

equations, together with correction factors accounting various design scenarios.

Assuming a three-hinge mechanism for a deformed plate with unit length under uniform pressure p , the plate width is b , as shown in Fig. 8. The critical loads q could be obtained by equaling external virtual work with internal virtual work (Jørgen 2007).

External work:

$$W_e = \int_0^s p \cdot w(x) dx = pb \cdot \frac{dw}{2} \quad (1)$$

Internal virtual work:

$$W_i = M_p(d\theta + 2d\theta + d\theta) = 4M_p d\theta \quad (2)$$

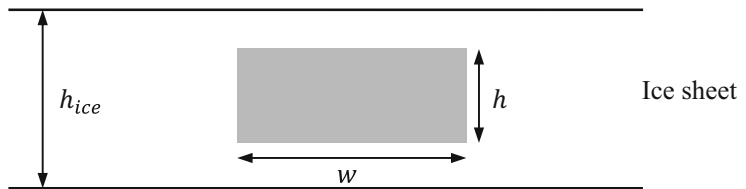
And we have:

$$dw = \frac{b}{2} d\theta \quad (3)$$

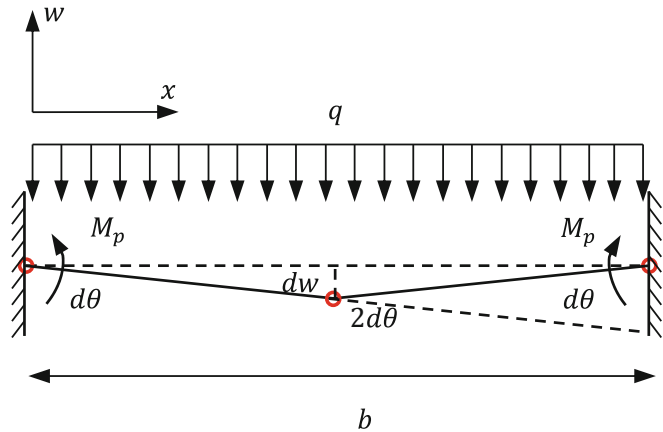
So:

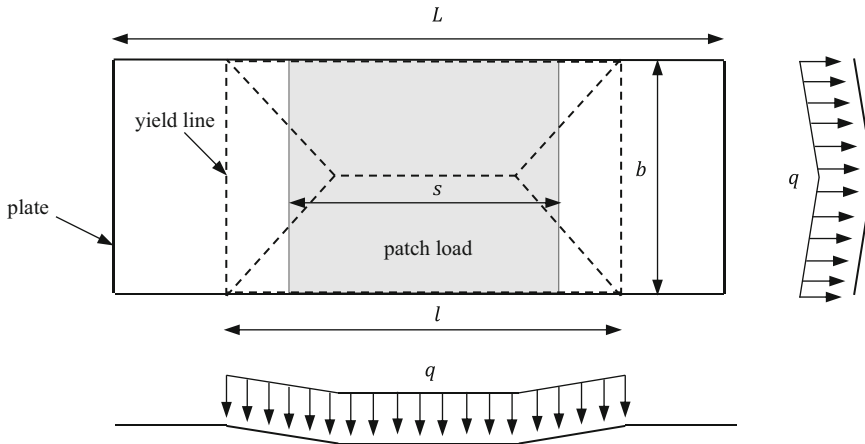
$$p = \frac{16M_p}{b^2} \quad (4)$$

Offshore Structure Design Under Ice Loads, Fig. 7 An example of effective contact area



Offshore Structure Design Under Ice Loads, Fig. 8 Plate strip analogy (plastic analysis)





Offshore Structure Design Under Ice Loads, Fig. 9 Roof-shaped yield line pattern for plate under uniform pressure load

The unit plastic bending moment is defined as follows:

$$M_p = \sigma_y W_p = \sigma_y \frac{t^2}{4} \quad (5)$$

where σ_y is the yield stress.

Combining Eqs. (1)–(5), the relationship between plate thickness and the critical external loads can be established.

$$t = \frac{b}{2} \sqrt{\frac{p}{\sigma_y}} \quad (6)$$

Equation (6) is obtained from plastic analysis for a plate with unit width under uniform distributed loads. For the plate exposed to ice load patches, a two-dimensional plastic analysis may be performed. Figure 9 shows a typical yield line pattern for a plate under uniform pressure load (patch load). The plastic analysis can be used to derive the design equations for plate thickness (Chen and Han 1988). Sobotka presents solutions for plate under loads other than uniform pressure (Sobotka 1989).

Based on the failure pattern as shown in Fig. 9, the plastic load-carrying capacity in bending for fully clamped plates is expressed by the following equation, see (Wood 1961):

$$p = \frac{12\sigma_y t^2}{l^2 \alpha^2} \quad (7)$$

where α is called the aspect ratio and defined as $\alpha = \frac{b}{l} \left(\sqrt{3 + \frac{b^2}{l^2}} - \frac{b}{l} \right)$. l is the length of load patch, and b is the patch load height.

Equation (7) can be rewritten as:

$$t = \frac{b}{2} \sqrt{\frac{p}{\sigma_y}} \left(\sqrt{1 + \frac{1}{3} \frac{b^2}{l^2}} - \frac{1}{\sqrt{3}} \frac{b}{l} \right) \quad (8)$$

In IACS (2016), the plate thickness design equation is as follows after let $l = s$ (without the safety factors).

$$t = \frac{b}{2} \sqrt{\frac{p}{\sigma_y}} \frac{1}{1 + 0.5(b/s)} \quad (9)$$

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Offshore Vessel

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Synonyms

[Offshore engineering vessel](#); [Offshore ship](#)

Definition

Offshore vessels are ships that specifically serve operational purposes such as oil exploration and construction work at the high seas.

Scientific Fundaments

Composition and Basic Type

As mentioned, above, the denotation of offshore vessels is a collective reference and as such includes a wide array of vessels employed in the high sea sector. They can be mainly classified into the following main groups:

- Oil exploration and drilling vessels
- Offshore support vessels
- Offshore production vessels
- Construction/special purpose vessels

Each of this category comprise of a variety of vessels.

Oil Exploration and Drilling Vessels

Oil exploration vessels, as the name suggests, help in exploration and drilling of oil at high seas. The main types of exploration vessels are:

Drilling Vessels Drill ships are special purpose ships which are used for drilling on the ocean beds at deep seas. Drill ships are inherently ships designed to provide optimum viability while on water, thus making it easy for the conglomerates to engage their services for better qualitative results in the overall scheme of drilling viability and functionality (Gouldin et al. 2017).

Drilling vessels can also be used as an analytical vessel to carry out sub-water researching operations in the high seas. The drilling equipment aboard these vessels can penetrate to really greater depths (anywhere over 600 m to over 3000 m) and can be relocated in the high seas as the requirement necessitates. Likewise, the drilling equipment aboard the oil drilling vessels can also be employed to shoal areas to carry out the necessarily required maritime operations.

In order to help them stabilize in the water, these vessels utilize equipment like GP systems



Offshore Vessel, Fig. 1 Drill ship. (<http://www.ussfreedom.org/home>)

and exceedingly strong support cables and ground tackles to keep them firmly positioned in the drilling area. The construction of a drill ship involves a crane-like framework to hoist and lower the required drilling apparatuses from and into the water from a hatchway specifically constructed to pass through the vessel, into the water's depths to aid the process. This feature of the offshore drilling ships makes them highly economical and conducive to the high seas drilling, in addition to the aforementioned aspects (Fig. 1).

Semisubmersible Vessels Semisubmersible vessel is majorly used in marine operations carried out in the high seas like oil drilling and production platforms for oil. In addition, semisubmersible ships are also used as heavy-duty cranes.

The semisubmersible vessel was developed because of the need for vessels that could stay afloat and carry out their required functions in the high seas amidst the constant movement of the waves.

The concept of a semisubmersible vessel emerged toward the early twenty-first century. According to many sources, Shell Company's Bruce Collipp is regarded as the pioneer and creator of these big ships. But it is also said at the same time that the idea behind the semisubmersibles was that of Edward Robert Armstrong, who utilized the

theory of landing planes in sea platforms supported by ballast tanks in a columnar form.

At first, semisubmersible rigs were designed only to be used in shallow waters. Such rigs could be used in water levels up to 30 m. But later on as the need was felt for rigs which could be operated in more depths, the invention of the marine equipment developed and widened.

The first recognized and acknowledged semisubmersible vessel was the one launched by the Blue Water Drilling Company in the year 1961. The name of the rig was the Blue Water Rig No. 1. Another semisubmersible rig was launched in the year 1963 and by 1972; the offshore marine operations had about 30 semisubmersible rigs operating (Minton 1967) (Fig. 2).

Offshore Barge A barge is a flat-bottomed boat, especially built for river and canal transport of heavy goods. Barges these days are actually used to transport low-value bulk items as the cost of hauling goods by barges are cheaper.

A typical barge measures 195 by 35 ft (59.4×10.6 m) and can carry up to 1500 t of cargo. Other than barges, the different types of shipping vessels that are used are container ships, bulk carriers, and tankers. All in all they serve the same purpose of carrying cargo, goods,



Offshore Vessel, Fig. 2 Semisubmersible vessels. (<http://www.Fallschirmjäger/wikipedia.org>)

materials, and sometimes even vehicles or people from one port to another.

A classic barge is very similar to a very large raft. In many cases, a barge is unpowered and has to be moved with the assistance of a tugboat. In this situation, the captain and first mate are aboard the tugboat, not the barge, as the tugboat is providing the power and steering for the barge.

On self-powered barges, the crew includes a captain and first mate to steer the boat and manage the crew, and a small superstructure is usually mounted on the rear of the barge (Fig. 3).

Offshore Support Vessels

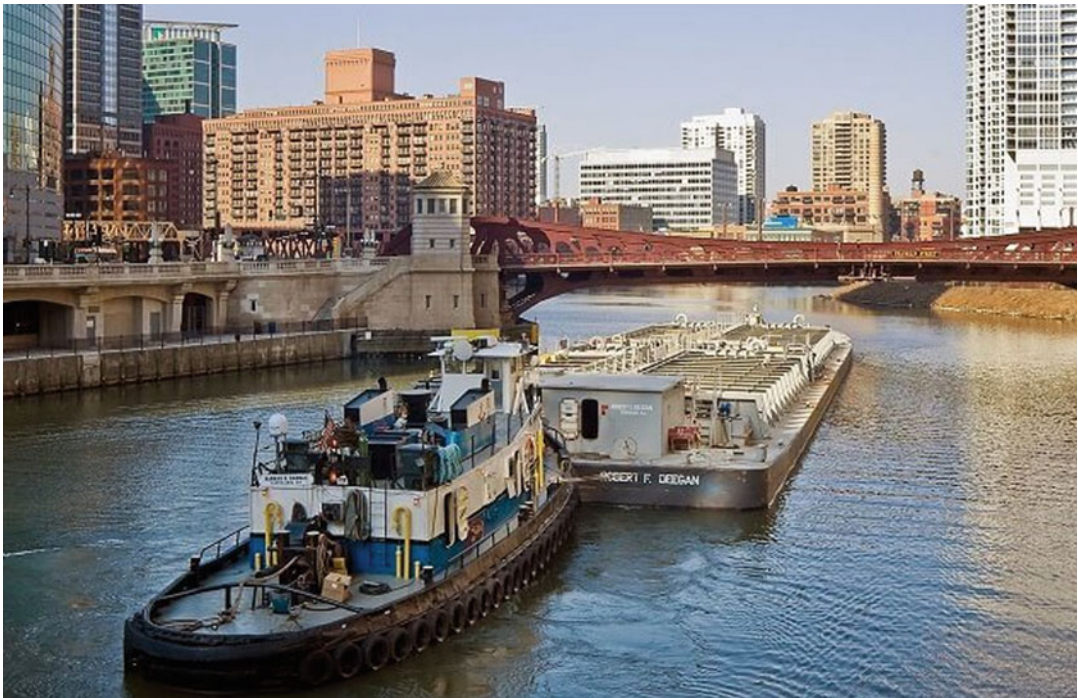
Certain offshore vessels provide the necessary manpower and technical reinforcement required so that the operational processes in the high seas continue smoothly and without any undesired interruptions. Such vessels are called as offshore support vessels.

Offshore supply vessels transport the required structural components to the designated high sea sector along with providing assistance to supply freight as well. The constructional aspect of these

vessels can be purpose-built to suit the operational demands. Some of the main types of offshore support vessels are:

Anchor Handling Tug Vessel (AHTV) AHTS vessels are a type of supply vessels that supply tugs and anchors to not just oil rigs but also to cargo-carrying barges. Technically, an AHTS is a very huge naval vessel, mainly because of the equipment that it carries – tugs and anchors along with the winches. In order to transport such a heavy bulk in a manner that they are lost while the AHTS is moving, it is but natural that the design and construction of such ships has to be accommodating to fit such equipment easily.

In addition to towing and tugging oil rigs, another major feature of such anchor handling vessels is that they also act as rescue vessels for other ships in times of some emergency. If a ship or a boat requires immediate anchor handling or towing or tugging, and if an anchor handling tug is in the oceanic vicinity, then they are a great source of help to such stranded vessels (Fig. 4).



Offshore Vessel, Fig. 3 Offshore barge. (<http://www.JeremyA/wikipedia.org>)

O



Offshore Vessel, Fig. 4 Anchor handling tug vessel (AHTV). (<http://www.BoH/wikipedia.org>)

Seismic Vessel For the purpose of seismic survey, seismic waves are the main components that are analyzed. The process involves a seismic detector that shoots such seismic waves to a selected underwater point. The time taken for the waves to refract back to their origin point determines whether that particular subsea area is feasible for the oil drilling purpose.

A survey vessel is the one that helps monitor such seismic waves. It is the primary requirement for any shipping concern engaged in or planning to engage in the process of oil and gas excavation from the oceanic reservoirs. A seismic vessel is fitted with all technological gadgets like GPS, computers, nautical charts, and any other equipment that would enhance the process of seismic survey.

Such vessels are built very carefully and only in selected locations across the world (Eisner et al. 2011). This is because the entire ship-building process for seismic vessel involves fitting all the necessary gadgets (mentioned above) without missing even a single one. Seismic vessels are more in demand in today's time considering the

amount of subsea drilling that is being carried out. They are also known as research vessels because in a completely different way, they do help research the oceans and seas (Fig. 5).

Platform Supply Vessels (PSVs) Platform supply vessels or PSVs is a type of offshore vessel which is mainly used for transiting essential equipment and additional manpower to reinforce the high seas' operations.

A platform support vessel is, at its broadest and most literal of implications, a much-needed support ship. Synonymously referred to as offshore supply vessels (OSVs), platform supply vessels help to sustain the demands of the constructional and maintenance projects, thus fulfilling a vital necessity in the nature of operations at the high seas.

The singularity of OSV ships stems from numerous factors, even aside from its unique role in the high seas constructional sector. Under a broader ambit, supply vessels help to lug not just heavy structural equipment but also smaller yet essential structural components like paving



Offshore Vessel, Fig. 5 Seismic vessel. (<http://www.SludgeG/Wikipedia>)

material (cement and concrete) and chemical compounds that help in efficient sub-water boring operations. In addition to these, food and provisions to the crew and personnel working in the high seas is also transported by way of these supply ships.

Personnel discharged from active line of operations in the high seas are also transported back to the nearest harbor facilities by way of these incoming supply vessels. Since OSV ships transit personnel, these vessels come furnished with cookhouses and other necessary facilities to facilitate an easy transiting for the personnel. In terms of their technical proportion, OSVs can measure anywhere between 65 ft and over 300 m. This aspect about platform supply vessels adds to their operational singularity.

A platform support vessel can be custom-built to suit the operational needs of its operators. Consequentially, not all PSVs are employed to transit drilling rig platforms or sub-water cables to help in the oil excavation operations (Dong et al. 2017). They are also utilized for the purposes of curbing the extent of oil spillage in the high seas

and also as handy vessels with fire-controlling instrumentation.

Presently, the need for PSVs has started to be felt more and more as compared to the past few years. This is on account of the constant rise in the number of operations in the high seas, which has in turn resulted in the advancement in the construction of the supply vessels. This demand fuelled the need for more of these vessels and coupled with the benefits of modern technologies can very well be considered as a positive thrust in a very viable and highly necessitated medium of operation (Fig. 6).

Offshore Production Vessels

Offshore production vessels refer to those vessels that help in the production processes in the drilling units in the high seas. FPSOs (floating production storage and offloading) can be enumerated as an example of these types of offshore ships. The main type of these vessels is:

Floating Production Storage and Offloading (FPSO)

The FPSO (floating production storage



Offshore Vessel, Fig. 6 Platform supply vessels (PSVs). (<https://www.marineinsight.com>)

and offloading) system is used extensively by oil companies for the purpose of storing oil from the oil rigs in the middle of the ocean and in the high seas. It is one of the best devised systems to have developed in the oil exploration industry in the marine areas.

The FPSO, as its name suggests, is a floating contraption that allows oil rigs the freedom not just to store oil but also to produce or refine it before finally offloading it to the desired industrial sectors, either by way of cargo containers or with the help of pipelines built underwater.

The use of this system ensures that shipping companies do not have to invest even more money by ferrying the raw and crude oil to an onshore refinery before transferring it to the required industrial areas. In simple terms, the FPSO saves time and money effectively (Huser and Rasmussen 2001) (Fig. 7).

Offshore Construction Vessel

Ships that primarily aid in the construction of various high seas structures are known as offshore construction vessels.

Other offshore vessels' of these type also include those that provide anchorage and tugging assistance and those kinds of ships that help in the positioning of deep sub-water cable and piping lines. Main types are:

Diving Support Vessel A diving support vessel, as the name suggests, is a vessel that is used for the objective of diving into oceans. Divers, who dive into the middle of the seas as a part of professional diving process, need proper diving support. This necessary support is provided by such a dive support vessel.

The concept of a diving support vessel came into existence four to five decades ago. From that



Offshore Vessel, Fig. 7 Floating production storage and offloading (FPSO). (<https://www.maersk.com>)

time onward, these ships have been extremely important to the field of commercial diving which forms a vital part of professional diving.

It has to be noted that professional diving means diving for the prospect of construction, repairing, and maintenance of oil rigs and other important offshore naval constructions. Such dive support ships are mainly used in the North Sea and the Gulf of Mexico since these are the areas from where crude oil is majorly excavated from subsea sources.

Such support vessels are flat-based or flat-bottomed because it makes the diving part easy for the divers. Additionally it has to be noted that such ships are equipped with the dynamic positioning system in order to help the vessel used for diving support stay steady on the water. In the absence of the dynamic positioning system, what could happen is that the ship could move away from the intended diving spot which would cause complications to the diver.

Another important feature in vessels that enable diving support is the saturation diving system. The saturation diving system enforces the presence of combination of certain important gases like helium and oxygen for the diver.

Without proper saturation diving system, the diver has to go very deep into the oceanic water, which could cause complications like lack of air leading to suffocation (Fig. 8).

Crane Vessel As the name suggests, a crane ship is that kind of seagoing vessel which has a crane attached to it. A crane vessel is of great significance when it comes to the aspect of constructing structures in the high seas. It is only because of such vessels that many important constructions are carried out in trickier parts of the seas and oceans.

The role and scope of a crane ship is similar to the cranes that are used in day-to-day construction and hauling business activities. The only difference is the fact that the former variation of crane is used in the seas while the latter on firm ground.

The designing of the crane vessel has changed in the 800 years of its origin. The first crane ship was created in the fourteenth century, and since then the technology has helped generations and generations of people. In today's times, along with the basic ship that carries a single crane attached to it, there are also concepts like "semisubmersible vessels" and the "sheerlegs."



Offshore Vessel, Fig. 8 Diving support vessel. (<https://www.wikimedia.org>)

All such crane vessels are capable of lifting heavy tonnage, but different varieties offer different features and USP to their clients. The main difference can be elaborated as follows:

- **Common crane vessels:** These types of naval cranes are more commonly known around. They were the ones that were first introduced in the fourteenth century, as mentioned above. These cranes can be used to haul and lift around 2500 t. Additionally, another major feature is that it is movable, which means, the crane can be moved to the place where the item to be lifted is located, thus offering lots of flexibility.
- **Semisubmersible:** These types of cranes offer a lot of stability to the equipment that is being carried. As the name goes, semisubmersible cranes submerge partially into the water to give the weight placed on top of them the balance required. This balance provided ensures that the item carried does not topple into the water and thereby cause serious problems not just to the business concerned but also to the marine ecosystem. The weight carrying capacity of such semisubmersible cranes varies

from one naval vessel to another. However, the heaviest limit that such semisubmersibles can carry extends to around 14,000 t.

- **Sheerlegs:** These types of cranes are immovable. In other words, the weight that has to be loaded on them has to be brought to them so that they can be hauled. The weight carrying capacity of such cranes varies from around 50 t to around 4000 t.

A crane vessel has enabled to solve the trickiest aspect of construction. Crossing the ocean and helping widespread places to come even closer have become so much easier. Also, in today's times, with the help of a crane ship, important oil rigs are constructed so as to enable the world to get precious oil from oceanic sources (Fig. 9).

Pipe Laying Vessel A pipe laying ship is a maritime vessel used in the construction of subsea infrastructure. It serves to connect oil production platforms with refineries on shore. To accomplish this goal, a typical pipe laying vessel carries a heavy lift crane, used to install pumps and valves, and equipment to lay pipe between subsea structures.



Offshore Vessel, Fig. 9 Crane vessel. (<https://www.MarkFromSavannah/wikipedia.com>)

Lay methods consist of J-lay and S-lay and can be reel-lay or welded length by length. Pipe laying ships make use of dynamic positioning systems or anchor spreads to maintain the correct position and speed while laying pipe.

Recent advances have been made, with pipe being laid in water depths of more than 2500 m.

The term “pipe laying vessel” or “pipe layer” refers to all vessels capable of laying pipe on the ocean floor. It can also refer to “dual activity” ships. These vessels are capable of laying pipe on the ocean floor in addition to their primary job. Examples of dual activity pipe layers include barges, modified bulk carriers, modified drill ships, and semi-immiscible laying vessels, among others (King 2008) (Fig. 10).

Development History

The history of offshore vessel originates from ocean vessels. In the seventeenth–eighteenth century, the great value of trade from India and the East Indies prompts the various East India companies – and particularly those of England and Holland – to invest in magnificent ocean-going

merchant ships. These ships needed to have capacity for storing large amounts of cargo; they need to be strong and well-armed to fight off pirates or even the ships of rival companies; and they need to be comfortable for their captains and for important passengers, busy making fortunes in the east.

These were the most beautiful ships of that time period. The largest classes, outdoing even the biggest warships, were 1200 t in weight. The largest ships of this kind were 50 m long and 12 m wide. With a ratio of 4:1 between length and width, these ships could not achieve a big speed. Speed was not a crucial factor for the East India companies, since they enjoyed the monopoly anyways. The ships used trade winds as assistance in the journey and made just one journey in and out from east per year. With the nineteenth century bringing competition, speed of merchant ships became a factor and the age of clippers came.

As a result of the growing demand for a more rapid delivery of tea from China, in 1843 the Clipper era begun. Clippers were very fast-sailing ships with three or more masts and a square rig.



Offshore Vessel, Fig. 10 Crane vessel. (<https://www.wikipedia.com>)

The demand for clippers continued during the “Golden Rush” when there was a need for quick ships that were used for the transportation of gold.

The decline in the usage of clippers started with the gradual introduction of steamships. Although clippers could reach much higher maximum speed than steamships, they depended on the wind and thus could not be accounted as punctual. As for steamers, they could keep the schedule and this made them more reliable.

The industrial revolution, new mechanical methods of propulsion, and the ability to construct ships from metal triggered an explosion in ship design. The quest for more efficient ships, the end of long running and wasteful maritime conflicts, and the increased financial capacity of industrial powers created an avalanche of more specialized boats and ships. Specialized ships built for entirely new functions, such as firefighting, rescue, and research, also began to appear (Woodman 2005). With the development of offshore engineering, the offshore vessel has also been born.

Key Applications

Offshore vessels are ships that specifically serve operational purposes such as oil exploration and construction work at the high seas.

There are a variety of offshore vessels, which not only help in exploration and drilling of oil but also for providing necessary supplies to the excavation and construction units located at the high seas. Offshore ships also provide the transiting and relieving of crewing personnel to and from the high seas’ operational arenas, as and when necessitated.

Cross-References

- FPSO
- Ice Breaking Vessel
- Polar Merchant Vessel
- Research Ship
- Service Ships
- Special Marine Vehicle

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Offshore Wind Turbine

- [Shallow Foundations](#)

Offshore Wind Turbine-Ice Interactions

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Introduction

Climate change caused by the excessive use of fossil fuels has become one of the hot topics in the last few decades and threatens human

security and societal development. Serious environmental pollution and energy crises cause people and governments to look for sustainable energy resources. Offshore wind energy is regarded as one of the fastest-growing renewable energy resources in the world. Compared with onshore wind energy, the deployment of offshore wind energy offers another promising solution because of the improved wind conditions, unlimited sites, negligible visual impact, and the possibility of installing extra-large wind turbines (Perveen et al. 2014).

Offshore wind energy has significant potential in northern regions with cold climates, such as Northern Asia, North America, and Northern Europe. In these regions, wind turbines may experience icing on the blades or drifting ice during wintertime. A great potential for offshore wind is ready to be tapped in the Baltic Sea, with more than 16 GW of Baltic projects (Recharge News 2016). The Icebreaker project of Lake Erie Energy Development Co. (LEEDCo) would include six wind turbines for a total generating capacity of up to 20.7 MW in Lake Erie, north of Cleveland. According to China's 13th Five-Year Plan for offshore wind energy (NDRC 2017), nearly 900 MW of offshore wind projects will begin construction in the Bohai Bay Rim area by 2020.

Bottom-fixed wind turbines are very suitable for the shallow and intermediate water depths (<60 m). In addition to the aerodynamic and hydrodynamic loads, ice loads are also an important design consideration for offshore wind

turbine (OWTs) in ice-infested waters. The design standards for lighthouses, bridges, and offshore platforms supply sufficient information about structural design against ice loads; however, as a high-rise flexible structure (a large RNA located 100 m above the MSL for 5 MW wind turbine), OWTs possess more sensitive dynamic responses and complex dynamic behaviors than offshore oil and gas platforms. The ice load models based on those standards and guidelines are not always suitable for the design of OWT support structures due to the excessive dynamic effects that are typically present in the OWT structures. Therefore, the challenges related to ice loads on OWTs (Fig. 1) have not yet been investigated in detail.

Icing on the OWT turbine and support structures can increase the gravitational and inertial loads, and icing on the blades will increase the mass and modify their aerodynamic performance, with possible effects on aerodynamic loading and power production (Etemaddar et al. 2014a, b). Drifting ice can induce dynamic loads on the support structures, and dynamic interactions with the breaking drifting ice around the structure can excite the structure's natural frequencies. Ice loads on the OWT support structures in cold-weather regions can add risks and increase the cost of construction and maintenance (Salo and Syri 2014). This challenge calls for a detailed study of the ice-structure interactions for OWTs.

An overview of the environmental conditions, including the ice conditions for the Baltic Sea, Bohai Rim area, and Great Lakes, is given in

Offshore Wind Turbine-Ice Interactions,

Fig. 1 An offshore wind turbine at Finland



section “[Metocean Conditions](#).” The icing on the blade and the ice failure modes of the drifting ice on the OWT support structure are discussed in section “[Ice Issues for Offshore Wind Turbines](#).” To design and certify OWTs for use in cold regions, we have to follow different design standards. Section “[Ice Issues for Offshore Wind Turbines](#)” summarizes the current existing ice load models from various international standards. Section “[Numerical Analysis of Ice Load Effects on OWTs](#)” summarizes the current ice load models from various international standards. This section also discusses the numerical analysis methods of the ice-structure interactions for OWTs. It requires consideration of the wind turbine and support structures as a whole. Experimental analyses that give deeper insight into the physical loads and responses have an important role in concept verification and validation of the numerical models (section “[Experimental Analysis of Ice Load Effects on OWTs](#)”).

Metocean Conditions

Ice Conditions in the Baltic Sea

Local ice conditions are important when designing an offshore structure such as an OWT. A comprehensive overview on the seasonal ice conditions occurring onsite is paramount when trying to determine the loads imposed on the OWT support structures, for example, by moving level ice and ice ridge impacts. To optimize the support structure design for locations in the Baltic Sea, mapping of the possible ice conditions occurring onsite is very important.

In the literature, several studies contain statistical data with regard to the seasonal ice conditions in the Baltic Sea and the maximum level ice thicknesses, ice drift speeds, and ice ridge characteristics specific to the Bay of Bothnia (Tikanmäki et al. 2012). Most work makes use of seasonal ice measurement data and daily ice charts from the Finnish Meteorological Institute (FMI). These very specific data contain measured ice thicknesses (Fig. 2), ice concentration, ice drifts, and ice ridge occurrences.

The thickness of the free level ice in the Bay of Bothnia can reach anywhere between 0.3 m and 0.9 m, according to measurements performed over the last 50 years (Määttänen 1999). While land-fast ice shows limited signs of movement, free ice features can show high ice drift speeds.

Ice Conditions in Bohai Bay

Offshore wind energy is an important component of China’s National Energy Development Strategy. According to the Chinese 13th Five-Year Plan for offshore wind energy (NDRC 2017), 15,000 MW offshore wind farms will be constructed, nearly 900 MW of which will begin construction in the Bohai Bay Rim area by 2020. As the only high-latitude cold sea area, Bohai and the northern sea area of Huanghai have great potential for the development offshore wind energy in China (Fig. 3) (IEA 2012). The huge wind energy potential is accompanied by the threat of sea ice and freezing. There is a large challenge in deploying OWTs in such an area because we have to ensure the safe and sustainable operation of wind turbines under the special threat of cold weather.

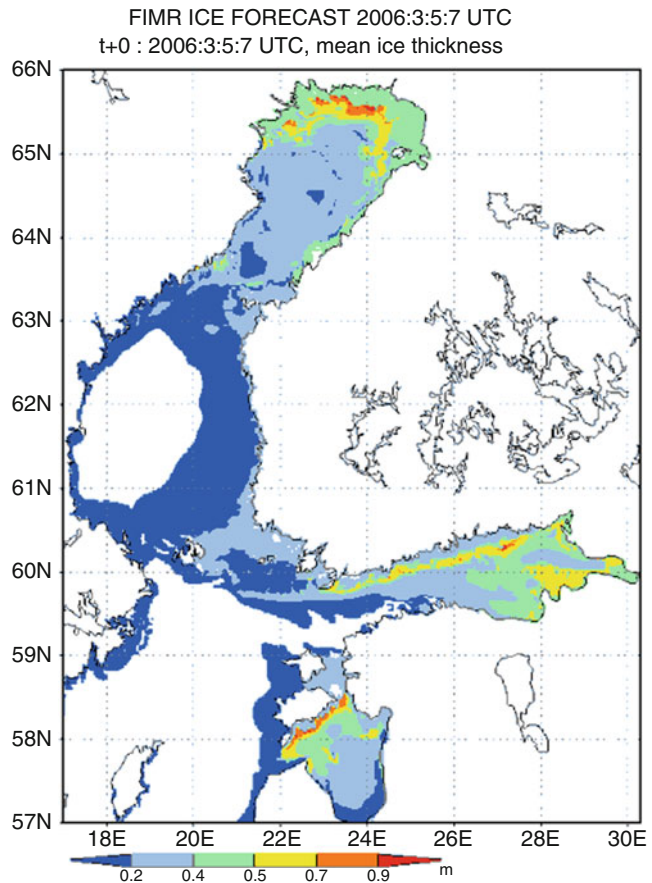
According to the “Regulations for offshore ice conditions and applications in the China Sea” (CNOOC 2002), the frozen sea area of China, Bohai, and the northern part of Huanghai are divided into 21 ice zones, as shown in Fig. 4. We can easily identify the design thickness of the level ice, the corresponding crushing strength, the flexural strength, and the ice drifting speed and drifting direction in different ice zones from the aforementioned regulations.

Ice Conditions in the Great Lakes

For the development of offshore wind energy in the Great Lakes, an important technological challenge is the presence of ice. There are exceptionally harsh winters in which the ice covers nearly 100% of the surface of the Great Lakes. The impact of ice on power-generating turbines operating offshore in the harsh winter weather of the Great Lakes is a major challenge. Lakes Superior, Huron, and Erie are often substantially covered with ice and have wind-driven ice floes at

Offshore Wind Turbine-Ice Interactions,

Fig. 2 Distribution of mean ice thickness (Finnish Meteorological Institute)



times. Figure 5 shows the estimated maximum thickness of the level ice in the Great Lakes.

Ice Issues for Offshore Wind Turbines

Icing Mechanism on the Wind Turbine Blades

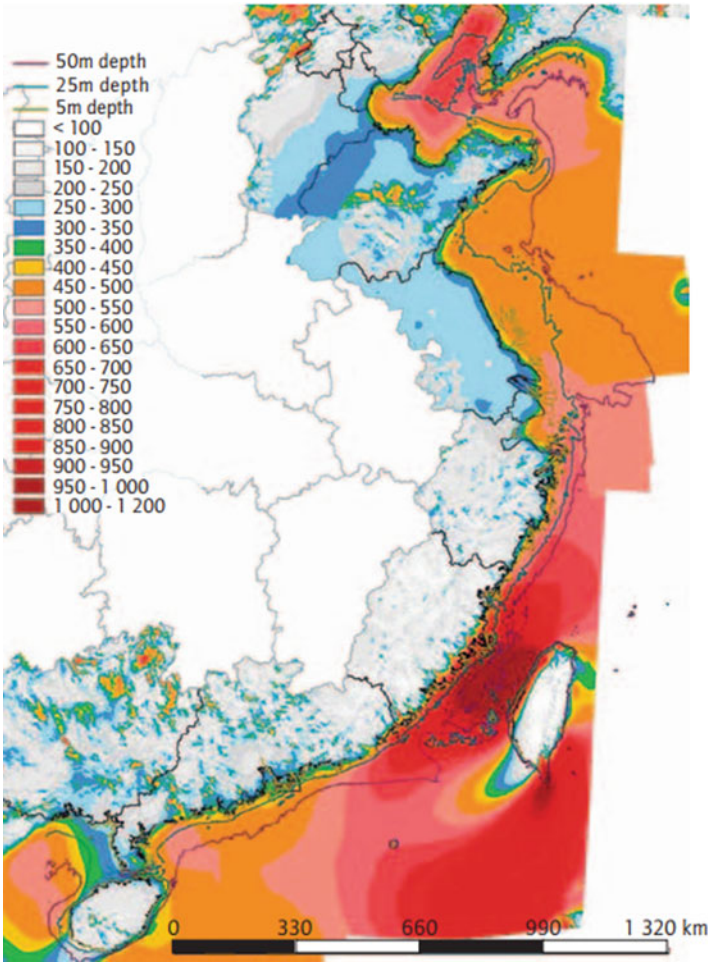
For an OWT in cold regions, at the water surface level and above, there is the buildup of atmospheric icing on turbine blades and support structures due to water in the air as in-cloud operations, rainfall, and sea sprays. Icing on the OWT turbine and support structures can increase the gravitational and inertial loads, and icing on the blades will increase the blade mass and mass imbalance and change the blade surface roughness, thus modifying their aerodynamic performance, with possible effects on aerodynamic loading and power production (Fig. 6). Additionally, ice buildup on turbine blades can lead to ice throw

or shredding, where fragments of mixed ice and snow fall off the blades. The size, speed, and distance of falling fragments vary and depend on the climate conditions and turbine operation. Icing of rotor blades and other wind turbine components have an effect on the design of turbines, the safety of O&M personnel, and the overall economics of a wind energy project. Anti-icing “paint” and deicing or anti-icing systems were developed for modern wind turbines to prevent the formation of ice and to detect and efficiently remove ice formed on wind turbine blades while letting the turbines operate at full power.

The atmospheric ice accumulation on wind turbine blades and its effect on the aerodynamic performance and structural response were investigated using a numerical method (Etemaddar et al. 2014a). Below the rated wind speed, for a 5 MW NREL wind turbine, a power loss up to 35% is observed, and the rated power is shifted

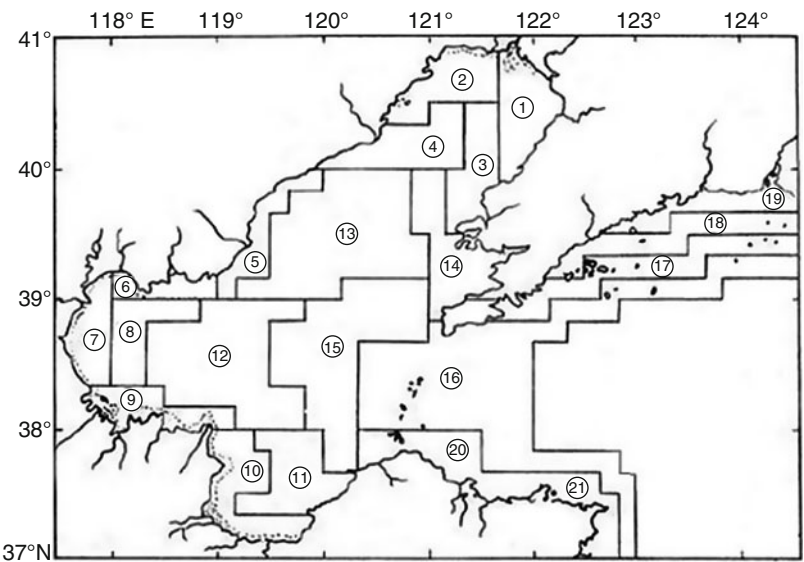
Offshore Wind Turbine-Ice Interactions,

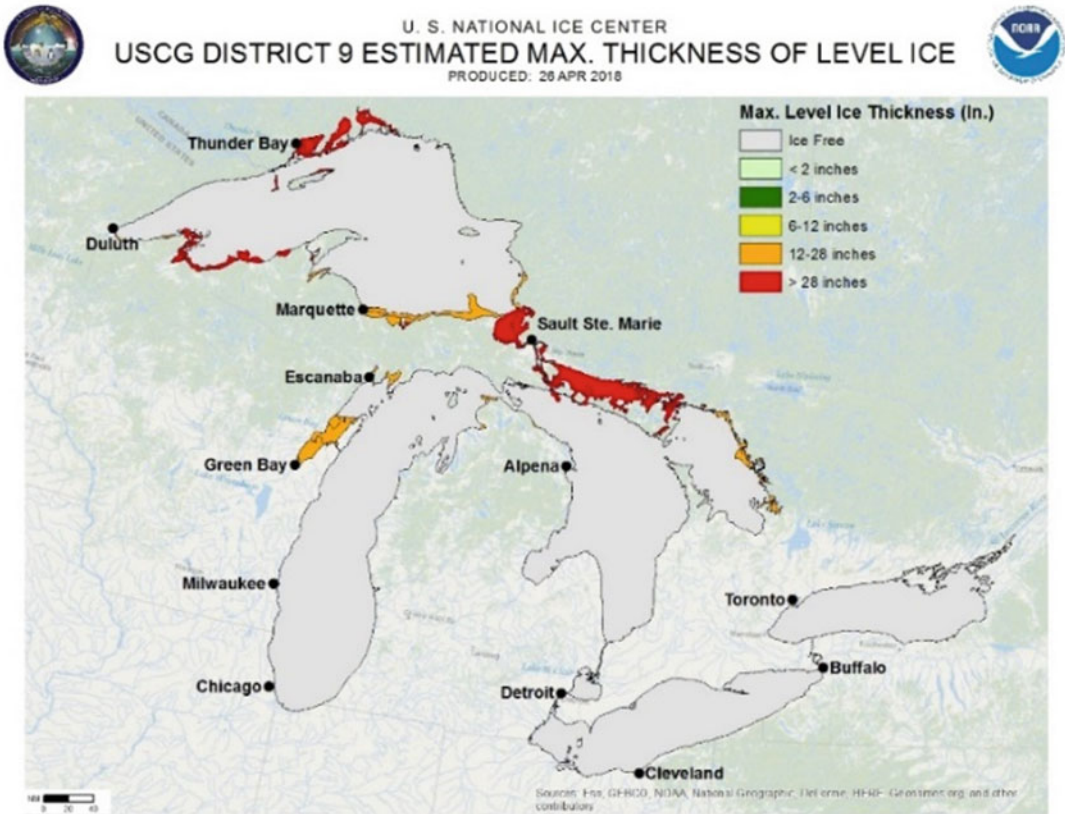
Fig. 3 Distribution of annual average wind power densities in 5–50 m depth areas (IEA 2012)



Offshore Wind Turbine-Ice Interactions,

Fig. 4 Ice zones map in the China Sea (CNOOC 2002)





Offshore Wind Turbine-Ice Interactions, Fig. 5 Great Lakes ice chart based on estimated thickness in inches (National Ice Center 2018)



Offshore Wind Turbine-Ice Interactions, Fig. 6 Icing on the wind turbine blades

from a wind speed of 11 to 19 m/s (Fig. 7). The thrust of the iced rotor below the rated wind speed is less than that of a clean rotor by up to 14%,

whereas above the rated wind speed, it is up to 40% greater than that of clean rotor.

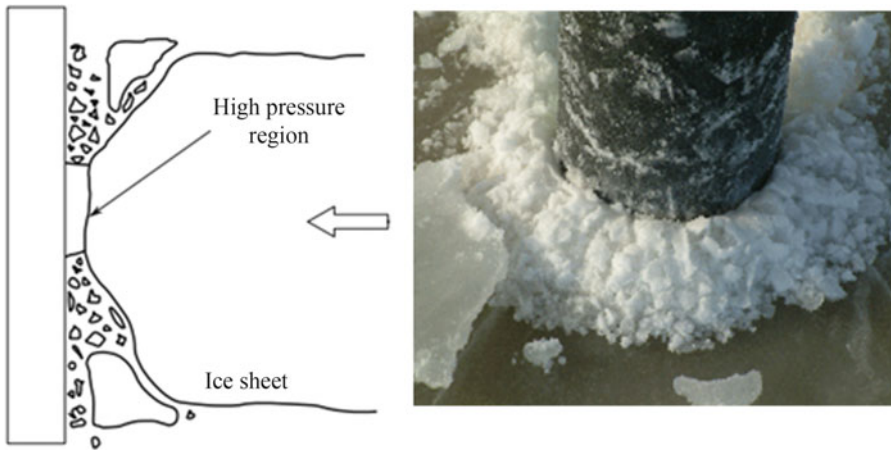
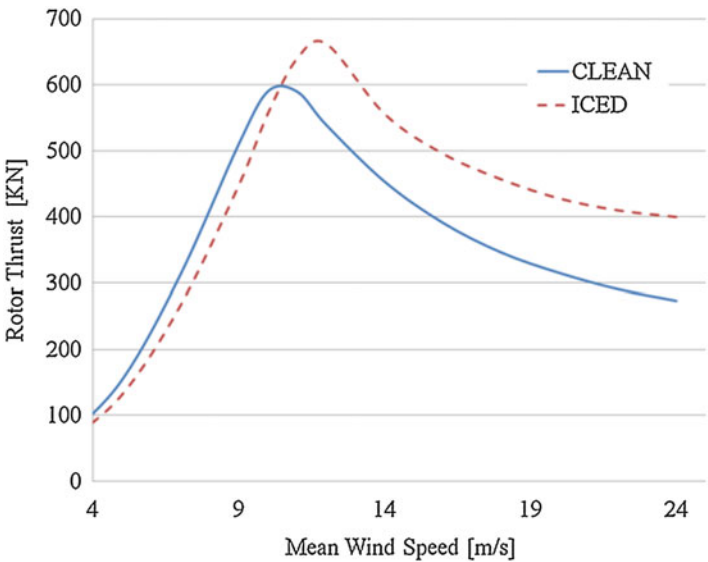
Drifting Ice Failure Modes on the Wind Turbine Support Structures

Offshore, the drifting level ice or an ice ridge on the sea surface can induce additional static and dynamic forces on the turbine support structure. The effects of sea ice occur as mechanical shocks and increased vibrations that may result in additional operational loads.

Due to differences in the ice thickness, the ice drifting speed, and the geometry of the structure, ice failure can be classified (ISO 19906 2010) as a crushing mode, which typically occurs against vertical structures (Fig. 8), and a flexural mode, which typically occurs against inclined structures. Depending on the ice speed, the crushing mode

Offshore Wind Turbine-Ice Interactions,

Fig. 7 Mean values of aerodynamic thrust before and after icing



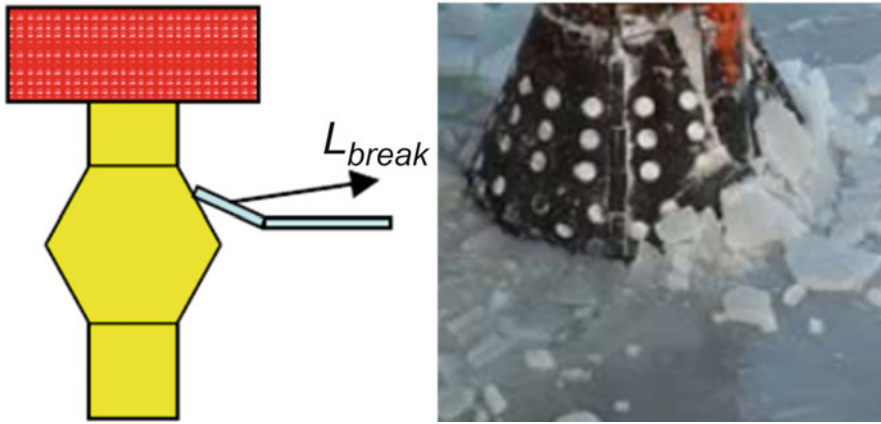
Offshore Wind Turbine-Ice Interactions, Fig. 8 Level ice failure against a vertical structure (Xu et al. 2010)

can be identified (Yue et al. 2009; ISO 19906 2010) as follows:

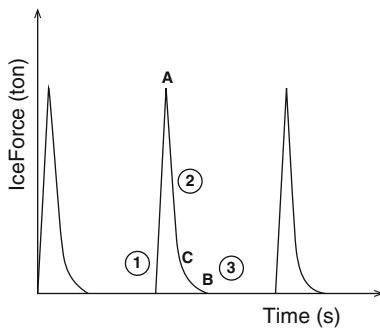
1. Intermittent ice crushing at low ice velocities, where the ice load time series exhibit a saw-tooth waveform with a specified period that is separated by unloading intervals
2. Frequency lock-in crushing at moderate ice velocities, where the ice load time series exhibit a periodic waveform at a specified fundamental frequency of the support structure

3. Continuous brittle crushing at high ice velocities, where the time series of both the ice loads and the structural response appear to be random

A widely used structural approach is to design the structural member with an incline/decline at the seawater level. A conical structure is commonly used for this purpose (Fig. 9). The cone forces the ice sheet to bend when it collides with the structure, which has two advantages: (1) the failure stress level is much easier to reach for



Offshore Wind Turbine-Ice Interactions, Fig. 9 Level ice failure against a conical structure (Xu et al. 2010)



Offshore Wind Turbine-Ice Interactions, Fig. 10 Level ice bending failure pattern against a conical structure: ① loading, ② unloading, ③ zero-stable force (Xu et al. 2010)

bending compared to compression, and (2) the failure level for ice is much less when bent than when compressed, which is typical for many materials. A typical ice load pattern for the bending failure mode could be identified as the loading, unloading, and zero-stable force phases (Fig. 10).

Numerical Analysis of Ice Load Effects on OWTs

Ice Load Models from International Standards

The ice load models regarding both vertical and conical structures in IEC and ISO standards are summarized in this section.

For the monopile (or vertical cylinder structures), IEC 61400-3, DNV-OS-J101 (DNV 2014), and the GL Guideline utilize the Korzhavin equation to estimate the global static force on vertical structures:

$$F = k_1 k_2 k_3 h D \sigma_c \quad (1)$$

where k_1 is the shape factor for the shape of the support structure on the ice impact side; k_2 is the contact factor for the ice contact against the support structure; k_3 is the factor for the ratio between ice thickness and the support structure diameter; D is the diameter of the support structure at the water line; and σ_c is the crushing strength of the ice.

ISO 19906 proposes a different approach to address the horizontal load due to ice crushing based on full-scale measurements and studies on the influence of the ice thickness and the structure width on the global ice action.

$$F_{cr} = p_G h D = C_R h^n (D/h)^m h D \quad (2)$$

where p_G is the global average ice pressure, C_R is the ice strength coefficient, and n and m are empirical exponents to account for the size effect.

For the conical structures, in IEC 61400-3 (2009), Ralston's formulae, which have also been used as recommended practice in DNV and API standards (API 2007), are adopted to

calculate the static ice loads on conical structures. For downward breaking cones, the static characteristic values for horizontal and vertical forces on the cone are

$$F_H = A_4 \left(A_1 \sigma_f h_i^2 + \frac{1}{9} A_2 \rho_w g h_i D_{wl}^2 + \frac{1}{9} A_3 \rho_w g h_i (D_{wl}^2 - D^2) \right) \quad (3)$$

and

$$F_V = B_1 F_H + \frac{1}{9} B_2 \rho_w g h_i (D_{wl}^2 - D^2) \quad (4)$$

where A_i and B_i are nondimensional coefficients determined from a reference (Ralston 1977), ρ_w is the water density, and D is the cone diameter at the bottom for the downward cone.

For dynamic loading on conical structures, IEC recommends the following vertically shifted sinusoidal model:

$$F_{H,dyn} = F_H \left(\frac{3}{4} + \frac{1}{4} \sin(2\pi f_b t) \right) \quad (5)$$

where F_H is the maximum load derived from the static characteristic value (Eq. 3), $f_b = V_i/(Kh_i)$, V_i is the ice drifting speed, h_i is the ice thickness, and $4 < K < 7$. The value of K that gives the highest load that should be used.

IEC also suggests a sawtooth model for dynamic loading on conical structures as in Fig. 11.

Popko et al. (2012) studied the sea ice loads on offshore support structures by comparing different ice models from different international guidelines and standards. They indicated that the interaction of sea ice, other metocean loads, and OWT should be considered. IEC 61400-3 offers quite good provisions for ice loads. However, some of the proposed models may be considered not state of the art. ISO 19906 gives a comprehensive set of ice loads for the Baltic Sea region. The DNV document lacks a broad description of ice loads but is harmonized with the IEC standard. Basic provisions for ice loads are given in both GL guidelines. Modern standards for the ice load prediction of offshore structures were reviewed by Frederking (2012).

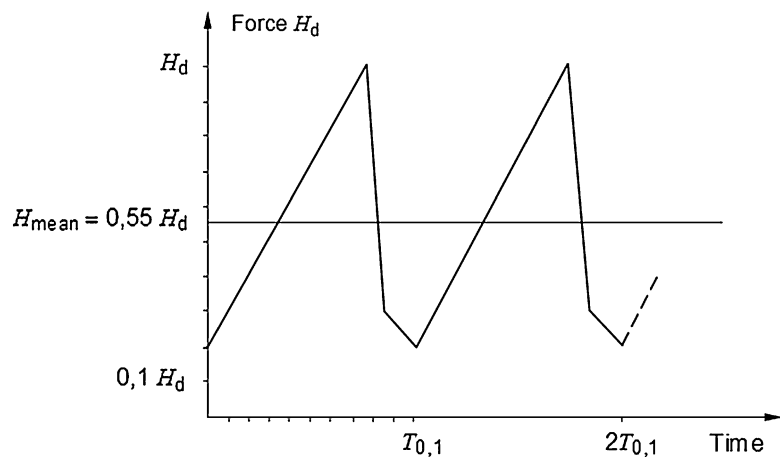
Numerical Analysis of Ice Loads Effects on OWT

The extents of ice cover, ice thickness, and ice strength are highly variable over space and time. The forces exerted on the wind turbine structure are associated with the geometry, motion, and failure strength of ice. Therefore, the ice forces are also highly variable. In contrast to offshore oil and gas platforms in cold regions, not only are offshore wind turbines subject to coupled forcing from ice, winds, waves, and currents, but their structural responses are also highly influenced by turbine dynamics and forces.

For offshore structures used in the oil and gas industry in cold regions, ice loads may be dominant over wind and wave loads (McGovern and

Offshore Wind Turbine-Ice Interactions,

Fig. 11 IEC sawtooth ice load model (IEC 61400-3 2009)



Bai 2014). Many studies have been conducted to predict the ice loads on cylindrical or sloped offshore structures. Bekker et al. (2009) carried out simulations of the ice-structure interactions between the ice cover and the offshore structures. Bekker et al. (2013) estimated the contact interaction of an ice hummock and ice floe to predict the ice load on a cylinder using commercial CFD software. Zvyagin and Sazonov (2014) used a probabilistic model to simulate the stochastic ice loads. Taylor and Richard (2014) developed a probabilistic ice load model based on an empirical method. Spencer et al. (2014) used the quantile regression method to analyze global ice pressure data for a wide range of structures.

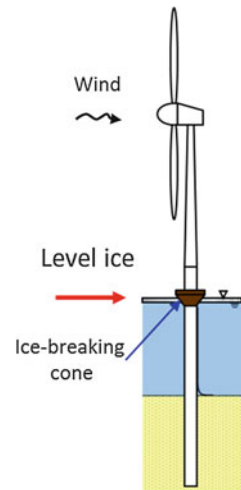
For a 5 MW OWT on a monopile with a diameter of 6 m interacting with drifting ice that has a thickness 0.5 m, the ice-structure interaction could generate dynamic forces as high as a maximum of 15 MN under conservative simplifications (Nguyen et al. 2014). Ice loads on the support structures for OWTs in cold regions could add risks and increase the costs of construction and maintenance (Salo and Syri 2014).

Conical structures have been suggested for most offshore structures where ice is present (Xu et al. 2014). The cone reduces the ice load magnitude and ice-induced structural response compared to a cylindrical structure with the same waterline diameter (Barker et al. 2014).

Wind turbine design and analysis rely on the use of aero-hydro-servo-elastic simulation tools to predict the coupled dynamic loads and responses of an OWT; a limited number of studies have considered fully coupled analyses, which simultaneously investigate the effects of both aerodynamic and ice loads on the behavior of offshore wind turbines. The excitation of a complete OWT structure and the vibration of its blades were reported by Hetmanczyk et al. (2011) in a numerical study on the dynamic ice loads on an OWT. Gravesen and Kärnä (2009) focused on the static ice crushing occurring on a vertical structure of an OWT in the South Baltic Sea based on the ISO standard (2010) with necessary corrections for the ice reference strength. The mitigation of ice-induced vibrations in an OWT was studied using a semiactive model (Mróz et al. 2008)

Offshore Wind Turbine-Ice Interactions,

Fig. 12 Monopile wind turbine with ice-breaking cone (Shi et al. 2016)

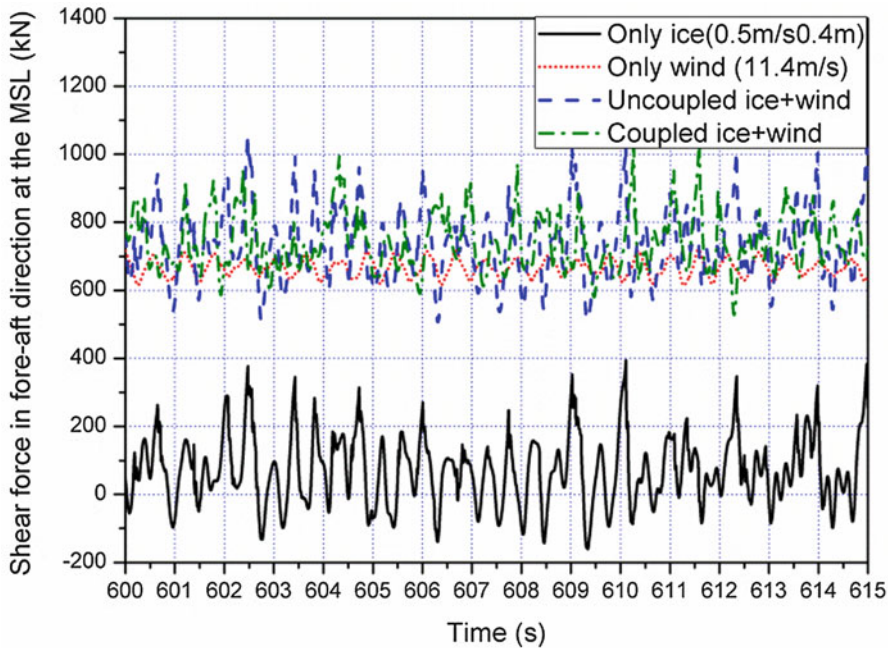


and a semiactive tuned mass damper (STMD) system (Kärnä and Kolari 2004). Fully coupled simulations are shown to have lower standard deviations than the superposition of responses under wind-only and ice-only loads (Shi et al. 2014, 2016, 2018) (Fig. 12). Figure 13 compares the shear force at mean sea level for a monopile OWT under various load conditions.

Experimental Analysis of Ice Load Effects on OWTs

Serious ice-induced vibrations of offshore structures have been seen to occur in situ. A limited number of studies measured the ice loads and responses in model tests and field observations involving ice-induced vibrations (Jefferies et al. 2011). Määttä et al. (2011) conducted nearly full-scale ice crushing tests in the Aker Arctic test basin. Frequency lock-in crushing may cause severe vibration at certain ice velocities for cylindrical structures (Bjerkås et al. 2014; Xu et al. 2014; Xu and Yue 2014). It is also important to determine the force and frequency of load cycles for fatigue assessment (Hendrikse et al. 2014). This phenomenon is particularly addressed in the ISO (2010) and IEC standards (2009).

Murray et al. (2009) did model tests on scales of 1:30 and 1:50 to predict the ice loads of an ice-resistant spar design. Nord et al. (2015) identified



Offshore Wind Turbine-Ice Interactions, Fig. 13 Comparison of the shear force in the fore-aft direction at MSL under various load conditions (Shi et al. 2016)

the force induced by level ice on a generic bottom-fixed offshore structure using a joint input-state algorithm to describe experimental ice-induced vibrations. Full-scale testing was conducted by Yue et al. (2009) on a cylindrical monopile in Bohai Bay to investigate the dynamic ice forces and structural vibrations; three speed-dependent ice force modes were observed. Following their previous work, Xu and Yue (2014) experimentally investigated the dynamic ice force on a jacket structure with an upward/downward ice-breaking cone in the Bohai Sea.

Barker et al. (2005) and Gravesen et al. (2003, 2005) performed extensive model testing to investigate the ice-induced vibrations in OWTs in Danish waters. Compared to wind loads, increases of 60% in the maximum force and 10% in the bending moment were estimated for large wind turbines in Denmark (Fig. 14).

Conclusions and Final Challenges

The development of offshore wind energy in cold regions is very attractive; however, it is a big

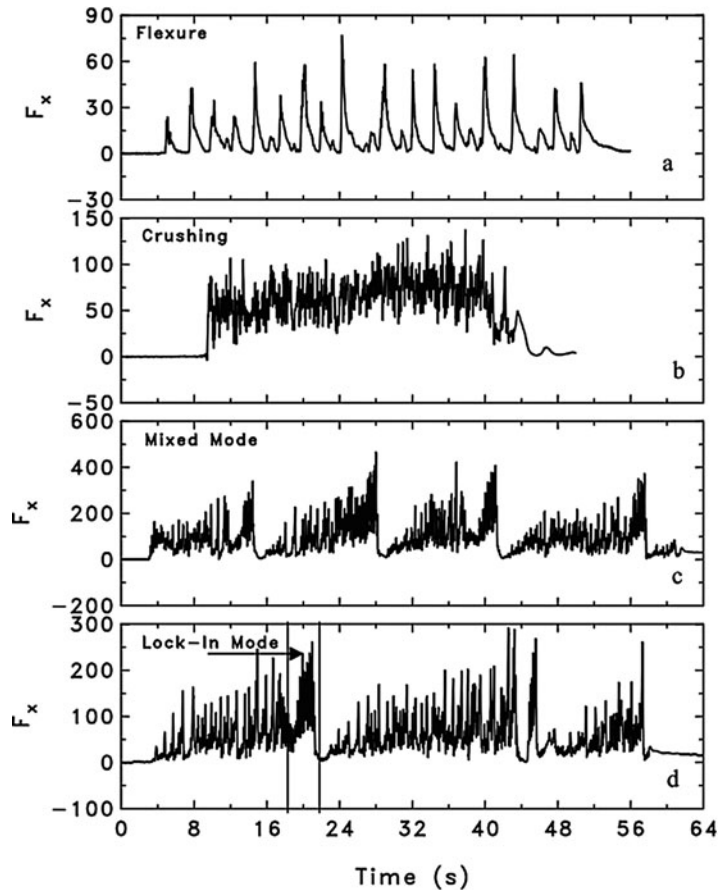
challenge to deploy OWTs in those regions due to the icing on the turbine blades and drifting ice at the mean sea level during the wintertime. Both icing and drifting ice can have a big effect on the power production and dynamic response of the OWT system. The ice load models from international standards are based on offshore oil and gas industry, which is not well suitable for OWT design and analysis. Great effort has been dedicated to develop fully coupled numerical model for OWTs to take into account the ice load effect. Some experimental studies have been carried out in model scale to validate various ice load numerical models. Numerous countries, in particular the United States, China, and Finland, have aspiration to take full advantage of the offshore wind resources in the Great Lakes, Bohai Bay, and Baltic Sea, facing the ice loads challenge for OWTs.

There remain a number of technical challenges; the key requirements are for:

- *Fully coupled numerical models*, able to simulate the wind turbine system and ice induced vibration dynamically

Offshore Wind Turbine-Ice Interactions,

Fig. 14 Typical time series traces of the failure modes observed during testing (Barker et al. 2005)



- More detailed ice-related design load cases from international standards for turbine design and certification
- A comprehensive program of ice tank test for OWT to develop more accurate ice load model specifically for OWT
- More field test data for OWT-ice interaction

Cross-References

- [Moored Ship in Ice](#)
- [Sloping Structure-Level Ice Interactions](#)

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Offshore Wind Turbines

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Definition

An offshore wind turbine is a modern device that is deployed in an offshore environment for generation of electricity from wind.

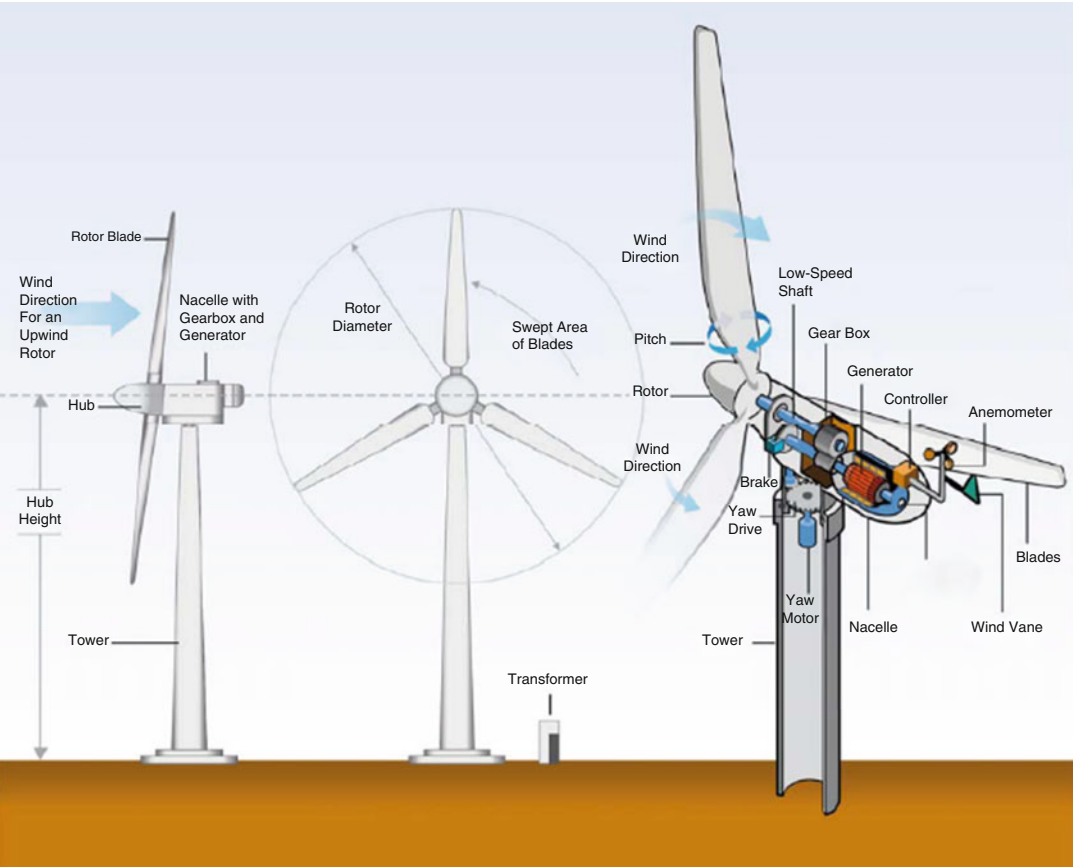
Function and Components of a Modern Wind Turbine

Wind energy is one of the most important renewable energy resources that are gaining increasing attention and significant development. A wind turbine is used to convert the kinetic energy in wind first to the mechanical energy through rotation of the blades and then to the electricity via a generator. Modern wind turbine, which has been commercialized and industrialized, is typically a three-blade, upwind, pitch-regulated variable speed horizontal axis wind turbine (HAWT), as shown in Fig. 1 (National Renewable Energy Laboratory (NREL) 2011). An offshore wind turbine is a wind turbine at sea with either bottom-fixed or floating support structures.

A wind turbine consists of tower, nacelle, hub, and rotor blades. Inside the nacelle, there is a gearbox which transforms the low rotational speed of the rotor into the high rotational speed at the generator side. For example, the rotor rotates at 12 rpm for the NREL 5 MW wind turbine at the rated wind speed, while the generator runs at 1164 rpm, indicating a gear ratio of 1:97 (Jonkman et al. 2009). For each blade, there is a pitch actuator at the blade root, typically a hydraulic system, which connects to the hub and allows the change of the blade pitch angle for power regulation for wind speed larger than the rated value. In addition, there is a yaw mechanism which allows the rotor plane facing the main wind direction in order to maximize the power absorption. Electricity generated by each turbine are collected at the substation of a wind farm via AC cables and transmitted to the grid through DC cables.

To reduce the gravity-induced dynamic loads in blades during normal operation, wind turbine blades are made of lightweight and high-strength materials, i.e., composite materials (mainly GFRP (glass fiber reinforced polymer) and occasionally more expensive but stiffer CFRP (carbon fiber reinforced polymer)). At the blade root, steel bolts are embedded into the composite materials and are used to connect to the hub. Gearbox and tower are also steel structures, and in some cases, tower is made of concrete.

In 2017, the top ten wind turbine manufacturers are Vestas, SGRE, GE, Goldwind,



Offshore Wind Turbines, Fig. 1 Modern wind turbine (NREL 2011)

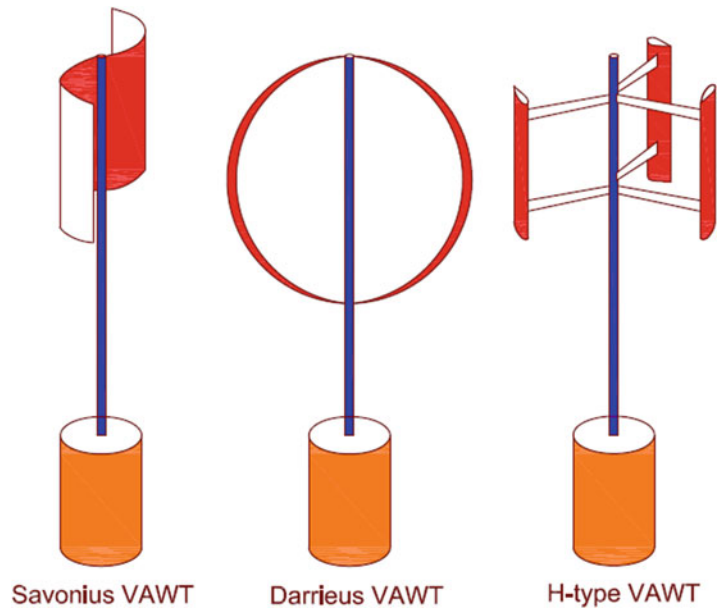
Offshore Wind Turbines, Table 1 Dimensions of MW-scale industry wind turbines

Rated power (MW)	Starting year for commercial use	Blade length (m)	Blade weight (ton)	Hub height (m)	Nacelle weight (ton)
1.3	1995	31	6	68	50
2	1999	44	10	78	75
5	2005	61.5	17	90	240
6	2013	77	25	100	280
8	2016	82	35	138	390

Enercon, Nordex, Senvion, United Power, Envision, and Suzlon (WindPowerMonthly 2017). The size of a single wind turbine, in terms of rated power, blade length, and hub height, has increased significantly in recent years because of the development of the offshore wind industry,

resulting into a big reduction in cost of energy. Rated power is the largest power that a wind turbine can generate and is mainly limited by the capacity of generator. Table 1 shows the typical dimensions of the MW-scale industry wind turbines.

Offshore Wind Turbines,
Fig. 2 Vertical axis wind
 turbines (VAWTs) (Cheng
 2016)



Thrust and Power of a Wind Turbine

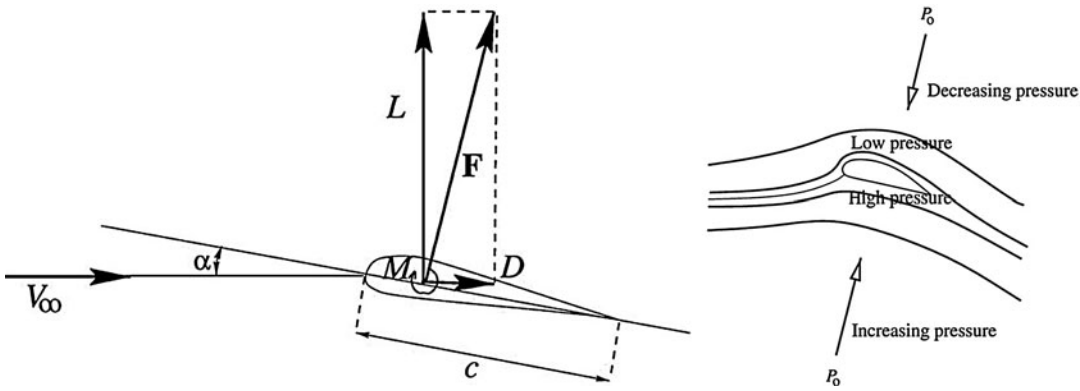
Other types of wind turbines, such as vertical axis wind turbines (VAWTs) as shown in Fig. 2, have also been proposed and tested but without a great success for industry development. As compared to HAWTs, VAWTs have a less optimal power coefficient, have large variations in aerodynamic loads and power per revolution, and are not feasible for power regulation through blade pitch control. However, VAWTs are wind direction-independent and can be combined with a generator at the platform level, which leads to a low center of gravity and can potentially improve the hydrostatic stability of floating wind turbines.

Wind turbine blade is specially designed with its aerofoils to utilize the aerodynamic lift force acting on blade cross-sections to drive the rotor to rotate and then absorb wind power. A typical aerofoil cross-section is shown in Fig. 3, in which both the lift (which is perpendicular to the direction of the local wind inflow) and the drag forces (which is along the direction of the local wind inflow) are indicated. Both the lift and the drag coefficients depend on the angle of attack of the flow, α , as shown in the figure.

The relative velocity near the blade tip induced by the rotation of the blade, for example, about

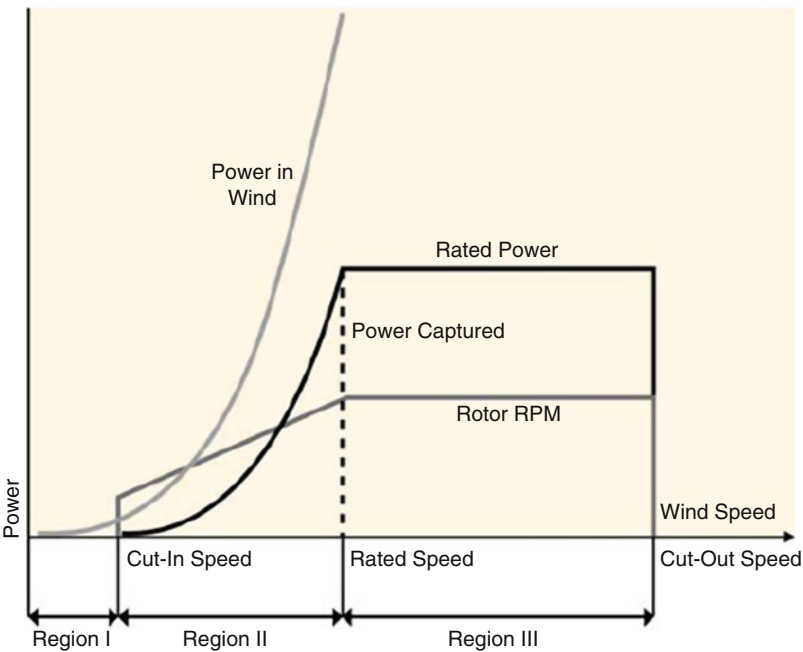
90 m/s for a 5 MW wind turbine running at the rated speed, is usually much larger than the normal wind velocity for example, about 12 m/s rated (Jonkman 2009). Therefore, the torque, which leads the rotor to rotate, is generated mainly by the lift forces acting on the cross-sections near the blade tip. The wind turbine thrust, which is the integrated force on the three blades along the wind direction, is also mainly due to the cross-sectional lift forces. Torque multiplied with rotor rotational speed gives the aerodynamic power, which is equal to the generator power for the wind turbine in a steady-state operation assuming zero energy loss. The aerodynamic efficiency, which is defined as the ratio of the aerodynamic power to the power available in wind inflow, is limited by the Betz limit, $16/27 = 59.3\%$ (Hansen 2008). Aerodynamic efficiency of modern wind turbines can reach 45–50%. The mechanical efficiency, which is defined as the ratio of the generator power to the aerodynamic power, is typically very high, 97% for drivetrain with gearbox.

Automatic control plays an important role in power production of wind turbines. The black curve in Fig. 4 presents the power curve of a typical pitch-regulated variable speed wind turbine. Wind turbine is parked, either idling for very low wind speed in Region I or standstill for



Offshore Wind Turbines, Fig. 3 Lift (L) and drag (D) forces on a wind turbine blade aerofoil (Hansen 2008)

Offshore Wind Turbines, Fig. 4 Power curve of a typical wind turbine (NREL 2011)



wind speed larger than the cut-out value. In Region II with wind speed lower than the rated value and higher than the cut-in value, wind turbine is in operation with a maximum power absorption coefficient through variable speed control, while in Region III with wind speed larger than the rated value and lower than the cut-out value, wind turbine operates to achieve constant power output via blade pitch control.

Power output of a wind turbine strongly depends on the speed of inflow wind.

However, wind speed is not constant at any offshore site, and it varies from second to second and from year to year. The capacity factor, which is defined as the ratio of the annual average power to the rated power, is used to characterize the long-term power performance of wind turbines. Because of steady wind conditions offshore, the capacity factor of offshore wind turbines is typically in the range of 40–60%, larger than that of onshore turbines, 30–40%.

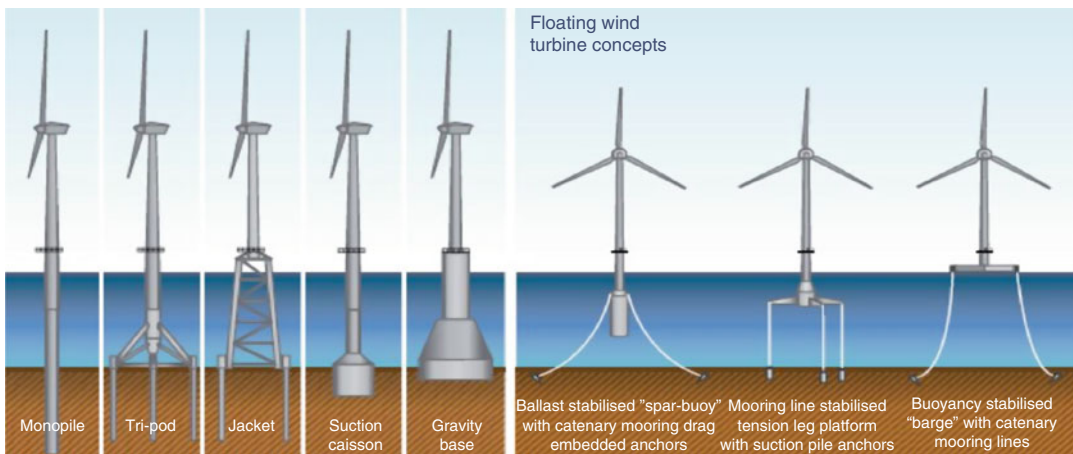
Types of Offshore Wind Turbine Support Structures

The main difference between onshore and offshore wind turbines lies on the support structures. In addition to the basic components that we see in an onshore wind turbine, as discussed above, an offshore wind turbine needs a support structure in the water. Different types of support structures have been proposed, depending on the water depth and soil conditions of the offshore site. These include bottom-fixed support structures that are mainly used in relatively shallow or moderate waters (in the order of 0–50 m of water depth), such as monopile, jacket, gravity base, tripod, and floating support structures for deep waters (with water depth larger than 50–100 m), such as spar, semisubmersible, TLP (tension leg platform) (see Fig. 5) (Wiser et al. 2011). Most of these support structure concepts were used for offshore oil and gas platforms in the past. However, the sizes of oil and gas platforms are usually larger than those used for offshore wind turbines.

Monopiles are cylindrical steel structures which are hammered to the sea bed with a depth close to the water depth. A transition piece is normally used before placing the tower and the remaining wind turbine components. It is connected to the monopile upper part with grout so that the verticality of the tower and the wind

turbine can be ensured. Typically, a monopile has a diameter of 6–8 m for water depth of 0–30 m, while the recently developed XL monopile has a diameter of 10 m for water depth of 40–50 m, supporting 6–8 MW wind turbine. Jacket support structures are typically three-leg or four-leg structures with connecting braces, which have a diameter of 1–2 m. Piles or suction buckets are used to fix the jacket structure to the sea bed. Jackets are transparent to wave loads and can provide a stiff foundation to support wind turbines. GBS (gravity base foundation) typically has a large concrete foundation which does not require a piling operation. They can be pre-assembled with the tower and the wind turbine at the shipyard, wet-towed to the offshore site and lowered down by ballasting.

Floating wind turbines are totally different as compared to the bottom-fixed wind turbines. According to Butterfield et al. (2005), they can be categorized as spar, semisubmersible, and TLP wind turbines, with respect to the floater types and the mooring systems to achieve hydrostatic stability, i.e., ballast-, water plane area- or mooring-stabilized. A spar wind turbine is a floating cylindrical steel structure with a diameter of 8–10 m and a draft of 80–100 m for supporting a 5 MW wind turbine. It uses heavy ballast at the lower part of the spar to lower the center of gravity and to provide the restoring moment in pitch. Typically, a spread catenary mooring system is used,



Offshore Wind Turbines, Fig. 5 Bottom-fixed and floating wind turbines (Wiser et al. 2011)

and in some cases, clump weights are used in combination with taut lines. A semisubmersible support structure has three or four columns separated with a distance to create a large area moment and to provide the restoring effect. The columns are typically connected by either braces or pontoons. Both spread and single point mooring can be applied. A TLP wind turbine is a floating structure stabilized by mooring lines, i.e., tethers of steel tubes. The floater can be a deep draft floater similar to a spar or a semisubmersible-type floater. Ship-type floaters have been proposed for floating wind turbines. However, because of large wave loads and wave-induced rotational motions, they are not suitable for floating wind applications.

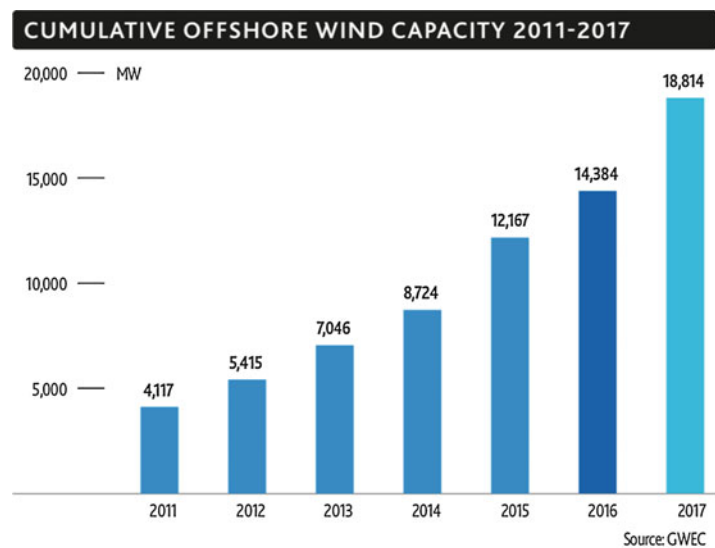
The power characteristics of different types of bottom-fixed offshore wind turbines are very similar, while those of floating wind turbines might be slightly different, because of the heeling angles that are induced by mean wind turbine thrust force and mean wave loads. Moreover, the features of dynamic responses of offshore wind turbine under simultaneous wind and wave loads might be very different for bottom-fixed and floating concepts. Blades are slender structures made of light materials, and they can deform significantly under the aerodynamic loads. Therefore, aeroelasticity should normally be considered in loads and response analysis of blades. A particular challenge is to design the steel-composite connection

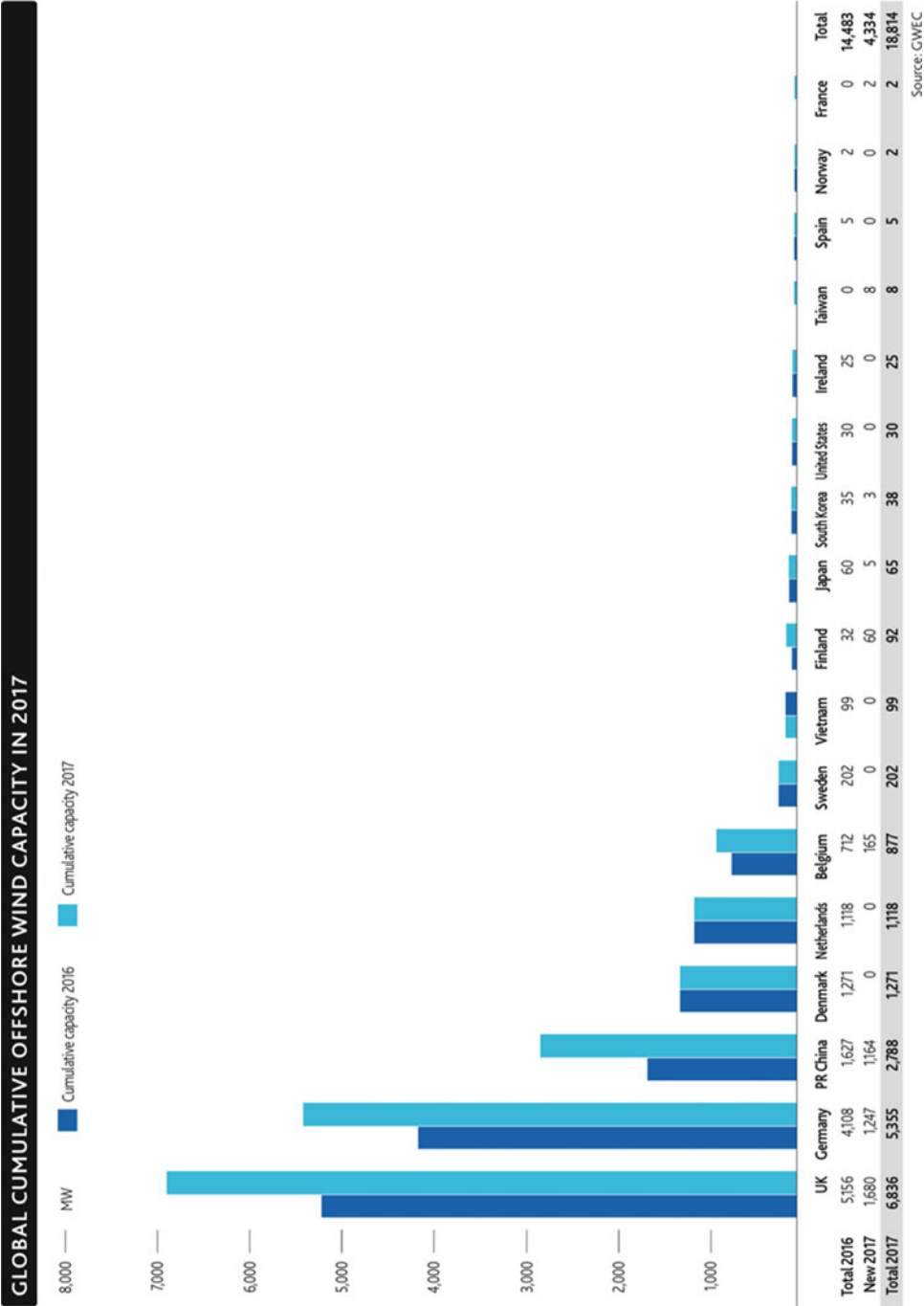
at the blade root, which takes most of the bending moment resulted from the aerodynamic loads along the blade. For bottom-fixed offshore wind turbines, the dynamic responses in blades, tower, and foundation are governed by the quasi-static responses due to both slowly varying wind loads and first-order wave-frequency wave loads as well as the resonant dynamic responses at the first and the second eigenmodes due to wind loads. For floating wind turbines, there are more prominent coupling effects between the wind- and wave-induced loads and responses. In addition to the rigid-body motions of floating wind turbines at both the resonant low frequency and wave frequency, there exist vibrational responses at the first eigenmode that are important for design of tower and blades.

Recent Industry Development

As compared to onshore wind turbines, offshore wind turbines today still represent a very small proportion, about 3%, of the total installed wind capacity. However, in the last 20 years, there is a significant development of offshore wind turbines with an average annual increase of 30% in installed capacity. By the end of 2017, it reaches 18.8 GW worldwide (GWEC 2018), as shown in Fig. 6.

Offshore Wind Turbines,
Fig. 6 Cumulative
offshore wind capacity
2011–2017 (GWEC 2018)





Offshore Wind Turbines, Fig. 7 Global cumulative offshore wind capacity 2017 (GWEC 2018)

Among different countries, UK and Germany are the two countries with the most installed capacity, followed by China, Denmark, and the Netherlands, as shown in Fig. 7.

Typically, offshore wind turbines are developed and commercialized through a farm configuration with 50–100 units. Bottom-fixed wind turbines are the majority of today's offshore wind farms, and in particular, monopile foundations count for more than 85% of the support structures. A general trend in recent years' development is that larger-scale wind turbines are being developed and installed in deeper water and farther away from shore. Up to now, there exist only 11 floating wind turbines of MW scale that are under the sea tests, including the small wind farm of 5 units with Hywind spar wind turbines in Scotland. Although there exists a significant potential for using floating wind turbines, they only counts for a few percentage of the total offshore wind capacity even for 2030.

Cross-References

- [Design Rules and Standards](#)
- [Economic Assessment](#)
- [Marine Operations](#)
- [Mooring System of Renewable Energy Devices](#)

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On-Bottom Stability of Submarine Pipelines

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Introduction

The on-bottom stability (also termed as “geotechnical stability”) of submarine pipelines mainly involves vertical, lateral, and axial pipeline-seabed interactions under ocean waves and/or current and engineering operating conditions.

Vertical Stability of the Pipeline on and in Seabed

Submarine pipelines that are intended to be buried in the seabed should be checked for possible sinking or floatation in order to satisfy the vertical stability on and in the soil. For the pipelines to be laid on the seabed with low shear or cohesive strength, the bearing capacity of the soil needs to be evaluated. If the seabed soil is, or is likely to be, liquefied under pure waves or combined waves and current, the liquefaction depth should be predicted accurately. Seabed liquefaction can affect both the vertical stability, i.e., sinking and floatation, and the lateral stability of submarine pipelines (see Det Norske Veritas and Germanischer Lloyd 2017).

Ultimate Bearing Capacity

In geotechnical engineering, the ultimate bearing capacity is defined as the theoretical maximum pressure which can be supported without failure by the soil immediately below and adjacent to a foundation. Submarine pipelines can be regarded as shallow foundations installed on the seabed. With reference to a strip footing, three distinct modes of failure have been identified, i.e., general shear failure, local shear failure, and punching shear failure. The evaluation for the ultimate bearing capacity can be considered in terms of plasticity theory, e.g., the slip-line stress field theory, the lower and upper bound theorems in limit analysis (Chen and Liu 1990).

The seabed soil can be essentially assumed to behave as a Tresca material under undrained conditions, or as a Mohr-Coulomb material under drained conditions. Taking into account the effects of the geometric curvature of the pipe and the adhesion or friction at the pipe-soil interface, the slip-line field solutions for the undrained and fully drained conditions can be derived, respectively (Gao et al. 2013, 2015). A general slip-line field solution for the ultimate bearing capacity of a pipeline on the soil obeying Mohr-Coulomb yield criterion is expressed as (Gao et al. 2015):

$$\frac{P_u}{D \sin \theta} = cN_c + qN_q + (0.5D\gamma' \sin \theta)N_\gamma \quad (1)$$

where P_u is the collapse load for the soil suffering general shear failure (in kN/m); D is the outer diameter of the pipeline (in m); $\theta = \arccos(1 - 2e/D)$ is the embedment angle (in degree), e/D is the embedment-to-diameter ratio; “ $D \sin \theta$ ” refers to the efficient width of the pipe-soil interface, which is related to the pipe penetration; c is the soil cohesion (in kPa); q is the surcharge pressure (in kPa/m) (note: for $e/D \leq 0.5$, q is set to zero; for $e/D > 0.5$, the pipeline embedment can be treated as $e/D = 0.5$ with an equivalent uniform surcharge pressure $q = (e - 0.5D)\gamma'$, where γ' is the buoyant unit weight of the soil (in kPa/m³)); and N_c , N_q , and N_γ are the bearing capacity factors for the cohesion, for the distributed load, and

for the buoyant weight of soils, respectively. For the clayey seabed under undrained conditions, if neglecting the effects of the pipe geometric curvature ($e/D \rightarrow 0$), the pipe-soil interface adhesion (i.e., the interfacial friction coefficient $\mu = 0$), and the internal friction of the soil (i.e., the angle of internal friction $\varphi = 0$), the slip-line field solution can be degenerated into Prandtl’s solution for conventional strip footings, i.e., $N_c = 2 + \pi$. With increasing pipeline embedment, the value of N_c decreases from $N_c = 2 + \pi$ at $e/D \rightarrow 0$ and finally reaches $N_c = 4.0$ at $e/D = 0.5$. It has been indicated that the geometric curvature effect is unneglectable when evaluating the ultimate bearing capacity of submarine pipelines.

Note that the aforementioned solutions were obtained under the assumption that the seabed soil is described with a perfectly plastic stress-strain relationship. Such approximation is applicable for the soils of low compressibility, which might meanwhile correspond to the general shear mode of failure. However, for a very soft clayey seabed, large settlements of the pipeline may occur under its submerged weight and laying disturbances without general shear failure occurring. In such cases, the limiting criterion for bearing capacity should be the maximum allowable settlement.

Seabed Liquefaction

Seabed liquefaction is the phenomenon that the seabed sediment loses a significant part of or all its shear strength under the action of ocean waves or earthquake. Both residual and oscillatory mechanisms for the wave-induced pore pressure response in the seabed have been identified in flume experiments and field observations. The residual liquefaction of the seabed is due to the pore pressure buildup under undrained or partially drained conditions, which usually takes place in fine sands or silty soils under severe wave loading and always combined with currents in the offshore field.

Besides residual liquefaction, the instantaneous liquefaction (also termed as “momentary liquefaction”) is particularly prone to occurrence in seabed sediments during severe storms, which

is essentially caused by the instantaneous upward seepage within the upper layer of the seabed under wave troughs. In the instantaneously liquefied layer of the seabed, the vertical gradient of excess pore pressure (j_z) should be identical to the buoyant unit weight of the soil (γ'), i.e., the improved criterion for instantaneous liquefaction (Qi and Gao 2018): $j_z \cong \gamma'$. Based on the analytical solution to Biot's consolidation equations for porous seabed response to waves (Yamamoto et al. 1978) and the above criterion for instantaneous liquefaction, the maximum depth of instantaneously liquefied layer (z_L) under wave troughs can be derived (Qi and Gao 2018):

$$z_L = \frac{p_0}{\gamma'} - \frac{1}{\text{Re} [\lambda(1 - \alpha) + \lambda'\alpha]} \quad (2)$$

where p_0 is the amplitude of wave-induced excess pressure at the seabed surface; the operator "Re{" means to take the real part of the complex variable in the brackets; $\lambda(=2\pi/L)$ is the wave number; L is the wavelength; α and λ' are the complex numbers related to the properties of the seabed and waves (see Yamamoto et al. 1978).

If the pipeline is to be buried in a liquefiable seabed, the designed burial depth should be larger than the maximum depth of instantaneously liquefied layer (see Eq. 2). To keep the buried pipeline vertically stable, the residual liquefaction of the soil around the pipeline needs to be further evaluated. Moreover, under the condition that waves and current coexisting in the field, the seabed liquefaction and the on-bottom stability of the pipeline should be accurately predicted.

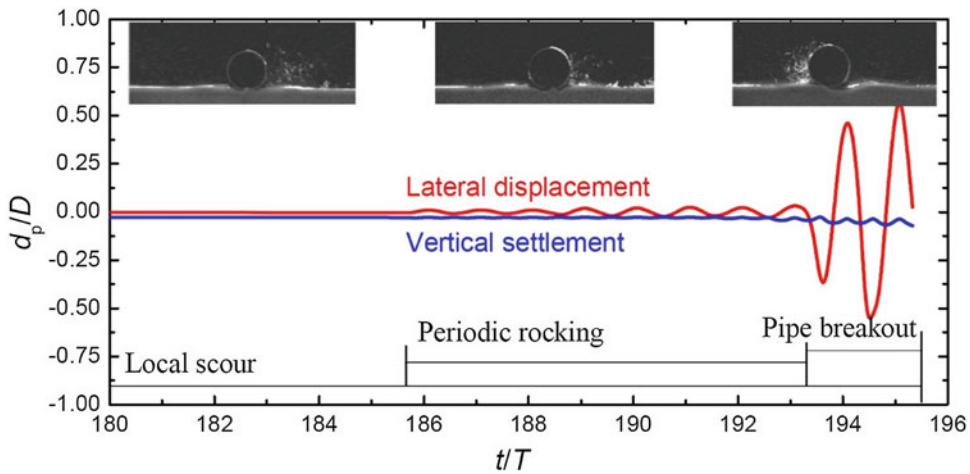
Lateral Stability

In submarine geological and hydrodynamic environments, the multi-mechanics processes can emerge in the proximity of as-laid pipelines, including shear flow above the seabed, sediment transport along the seabed surface, the excessive pore pressure in the soil, etc. They are generally coupled with each other and have significant influence on the lateral stability of submarine pipelines (Gao 2017).

The triggering mechanisms for the pipeline lateral instability involve not only pipe-soil interactions (Wagner et al. 1989; Zhang et al. 2002; Youssef et al. 2013; Gao et al. 2016) but also flow-pipe-soil coupling process (Gao 2017). Experimental observations (see Fig. 1) with an oscillatory flow tunnel showed that there always exist three characteristic stages during the lateral instability of a pipe shallowly embedded in sands under a storm-like wave loading, i.e., (a) local scour of sands around the pipe, (b) periodic rocking of the pipe with small amplitudes, and (c) pipe breakout from original location (see Gao 2017). In the process of the pipe losing lateral instability, local scour always emerges as an indicator for the flow-pipe-soil coupling effect, which was observed taking place at the rear and front of the pipe. The pipe periodic rocking due to the vortex shedding may increase its penetration into the seabed and further affect the lateral stability. The occurrence of pipeline instability is characterized by a distinct lateral displacement (e.g., $d_p/D > 0.5$, where d_p is the pipe lateral displacement, see Fig. 1).

The criteria for pipeline lateral instability are crucial to the on-bottom stability design. Based on similarity analyses for physical modeling experiments, it was found that the controlling non-dimensional parameter of hydrodynamics for pipeline lateral instability is the Froude number ($Fr = U_m/\sqrt{gD}$) and another parameter is the non-dimensional submerged weight of pipelines ($G = W_s/\gamma'D^2$, where U_m is the maximum velocity of wave-induced water particle movement and g is the gravitational acceleration. Both the scaling of the Froude number and that of the Keulegan-Carpenter number ($KC = U_mT/D$, where T is the wave period) can be concurrently satisfied in the physical modeling with wave flumes. KC number essentially controls the generation and development of vortex around the pipeline under oscillatory flow in waves. A unified formulation of criteria for pipeline lateral instability in waves and currents can be expressed as (Gao et al. 2003)

$$Fr_{cr} = a + b \frac{W_s}{\gamma'D^2} \quad (3)$$



On-Bottom Stability of Submarine Pipelines, Fig. 1 Characteristic stages during the pipeline losing lateral stability under a storm-like wave loading (Adapted from Gao 2017)

where Fr_{cr} is the critical values of Froude number for pipeline lateral instability. In Eq. (3), the two parameters a and b are relative to the hydrodynamic loads (periodic waves or steady current), the end constraint conditions (anti-rolling or freely laid) of a shallowly embedded pipe, and the soil properties of the seabed. On the basis of the results of a series of experiments, the values of these two parameters were determined as $(a, b) = (0.07, 0.62)$ for the anti-rolling pipes in waves and $(a, b) = (0.042, 0.38)$ for the freely laid pipes in waves ($5 < KC < 20$). But for the freely laid pipes in a steady current ($KC \rightarrow \infty$), $(a, b) = (0.102, 0.423)$, indicating the pipes are more stable in currents than in waves due to the inertia effect of wave movements. It should be noted that the above recommended values for the two parameters a and b were based on the results of model tests on a uniform medium sand-bed. The particle size is closely related to sediment transport and further affects the lateral stability of the pipeline.

The instability criteria in the unified formulation by Eq. (3) provided alternative expressions to the pipe-soil interaction model by Wagner et al. (1989) for the on-bottom stability of a shallowly embedded pipeline, as addressed by Fredsøe (2016).

Besides the aforementioned lateral instability, submarine pipelines could also be under the threat

from tunnel erosion due to the seepage failure of its underneath soil, especially under the action of shear flow near the seabed, which may further bring the spanned pipeline undergo vibrations, i.e., vortex-induced vibrations (Gao 2017). It has been revealed that the two typical scenarios, i.e., the lateral instability of the pipe and the tunnel erosion of the soil, are always competitive between each other in the submarine geological and hydrodynamic environments (Shi and Gao 2018).

Axial Pipe-Soil Interactions

As offshore developments extend into deep waters, the relatively high pressure and high temperature (HPHT) becomes a dominant factor for the pipeline safety. The HPHT pipeline would undergo expansion and contraction during start-up and shutdown cycles in the operating life, which may induce pipeline walking on the seabed. It should be noted that the pipeline walking is not a limit state; nevertheless, the excessive compressive force can lead to severe global buckling of the pipeline and even the failure of associated infrastructures (Randolph and Gourvenec 2011).

The ultimate axial soil resistance (F_{Ru}) to a conventional strip footing with flat bottom is generally linked with the normal pipe-soil contact

force (F_N) and the interface friction coefficient (μ), i.e., $F_{Ru} = \mu F_N$. But for submarine pipelines, the assessment of ultimate axial soil resistance becomes much more difficult than that for conventional strip footings. Due to the effect of pipeline curvature, the integrated normal pipe-soil contact force (F_N) would exceed the submerged pipeline weight (W_S), i.e., $F_N = \zeta W_S$, by incorporating a wedging factor (ζ):

$$\zeta = \frac{2 \sin \theta_0}{\theta_0 + \sin \theta_0 \cos \theta_0} \quad (4a)$$

for clayey soils (White and Randolph 2007) and

$$\zeta = \frac{1 - (2\theta_0/\pi)^2}{\cos \theta_0} \quad (4b)$$

for sandy soils (Shi et al. 2019), respectively. In Eqs. (4a) and (4b), θ_0 is the semi-angle subtended at the pipe-soil contact chord.

The axial soil resistance expressed with an equivalent friction coefficient could span an order of magnitude, which is related to many influential factors, including the soil types, drainage condition, and the pipe roughness. A theoretical framework was developed within a critical-state context using effective stresses, applicable to any degree of drainage in the soil, quantifying the magnitude and duration of excess pore pressures generated near the pipe-soil interface (Randolph et al. 2012). It has been recognized that the axial soil resistance is crucial for assessing the effective axial force along the pipeline and the corresponding global buckling predictions.

Cross-References

- Design of Pipelines and Risers
- Pipeline Soil Interactions

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OPC – OLE for Process Control

- Central Monitoring System

Open Water Experiment

► Open Water Test

Open Water Test

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Synonyms

Cavitation test; Open water experiment; Open-water test

Definition

The open water test is a propeller model running alone in uniform current, which can be carried out in the model laboratory, circulating water channel and cavitation tunnel. The open water test can estimate the propeller operating performance as a simple and reliable way to determine and analyze propeller performance. Torque, thrust, speed, and rotate speed are tested at the same time and a series of characteristic curves are obtained through the propeller model open water test and provided information for the later research. Propeller Series Diagram is compiled based on the open water characteristic curves obtained from a large number of experimental results in the tank, used for propeller design, calculation and other applications. The B type propeller in the model pool of the Dutch ship is the most famous propeller pattern, while the AU type propeller pattern in Japan is the most frequently used as well.

The propeller model test is a convenient method to examine and analyze the performance of propellers, which is significant in studying the

hydrodynamic performance of propeller. It can provide sufficient information for propeller designs and a reliable basis for the development of propeller theory.

There are several purposes of open water test:

1. With a set of experiments, drawing the results into a particular graph for design.
2. According to the results, analyzing the influence of various geometry elements on the propeller.
3. As an indispensable means of checking and verifying theoretical methods.
4. Comparing the advantages and disadvantages of various design schemes and choosing the prime by cooperating with self-propelled experiment.

Scientific Fundamentals

Historical Development

As the most widely used Marine propeller, the performance of the propeller research mainly concentrated in the pool experiments. As far as the propeller itself is concerned, its operating efficiency in a uniform flow is the most basic important performance index, and the open water test is the most commonly used effective tool of studying this kind of performance. To date, the open water test of single propeller has been studied for hundreds of years, while the open water test of combined propeller such as ducted propeller and counter-rotating propeller has been conducted for decades.

Basic Classification

Model tests carried out in the laboratory.

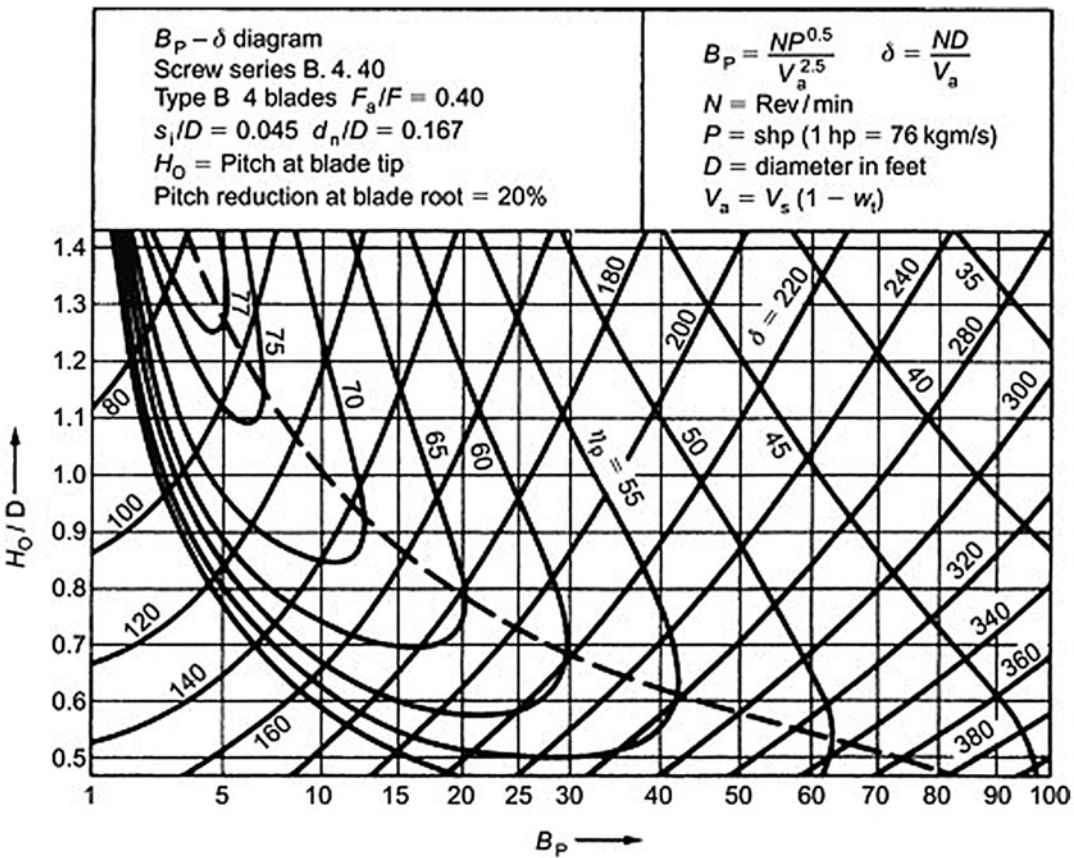
The open water test can be carried out on the propeller, rudder, etc., by the model test in the experimental pool, and the measured data can be sorted out and analyzed to obtain a series of characteristic curves for subsequent calculation.

Open water tests of propellers are used in conjunction with tests behind models to determine the wake and relative torque efficiency (Tupper 2013). Also, methodical propeller testing is

carried out in a towing tank. The propeller is powered from the carriage through a streamlined housing. It is pushed along the tank with the propeller ahead of the housing so that the propeller is effectively in undisturbed water (Tupper 2013). Records of thrust and torque are taken for a range of carriage speeds and propeller revolutions (advance coefficient) (Tupper 2013). Such tests eliminate cavitation and provide data on propeller in uniform flow. This methodical series data can be used by the designer, making allowance for the actual flow conditions. (Tupper 2013). There have been many methodical series. Those by Froude, Taylor, Gawn, Troost, and van Lammeren are worthy of mention. The reader should refer to published data, if it is wished to make use of these series. A typical plot for a four-bladed propeller from Troost's series is presented in Fig. 1 (Tupper 2013).

Numerical Calculation

First of all, the numerical algorithm of the open water performance of the propeller with regard to the numerical calculation of the open water performance has been developed very early. As early as the early twentieth century, Lanchester and others began to study the propeller theory and further study was completed by Prandtl. Then Goldstein put forward the Goldstein theory and the Goldstein factor in 1929. Theoretically, the flow field can be simulated by solving the N-S equation and related equations, so as to solve the problems of propeller correlation performance. However, the process of solving the N-S equation becomes very complicated due to some assumptions made in the process of establishing the N-S equation, as well as the complexity of the practical problems and boundary conditions, and the huge amount of calculation seriously restricts the



Open Water Test, Fig. 1 A typical plot for a four-bladed propeller from Troost's series

development of this method. After nearly a hundred years of research by related scholars, three classical theoretical methods, namely lift line theory, lift surface theory and surface element method, were formed on the basis of potential flow theory, which greatly promoted the study of propeller hydrodynamic performance.

At present, many scholars consider the performance of propeller by building solid model. Solid modeling means that hull, propeller, and other appendages no longer adopt the equivalent alternative method, but build solid model and calculation domain together, in which each model is either static or moving. In the solid modeling calculation, the paddle processing can be divided into three types: multiple reference frame (MRF) method, mixed surface method, and slip surface method.

MRF method is to divide the whole computational region into multiple subdomains. Individual subdomains have their own forms of motion such as stationary, moving and rotating, and these subdomains can be considered to maintain the steady state approximation at different moving or rotating speeds. By solving the governing equations of each subdomain and exchanging the flow information of each subdomain on the general interface, the flow information of the whole flow field computing domain can be obtained. If the meshes on both sides of the subdomain interface are not completely consistent, the flow information on the interface is transmitted by interpolation. MRF method is suitable for many conditions of steady flow, especially when the flow area is almost uniform at the boundary.

The hybrid plane method also divides the whole computing domain into several independent subdomains. Instead of each of these little domains being seen as a steady state approximation at different velocities of motion or rotation, each little domain is a steady state. It is considered as a steady flow in the subdomain, and a mixed surface similar to boundary conditions is constructed on the interface between the subdomains to transfer the flow information of both subdomains. The mixed plane method can solve the unstable problems caused by the circumferential changes among the subdomains, such as the

steady-state and flow separation, and then obtain a stable solution.

When using the slip surface method for calculation, there are multiple subdomains in the entire flow region, and there are relative motions between these subdomains, as well as interfaces between every two of them. During the calculation process, the part of the interface that is connected with the adjacent subdomain is called the internal region, and the other part that is not connected with the adjacent subdomain is called the plane region as solving the plane problem. And the part in dealing with periodic or aperiodic problems is called the periodic region. In addition, while using the sliding grid technology, the grids on the interface are irregular, namely the grids on both sides of the interface are not completely consistent. In other words, the relative movement between two adjacent subdomains should be recalculated to obtain the boundary position of the internal region in the specific calculation. This kind of calculation is complex, requiring a high performance of the computer. Compared with the MRF method and the mixed surface method, the slip surface method has a certain advantage in solving periodic or aperiodic problems.

At present, sliding grid technology is often used to transfer flow field information. In the meantime, the calculation methods include lift line method based on potential flow theory, lift surface method, surface element method, and viscous flow method based on RANS equation. Four methods are described below.

The lift line method proposed by Goldstein in 1929 is applied to propeller performance prediction. In this method, all the attached vortices of the propeller are concentrated on a spreading line from the blade root to blade tip, which is called the lift line of the propeller. The strength of the lift line changes along the spanwise direction, and the tail vortex surface is formed downstream of the propeller by the change of the strength of the attached vortex. With the circulation distribution, it is possible to obtain the lift force of each micro-segment vortex element and then obtain the hydrodynamic performance parameters of the propeller accordingly.

The lift surface method is based on the thin airfoil characteristics of Marine propellers. On the one hand, the thickness part only contributes to the pressure distribution of the blade, which is expressed by the source-sink distribution on the chord surface of the wing. On the other hand, the bending part is the origin of lift, which is represented by the vortex distributed on the chord surface of the wing. The boundary conditions are satisfied on the wing chord. Because the boundary conditions of lift surface method are stricter than the lift line method, it can result in more accurate results.

The surface element method is a kind of boundary element method, and its basic idea is to transform the Laplace equation of the incompressible, inviscid, and irrotational potential flow problem into an integral equation on the boundary by using Green formula. Thus, the flow around a three-dimensional object is transformed into the solution of the unknown singular point plane distribution on any object surface.

The viscous flow method using RANS equation is to directly solve the RANS equation method to obtain the propeller viscous flow field, so as to obtain the propeller hydrodynamic performance parameters.

FINE/Marine, Fluent, etc., are commonly available software.

Principles of Open Water Test

The advance speed and rotation speed of the propeller model are obtained according to the similarity theorem of the propeller including geometric similarity, kinematic similarity, and dynamic similarity. The advance coefficient, Reynolds number, and Froude number are presented with dimension analysis method.

$$J = \frac{V_A}{nD}$$

$$R_n = \frac{V_A L}{\nu}$$

$$F_r = \frac{V_A}{\sqrt{gL}}$$

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

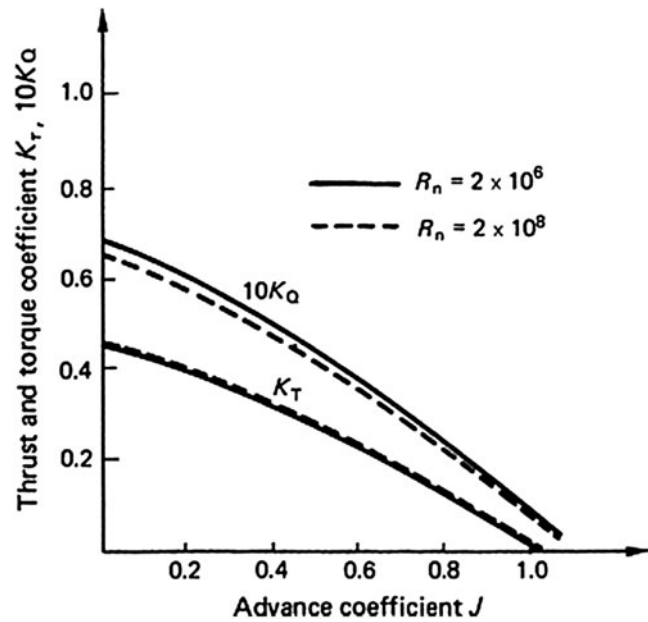
$$\eta_0 = \frac{K_T}{K_Q} \cdot \frac{J}{2\pi}$$

J represents the advance coefficient. K_T is the symbol of thrust coefficient and K_Q is the symbol of torque coefficient. η_0 represents the open water efficiency.

The similarity theorem is satisfied when the three parts of the propeller and the model are equal. Furthermore, the effect of the wave is negligible when the paddle shaft is at a certain depth underwater, and the Froude number representing gravity similarity is abandoned. In addition, when the critical Reynolds number (3×10^5) is met, the influence of viscosity can be ignored. However, the scale effect needs to be considered.

Open water characteristics are frequently determined from model experiments on propellers run at high-speed and having diameters of the order of 200–300 mm (Carlton 2012). It is, therefore, reasonable to pose the question of how the reduction in propeller speed and increase in diameter at full-scale will affect the propeller performance characteristics (Carlton 2012). Figure 2 shows the principal features of scale effect, from which it can be seen that the torque coefficient is somewhat reduced for a given advance coefficient while the thrust characteristic is largely unaffected (Carlton 2012).

The scale effects affecting performance characteristics are essentially viscous in nature and are primarily due to boundary layer phenomena dependent on Reynolds number (Carlton 2012). Due to the methods of testing model propellers and the consequent changes in Reynolds number between model and full-scale, or indeed a smaller model and a larger model, there can arise a different boundary layer structure to the flow over the blades (Carlton 2012). While it is generally recognized that most full-scale propellers will have a primarily turbulent flow over the blade surface, this need not be the case for the model where

Open Water Test,**Fig. 2** The principal features of scale effect

characteristics related to laminar flow can prevail over significant parts of the blade (Carlton 2012).

In order to quantify the scale effect on the performance characteristics of a propeller, an analytical procedure is required (Carlton 2012). There is, however, no universal agreement as to which is the best procedure. In a survey conducted by the 1987 ITTC, it was shown that from a sample of 22 organizations, 41 percent used the ITTC–1978 procedure; 23% made corrections based on correlation factors developed from experience; 13%, who dealt with vessels having open shafts and struts, made no correction at all; a further 13% endeavored to scale each propulsion coefficient, while the final 10% scaled the open water test data and then used the estimated full-scale advance coefficient (Carlton 2012).

Hence, similarity theorem is all about kinematic similarity, implemented through equalizing the advance coefficient of the propeller and model.

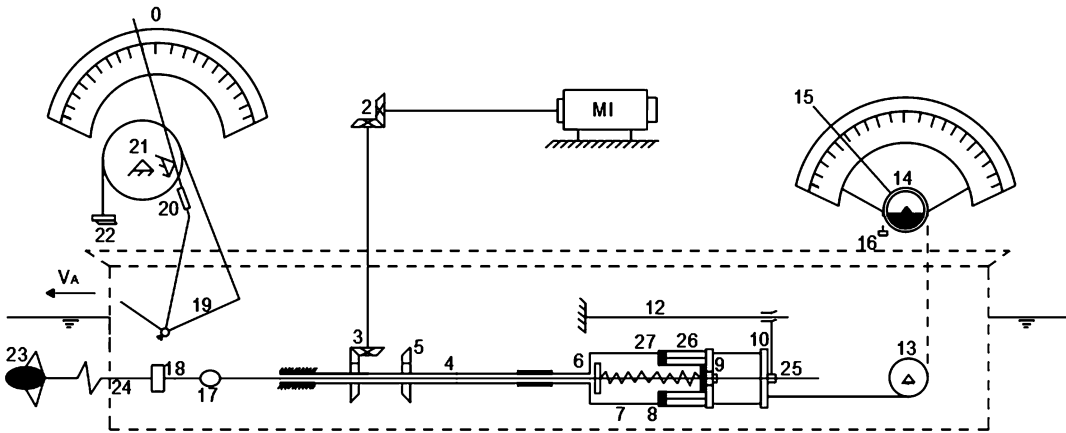
$$K_T = f_1(J)$$

$$K_Q = f_2(J)$$

There are two ways to obtain the advance coefficient: In the towing tank, the model rotational

speed is kept constant and the advance speed is changed; in the cavitation tunnel, the advance speed of the model remains still and the rotational speed is changed. The thrust coefficient, torque coefficient, and open water efficiency under different advance coefficients were obtained through experiments and plotted as the propeller characteristic curves (Zhu 2016).

The main measuring device for the open water test is data acquisition system and propeller dynamometer, which is classified as a mechanical dynamometer (shown in Fig. 3), electric test dynamometer, and electromechanical integrated dynamometer due to the different power sources. The data acquisition system consists of two parts. One part is a data collection device, which measures data by an open water dynamometer machine and transmutes to the data collection device for filtering, amplifying, and processing. The other part is data acquisition system supporting software. The data acquisition device is connected to a computer with software, which will be processed in the back of the data input, and the software will be able to visualize and move on to the data. At the same time, the software is also a platform for the operation of data acquisition equipment, with the control of data acquisition, system balance, and other functions.



Open Water Test, Fig. 3 J04 Propeller Dynamometer (mechanical dynamometer)

In the preparation stage of the test, the whole set of test equipment including open water dynamometer and data acquisition system is tested and calibrated to obtain the calibrated value. Besides, in the open water test, the rotational speed and advance speed of the propeller should be determined first. The selection of these two values is very important. First of all, the rotation speed of the propeller should take into account the influence of factors such as the material of the propeller itself, and the selection of the advance speed should take into account the influence of factors such as the performance of the trailer. The interval of the advance speed shall not be too large. Otherwise, the test data points will be too small, which will not be able to accurately reflect the trend of the water performance curve of the open water. On the contrary, the interval will be too small to increase the number of tests. Then the experiment can be carried out with the change of advance speed.

During the test in the cavitation test tube, the similarity condition must be satisfied that the number of cavitation is equal as well as the coefficient of advance. The cavitation test cylinder is composed of a sealed circulating water cylinder, a driving water pump, a pressure regulating device, and so on. There is an observation window in the working section, and throughout the test, water circulates and is depressed to make the number of cavitation equal to the number of the real

propellers. The test process is observed and recorded with stroboscope and high-speed camera, and the thrust, torque, and other test data of propeller are measured by dynamometer.

Propeller tests (open-water tests, cavitation tests) are usually performed in cavitation tunnels (Bertram 2012). A cavitation tunnel is a closed channel in the vertical plane recirculating water by means of an impeller in the lower horizontal part (Bertram 2012). In this way, the high hydrostatic pressure ensures that even for reduced pressure in the tunnel, the impeller itself will not cavitate (Bertram 2012). The actual test section is in the upper horizontal part. The test section is provided with observation glass ports (Bertram 2012). The tunnels are designed to give (almost) uniform flow as inflow to the test section. If just the propeller is tested (with the driving shaft downstream), it is adequately tested in open water (Bertram 2012).

Vacuum pumps reduce the pressure in the tunnel and usually some devices are installed to reduce the amount of dissolved air and gas in the water. Wire screens may be installed to generate the desired amount of turbulence (Bertram 2012).

Cavitation tunnels are equipped with stroboscopic lights that illuminate the propeller intermittently such that propeller blades are always seen at the same position (Bertram 2012). The eye then perceives the propeller and cavitation patterns on each blade as stationary (Bertram 2012).

Usual cavitation tunnels have too much background noise to observe or measure the noise-making or hydro-acoustic properties of a propeller, which are of great interest for certain propellers, especially for submarines or anti-submarine combatants (Bertram 2012). Several dedicated hydro-acoustic tunnels have been built worldwide to allow acoustical measurements. The HYKAT (hydroacoustic cavitation tunnel) of HSWA is one of these (Bertram 2012).

Key Applications

The open water test is mainly used to measure the hydrodynamic performance of various propellers, such as ducted propellers, pod propellers, and combined propellers. It is also sometimes used to measure hydrodynamic characteristics of rudder and other attachments, such as the open-water test of a connecting rod flap rudder (Zhu 2013).

Open Water Test of Pod Propellers

Ever since the concept of pod propeller was proposed, after more than 20 years of development, it has been successfully applied in many ship types. Its excellent performance, which has been shown in practical applications, attracts more and more ship-owner and ship designers' concerns, as well as more and more attention to their research on hydrodynamic performance. The open water test for the pod thrusters is to test the entire pod parts, including the propeller, the pod, the stents, and the fins, as part of the propulsion unit, and the open water test for the pod unit requires the development of a specialized testing device. The complete open water test of pod propeller not only measures the thrust and torque of a propeller, but also measures the thrust of the whole pod unit (Shen 2007). The characteristic quantities such as the lateral force and the rudder moment of the whole unit can be investigated more completely. The dynamometer includes propeller dynamometer and unit force balance. The thrust and torque of the propeller are measured by a dynamometer located in the cabin. The axial force, lateral force, and rudder torque of the whole unit are measured by a three-component force balance. Besides, the

propeller is driven by a motor through a driving shaft. Water tightness is considered in the design of the dynamometer and transmission mechanism as well. The whole test device is composed of free stress part and fixed constraint part. The free force part consists of propeller, cabin, bracket, tail fin, and other parts. The force is measured by propeller dynamometer and three-component balance. It is important that free-stressed parts are not affected by other test equipment. The fixed restraint is secured to the trailer and should be avoided that the force on it affects the balance measurement. The test apparatus is capable of accurately adjusting the submerged depth of the propeller shaft in the subsea and ensuring that the paddle axis is parallel to the horizontal plane. Combined with the test results of the pool and other relevant data analysis, it is concluded that in the open water test of the pod propeller, the bracket clearance should be kept as small as possible, and the hub clearance should be controlled in an appropriate range.

Open Water Test of Combined Propellers

Compared with common propellers, combined propellers have the great advantage of reducing the radiation noise of propellers, improving the efficiency of open water and the initial critical speed of cavitation. At the same time, if the propeller can match the hull properly, it can even obtain higher propulsion efficiency. The structural features of such thrusters are multicomponent combinations, so that the characteristics of hydrodynamic interference among components will be the focus of their performance research. In order to study the hydrodynamic interaction between the components of the combined propeller, the method of decomposition test may be employed.

New Scientific Discovery

Changes in Testing Methods

By using the traditional test method, on the one hand, it needs to be operated by many people, and it is easy to bring artificial reading error, and on the other hand, processing the data of these measurements is quite a time-consuming and

laborious work. With the rapid development of computer technology, a variety of electrical measuring sensors have appeared continuously, which provides convenient conditions for the computerization of open water test. By using the computer to collect and process the data, we can analyze and modify the program in real-time, reduce the error, and realize the automatic control of the experiment process.

Cross-References

- [Numerical Simulation Floe Ice–Sloping Structure Interactions](#)
- [Resistance Test](#)

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Open-Water Test

- [Open Water Test](#)

Operational Challenges

- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

Optical

- [Fiber-Optic Cable](#)

Optical Compass

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Synonyms

Digital signal processor (DSP); Fiber-optic gyro (FOG); Fiber-optic gyro attitude reference system; Fiber-optic gyro motion sensor; Inertial measurement unit (IMU); Motion reference unit (MRU); Strapdown inertial navigation system (SINS)

Definition

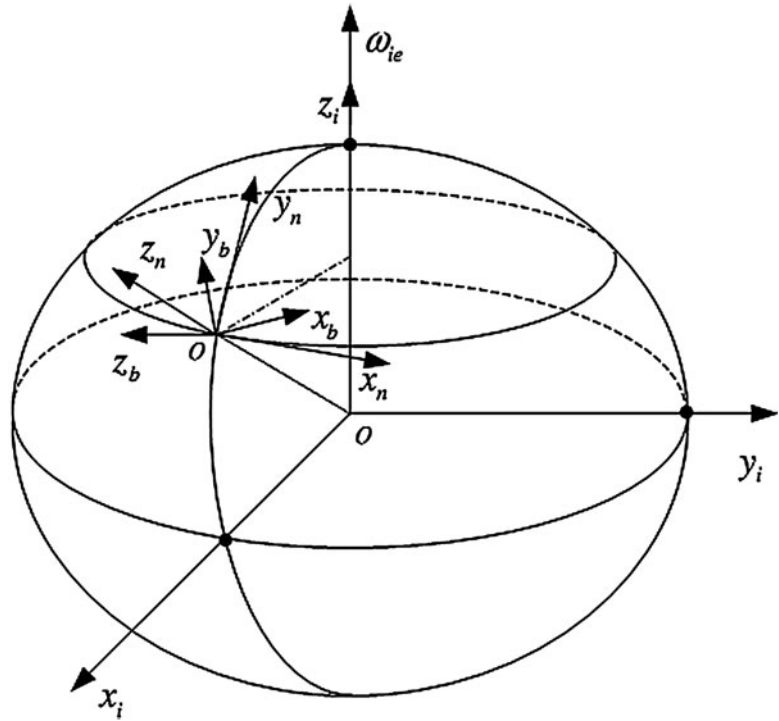
Fiber-optic gyro compass is a strapdown inertial navigation system (SINS) with three fiber-optic gyros and three accelerometers. This compass outputs the information of the heading, roll, and pitch. Comparing to the traditional mechanical gyro compass, fiber-optic gyro compass has less setting time, higher measuring accuracy, better dynamic characteristics, and longer service life. It is specially suitable for the navigation and control of high-speed platforms with high rate of turn.

Scientific Fundamentals

Frame Definitions

A fiber-optic gyro compass relates to several basic navigation frames (Li et al. 2013) drawn in Fig. 1.

1. The b-frame. Body frame, the strapdown inertial sensor frame, with its origin at the center of mass of the accelerometer and gyro triad and

Optical Compass,**Fig. 1** The frames definition

axes parallel to nominal right-handed orthogonal sensors input axes.

2. The i-frame. Inertial frame, stabilized in the inertial space.
3. The n -frame. Navigation frame is the local level coordinate frame here. Its x axis points to East (E), y axis points to North (N), and z axis points upward (U).

Fiber-Optic Gyro

In a fiber-optic gyro compass, fiber-optic gyro (FOG) is used for measuring the rotation angle. Recently, FOG is being widely used for military and defense applications, due to its significant advantages such as small size, low cost, light weight, no moving parts, large dynamic range, low power consumption, and possible batch fabrication (Narasimhappa et al. 2014).

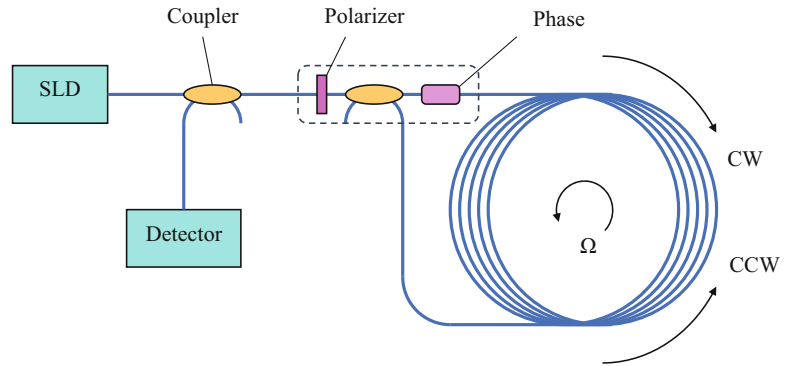
FOG is a kind of optical gyros and its principle is based on the Sagnac effect, as shown in Fig. 2. After passing through the coupler, the light beam from SLD source is divided into two light beams by the polarizer. The two light beams propagate along the two opposite optical paths in the fiber-

optic ring. If there is no angular velocity ($\Omega = 0$), the CW light beam and CCW light beam arrive the detector simultaneously and interfere with each other. The detector could convert the light intensity of interference into electrical signals. If there is one angular velocity ($\Omega \neq 0$), the output of the detector will be changed. The polarizer could also modulate the phase of the light beam by applying voltages to improve the angular velocity detection.

The performance of FOG degrades due to the variation in environmental factors such as temperature, vibration, and pressure. Among different types of error in the FOG signal, random drift error leads to decrease the FOG performance over a period of time. The precision of FOG depends on the bias drift and noise in the measurement. FOG has mainly two types of error (i) deterministic error and (ii) stochastic error. Deterministic errors are due to the scale factor, bias, and misalignment which can be eliminated by suitable calibration techniques in the laboratory environment. However, stochastic errors are due to the environmental temperature changes, electronic components, and other electronic equipment interfaced with it. It is difficult to

Optical Compass,

Fig. 2 Basic structure of the fiber-optic gyro



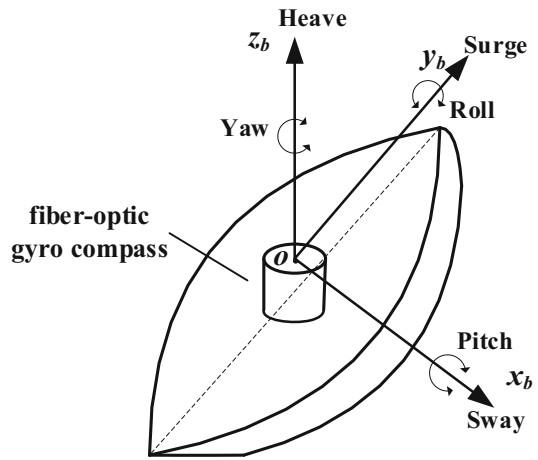
eliminate these errors by calibration. Thus, stochastic models are required to characterize these errors, and signal processing techniques are required to suppress these errors.

Working Principle

In favor of modern technologies, gyrocompass manufacturers have abandoned gimballed platform and mechanical gyros, and began to use mathematic platform and optical gyros instead. Comparing to the traditional mechanical gyro compass, fiber-optic gyro compass has less setting time, higher measuring accuracy, and better dynamic characteristics. It is specially suitable for the navigation and control of high-speed platforms. Because of the solid-state solution, the strapdown gyro compass has high reliability and is maintenance-free during its service life. Thus, strapdown gyro compass is the current and next-generation marine navigator to provide attitude measurements.

The heart of a fiber-optic gyro compass is its inertial measurement unit (IMU), whose mechanism is composed of three orthogonal fiber-optic gyros and three orthogonal accelerometers. As depicted in Fig. 3, a full range IMU corresponding to the three axes of the ship b -frame system will measure six degrees of freedom movement relative to n -frame system including the three rotational moments (pitch, roll, and yaw) and the three translational moments (heave, surge, and sway).

The basic working principle of a fiber-optic gyro compass is the strapdown inertial navigation algorithm. A block diagram representation of this algorithm is shown in Fig. 4. Strapdown inertial



Optical Compass, Fig. 3 Fiber-optic gyro compass installed in ship

navigation system is a self-contained architecture that double integrates three acceleration components with respect to time and transforms them into the n -frame to deliver position, velocity, and attitude components in real time. The three orthogonal linear accelerations are continuously measured through three-axis accelerometers while three gyroscopes sensors measure the three orthogonal angular rates in an inertial reference frame. Digital signal processor (DSP) is used to calculate the mathematic platform.

For short time, the integration with respect to time of the linear acceleration and angular velocity monitored by the gyroscopes results in accurate attitude, position, and velocity. However, sensor errors and mathematical integration over a long time of the fiber-optic gyro compass may

The relationship between the navigation frame and the body frame is realized by continuously updating this attitude matrix. To limit the errors in the derived navigation parameters, it is very important to determine the initial value of such matrix with high accuracy.

In static ground alignment, the system attitude can be determined directly by using the gravity and earth rate signals obtained by using accelerometers and gyros in the navigation frame. However, the swing of marine will be caused by the wind wave. During the alignment process of SINS, gyros are employed to monitor the components of the earth rotation rate along their sensitive axes to determine the initial attitude of the moving platform. The angular rate error which is decided by the wind wave is much larger than Earth rate. So, the signal-to-noise ratio of gyro's output is poor, and the frequency band of angular rate error is wide. It is impossible to directly get the Earth rate from gyros. Therefore, a feasible alignment solution for marine SINS may not only use the Earth rate and gravity obtained by using gyros and accelerometers, but also use some external information (GPS, LOG, etc.) (Sun and Sun 2010).

The alignment process of compass is generally divided into two steps: coarse alignment and fine alignment. The goal of the coarse alignment stage is to solve the rough coordinate transformation matrix from the carrier system to the navigation system as soon as possible, which provides a good initial condition for the fine alignment stage. Therefore, rapidity is the main performance design index in the coarse alignment process. Generally, there are some errors between the attitude matrix obtained by coarse alignment and the real navigation system, especially the error of heading angle, which is much bigger than the error of horizontal attitude. In addition, the errors of inertial devices, such as the constant drift of gyroscope, will lead to the deterioration of the performance of navigation equipment after the start-up of the equipment. It is usually necessary to estimate and compensate for these errors in the process of fine alignment. The aim of fine alignment is to obtain the attitude angle which is infinitely close to the real value and compensate for

other errors in the alignment process. Accuracy and rapidity are the two most important indicators of fine alignment.

Damping Technology of Strapdown Compass

In strapdown solution, in order to avoid the influence of acceleration information on accelerometer data, it is necessary to select appropriate system parameters to make the system meet Schuler tuning conditions which are the undamped working state. However, when the system works in the undamped state, there are three kinds of oscillation errors caused by various kinds of disturbances. The three kinds are Schuler oscillation error with a period of 84.4 min, Foucault oscillation error with latitude variation, and Earth oscillation error with a period of 24 h. Among those oscillation errors, Foucault oscillation error and Schuler periodic oscillation error are mutually modulated. Those oscillation errors will lead to oscillation errors of navigation parameters in pure inertia solution, and even divergence of navigation errors in long-term operation. Those oscillation errors cannot be ignored. Therefore, in practical applications, the strapdown compass is usually switched between undamped and damped states according to different sea conditions.

There are two main working states of a fiber-optic gyro compass: undamped state and damped state. The fiber-optic gyro compass normally works in the damped state which is also called compass state. The compass-loop control is used to find the north and indicate the ship's course and attitude information. When the ship encounters strong motions such as bad sea conditions or frequent velocity variations, the disturbance motion information is measured by accelerometer, and then controlled by compass channel as part of gravity. In order to reduce the impact of external maneuvering environment on the compass output, the fiber-optic gyro compass needs to enter the undamped state in a short time.

A general way to compensate the maneuvering error caused by the motion of the carrier is the "broken pendulum" often used in the platform compass system. This way refers to the accelerometer measurement data. If the sensitive data exceeds one threshold, the fiber-optic gyro

compass will not calculate the attitude information based on the accelerometer data. Therefore, the interference of errors caused by the acceleration of carrier maneuver can be overcome by an appropriate threshold.

Integrated Navigation Technology

With the rapid development of the fiber-optic gyro, the fiber-optic gyro compass will be more accurate, smaller in size, more reliable, cheaper, and lower in energy consumption. However, the navigation error of the fiber-optic gyro compass will accumulate over time, which is difficult to meet the long-distance and long-time navigation requirements. Therefore, the external navigation information from other navigation systems is necessary for the fiber-optic gyro compass to correct the diffuse system error.

The common working principle of an integrated navigation system is shown in Fig. 5. Each navigation system provides corresponding navigation information, which is fused by filtering method to obtain more accurate and reliable navigation information. According to different application

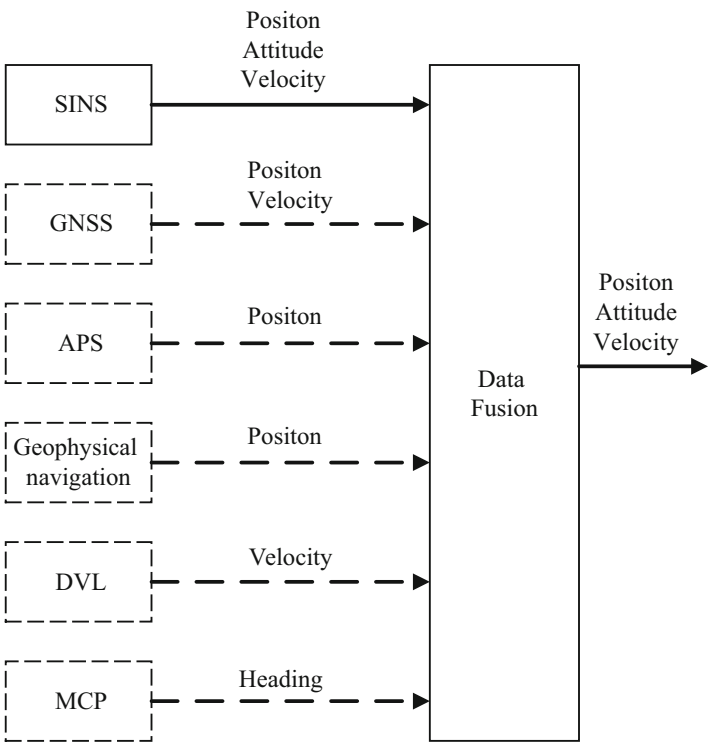
environment and requirement, one or more other navigation modules are chosen for an integrated navigation. The general process of building integrated navigation system can be described as follows: by modeling and analyzing the system structure abstractly, the state equation and measurement equation of the system are established, and the state information or state error information of the system is estimated optimally by Kalman filter, and the optimal estimation is used to correct the navigation parameter information of subsystems.

Typical Applications

Navigation

At sea, mechanical gyro compass has always been used as a source of course information for ships. The mechanical gyros can sense the angular deviation of the navigation reference coordinate system relative to the inertial coordinate system, and provide this signal to navigation and positioning systems. It can be applied to the satellite positioner ARPA, integrated navigation system,

Optical Compass,
Fig. 5 The principle of integrated navigation system



AIS, etc. The improvement of the mechanical gyro accuracy is seriously restricted by the unbalanced mass of the high-speed rotating “rotor,” the cross-coupling effect of the rotational degrees of freedom, the moment of inertia of the rotor, the harmful moment of the rotor support, and other factors. Also, the start-up time of the mechanical gyro compass is up to 6 h long. The fiber-optic gyro compass is not affected by this aspect. In addition to providing high-precision course information, it can also provide information of pitch, roll, and ship rotation angular velocity. The fiber-optic gyro compass hardly needs start-up time. Therefore, it will replace the mechanical gyro compass and be widely used in navigation.

Survey

With the increasing demand for understanding underwater topography and geomorphology, multi-beam bathymetry system is becoming more and more popular as an underwater topographic survey system with the advantages of full coverage, high efficiency, high precision, and high resolution. It has been widely used in underwater completion survey, pipeline route survey, breakwater mapping, before dredging and after dredging survey, coastal zone mapping, revetment survey, port mapping, river survey, IHO special survey, search and rescue, site obstacle survey, pipeline inspection, underwater inspection, object positioning, etc. When Multi-beam bathymetric system is used for bathymetric survey, due to the influence of wind and wave, the surveying ship will inevitably produce three rotational moments (pitch, roll, and yaw) and the three translational moments (heave, surge, and sway), which will lead to the inclination of the coordinate system of the surveying ship and the systematic deviation of the measured water depth. A fiber-optic gyro compass can provide six-degree-of-freedom motion information for ship measurement, and assist multi-beam bathymetric system to make systematic correction.

Dynamic Positioning

Dynamic positioning ship has become one of the most important marine equipment in the ocean engineering, which consists of measuring,

propulsion, and control devices. The accuracy of measurement system has a great influence on the accuracy of ship positioning control. A fiber-optic gyro compass could be used as a motion reference unit (MRU) to measure the ship’s motion. It measures the roll, pitch, and heave of the ship or platform during operation. Comparing the measured data with the preset values, the control system generates control instructions to keep the ship’s position.

Cross-References

- [Fiber-optic Gyro Attitude Reference System](#)
- [Fiber-optic Gyro Motion Sensor](#)

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Optical Fiber

- [Umbilical Cable](#)

Optical Fibers

- [Fiber-Optic Cable](#)

Optimal Control

- [Intelligent Control Algorithms in Underwater Vehicles](#)

Optimal Design

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Synonyms

Design optimization; Optimized design; Optimum design

Definition

Optimal design is usually considered as the design process that seeks the “best” possible solution(s) for a mechanical structure, device, or system, satisfying the requirements and leading to the “best” performance, through optimization techniques. It also refers to the design points that best satisfy objective(s) which are in contrast to non-optimal design. George Dieter has given a formal definition of **design** as follows: “Design establishes and defines solutions to pertinent structures for problems not solved before, or new solutions to problems which have previously been solved in a different way” (Dieter and Schmidt 2009). **Optimal** indicates a searching and decision-making process that is required to determine the best possible design alternatives. Optimization techniques are used to evaluate the trade-off among design alternatives and determine the best one(s).

Scientific Fundamentals

Brief Background

Engineering design process can be viewed as a sequential process. The sequence of design operations includes exploring alternative concepts, formulating a mathematical model, specifying specific subsystems, etc., while the design of complex systems can be broken down into a sequence of design steps. Design solutions come out as a result of repeated trials and errors. With the help

of computer technologies, time-consuming and repetitive operations, like experiments, are replaced by computer program or simulations. Complex problems can be analyzed faster, and designers’ capabilities are extended.

The importance of optimal design has been realized in 1980s since the high-quality products from Japanese companies invaded the market of the western world. Mathematical fundamentals for the design process have been developed for a long time, and it is still on its way to maturity. The best way to achieve high quality in a product is to design it from the beginning and assure that it is maintained throughout the manufacturing process. The enemy of quality is variability in performance and manufacturing. In this case, *robust design* has arisen to reduce variability while simultaneously leading the performance toward the optimal design point (Dieter and Schmidt 2009).

To guarantee the product quality, the concept of quality control arose. All actions taken throughout the engineering and manufacturing of a product to prevent and detect product deficiencies and product safety hazards can be regarded as **quality control**. According to the “80/20 rule,” it is clever to concentrate on the vital few causes that worsen the product quality. *Cause-and-effect analysis diagram* can be used to identify possible causes of a problem. **Control chart** is an important quality control tool, which collects and keeps performance data for detecting variations.

Since there is more than one possible solution to a design problem, the need for optimization is inherent in a design process. Design optimization methods in engineering design can be summarized by four broad categories: by evolution, by intuition, by trial-and-error modeling, and by numerical algorithms. There is no one universal optimization method for engineering design problems, and different kinds of methods fit for different problems. Computer-aided optimal design (or to be more specific, design optimization) generally starts with an initial or existing design point, and numerical analysis is used afterward to calculate the performance measurements and the sensitivity of the performance measurements with regard to uncertainties in design parameters.

New design candidates will be generated during the running of optimization algorithms, while optimum design(s) will be achieved after the execution of the optimization. It is worth mentioning that although we often use “optimal design” and “design optimization” interchangeably, they are not strictly equivalent. The former one usually refers to the whole design process or the optimal solution itself, while the latter usually refers to one important step in the design process or technology used to achieve the design solution(s).

Principles of Optimal Design

Design System

Designing is a systematic work. A design solution is also called a system (Arora 2004). An original system can be made up of several components which are also systems with their own functions and input/output characteristics, as shown in the diagram in Fig. 1. That is to say, a system can usually be decomposed into interconnected subsystems, which can be further broken down and analyzed at a particular but lower level of complexity. For example, a gas-turbine system contains the components, e.g., compressor, combustor, turbine, etc. Each of the components is a complicated system itself. Meanwhile, the gas-turbine system is just one component of a petroleum plant. It is of vital importance to decide whether to represent the design task as a unit or as a collection of components (subsystems).

Mathematical Model

Engineers always use an abstract representation instead of a real system for the purpose of analysis. An abstract description of the real-world system is called a model, which provides an approximate representation of a physical system

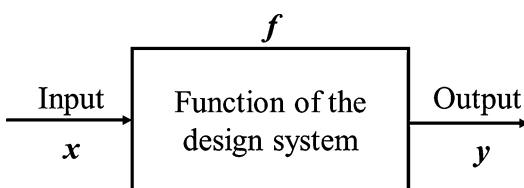
with complex functions. Mathematical models have been widely accepted and applied in representing design systems through mathematical relations. The system, as shown in Fig. 1, can be represented by $y = f(x)$, where x , y represent the vectors of input and output quantities, respectively, and f can be equations, algebraic or differential, or a computer-based simulation program.

To model a system, we need to generalize the design problem and identify the following model elements: system variables, system parameters, system constants, and mathematical relations. *System variables* specify different states of the system by assigning different values. *System parameters* are assigned with one specific value in one particular model statement and can be used to take care of uncertainty or variations when considering robust design. *System constants* are fixed by underlying phenomenon which cannot be influenced by the designers. *Mathematical relations* relate to the system variables, parameters and constants using equalities and inequalities, or even implicit formulations. Determination between variables and parameters is subjective but important at the modeling stage. To model a design mathematically, we must be able to define it completely by assigning values to each quantity involved, such as geometric configuration, the material used, and the task it performs. A design modeling study will help increase our understanding on how a system works.

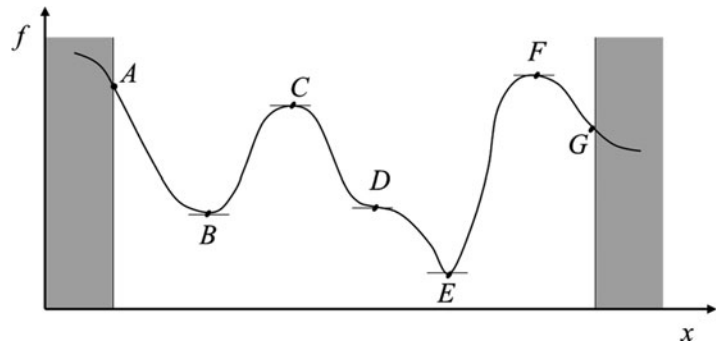
Decision-Making and Design Optimization

A design process can generate more than one alternative in general. A need for making a decision on choosing one or a few of them in this case arises. To evaluate different alternatives and rank them in a rational order, at least one criterion is required. Such criterion may not be unique or may change with time. The design model with the evaluation criterion is a decision-making model or an optimization model. The criterion is called the *objective* of the model, and the selected “best” design is called *optimal design*. Design optimization is to select the “best” design solution (s) according to available requirements.

To quantitatively formulate a design optimization problem and solve it, a mathematical



Optimal Design, Fig. 1 Design system diagram

Optimal Design,**Fig. 2** Local and global optima

representation is built. $\mathbf{x} = (x_1, x_2, \dots, x_n)^T$, in a n -dimensional design space, is the vector of *design variables* whose value describe a design alternative. The acceptable limits on $\mathbf{x}$ are called *lower* and *upper* bounds. The objective is represented by a function of design variable, $f(\mathbf{x})$. It is possible that there are multiple objectives in one design optimization problem too. In that case, there will be multiple objective functions, like $f_1(\mathbf{x}), f_2(\mathbf{x}), \dots, f_m(\mathbf{x})$. Available requirements are *constraints* by the function relationship among $\mathbf{x}$. $h_k(\mathbf{x}) = 0, k = 1, \dots, K$ and $g_j(\mathbf{x}) \leq 0, j = 1, \dots, J$ represent equality and inequality constraints, respectively. Any design satisfying the constraints contributes to the *feasible design space*. Different forms for f, g_j , and h_k are usually derived from basic equations and engineering laws. Also they can be derived through curve fitting using empirical or experimental data or through simulation modeling, which is the current statement of the complex procedure involving internal calculations and computer programs. In practice, mathematical models in design optimization can be a mixture of above. Then the design optimization problem is formulated as

$$\begin{aligned} & \min_{\mathbf{x}} f(\mathbf{x}) \\ & \text{s.t. } g(\mathbf{x}) \leq 0 \\ & \quad h(\mathbf{x}) = 0 \\ & \text{where } \mathbf{x} = [x_1, x_2, \dots, x_n]^T \end{aligned} \quad (1)$$

Design optimization involves the selection of design objectives, determination of constraints, and the set of design variables that optimize the objective(s) while satisfying all constraints.

A complete optimization study to find the optimal design(s) could be expensive and time-consuming for large systems. Thus, several iterations, instead of a complete optimization study, are performed until sufficient improvement in the design has been obtained.

Local and Global Optima

As shown in Fig. 2, point B and E (point C and F) are minima (maxima) with regard to the values in the local vicinity. They are called *local minimum* (*local maximum*). The boundary points A and G are also taken into consideration when determining the overall optimum. From all local minima (local maxima) and boundary points, the smallest one is defined as the *global minimum* (*global maximum*). It can be observed that both local and global optima may not be unique. Sometimes an infinity of solutions may be detected.

Finding global solutions to optimization problems is important for design optimization. However there is yet no universal solution strategy proven by general optimization theory in practice.

To identify the global optimum, a thorough study is conducted to go through all the local optima and determine the global one even under different ranges of parameters. However, this may be difficult from the mathematical viewpoint or expensive from the computational viewpoint.

Mathematical methods can be used to identify the solution(s) to the problem. However, to address the solving task in a purely mathematical way while divorcing from the physical significance of the model and its elements can be a tactical error. Optimization problems in design are generally difficult to solve mathematically.

On the contrary, the “best” design solutions may not need to be mathematically proven optima. Typical solving methods in mathematical programming use iterative numerical search procedures. Starting from an initial point, the direction of improvement is identified, and a step is taken along that direction. Consequently, a sequence of points that converge to the minimum can be created. Such numerical iterative computation is driven by local knowledge, which is easy to get but has limited ranges. Global knowledge is superior and results in better techniques but difficult to acquire.

Optimization Methods

Analytical Optimization Methods *Graphical optimization* is a method that finds the optimum solution by drawing the objectives and constraints graphically. Since the graphics can only be drawn on a two-dimensional space, this method can only be used to solve the problems with two design variables. To obtain the optimum solution, the constraints are drawn on the two-dimensional design variable space which forms the feasible region. Then by drawing the iso-objective contours, the design point that makes the objective function reach the minimum in the feasible region is identified as the optimum point. Due to its property and limitations, this method is useful for the illustration of optimization methods.

Optimization problems without constraints are called *unconstrained optimization problems*. For this kind, there exist necessary and sufficient conditions for local minimum.

Necessary condition: if $f(\mathbf{x})$ has local minimum at point $\mathbf{x}^*$, then the gradient of $f(\mathbf{x})$ at $\mathbf{x}^*$ is 0.

$$\nabla f(\mathbf{x}^*) = \mathbf{0} \quad (2)$$

Sufficient condition: if the Hessian matrix $\mathbf{H}(\mathbf{x}^*)$ is positive definite at $\mathbf{x}^*$, then $\mathbf{x}^*$ is sufficiently a local minimum.

To obtain a local minimum for a function $f(\mathbf{x})$, which is the objective of an unconstrained problem, we first obtain a solution $\mathbf{x}^*$ satisfying Eq. (2). Then we check if the Hessian matrix is positive definite at $\mathbf{x}^*$. If its Hessian matrix is

positive semidefinite, higher order derivatives need to be checked.

For *constrained optimization* problem, an optimum point can be mathematically obtained using the theorem called Karush-Kuhn-Tucker (KKT) condition. The constraints are added to the objective function by multiplying each with a Lagrange multiplier, resulting in a Lagrange function. The gradient vector of Lagrange function with regard to the design vector $\mathbf{x}$ and Lagrange multipliers becomes a zero vector at local minimum. Original constraints and additional restrictions for Lagrange multipliers should be satisfied. More details about KKT condition can be found in the reference (von Laarhoven and Aarts 1987). If the objective $f(\mathbf{x})$ is a convex function defined on a convex feasible domain, the prementioned KKT condition is both necessary and sufficient condition for a global optimum.

Numerical Optimization Methods In real practice, finding the optimum solution(s) through mathematical calculations is quite difficult. It is often difficult to solve simultaneous equations which make the gradient vector be a zero vector. Besides, as the number of constraints increases, the process to evaluate many cases in KKT condition for inequality constraints becomes more complicated. It is almost impossible to solve a design optimization problem with mathematical calculations when many constraints exist in the problem. Instead, *numerical methods* using computer programming are utilized. Various numerical methods are developed specifically for different problems.

Numerical methods start from an initial design point and find an optimum in an iterative way. At each iteration, a new design is found. When at least one of the convergence criteria is satisfied, the design in the current iteration is considered as an optimum. The design is improved at each iteration through Eq. (3).

$$\mathbf{x}^{(k+1)} = \mathbf{x}^{(k)} + \Delta \mathbf{x}^{(k)}, k = 0, 1, 2, \dots \quad (3)$$

where k is the iteration number. A new design $\mathbf{x}^{(k+1)}$ is obtained by adding the change of design vector $\Delta \mathbf{x}^{(k)}$ to the current design $\mathbf{x}^{(k)}$. The change $\Delta \mathbf{x}^{(k)}$

consists of the moving direction $\mathbf{d}^{(k)}$ at the point $\mathbf{x}^{(k)}$ in the design space and the scalar step size α_k in the direction $\mathbf{d}^{(k)}$.

$$\Delta \mathbf{x}^{(k)} = \alpha_k \mathbf{d}^{(k)} \quad (4)$$

The evaluation of α_k and $\mathbf{d}^{(k)}$ depends on the algorithm used. The determination of α_k and $\mathbf{d}^{(k)}$ should reduce the objective function $f(\mathbf{x}^{(k)})$ when $\mathbf{x}^{(k)}$ is not an optimum yet. This condition is expressed as in Eq. (5).

$$f(\mathbf{x}^{(k+1)}) = f(\mathbf{x}^{(k)} + \alpha_k \mathbf{d}^{(k)}) < f(\mathbf{x}^{(k)}) \quad (5)$$

The sensitivity information, i.e., the (partial) derivative of objective and constraint functions, is used to decide the descent direction in numerical methods. Mathematical derivative functions or finite difference method can be used depending on different situations.

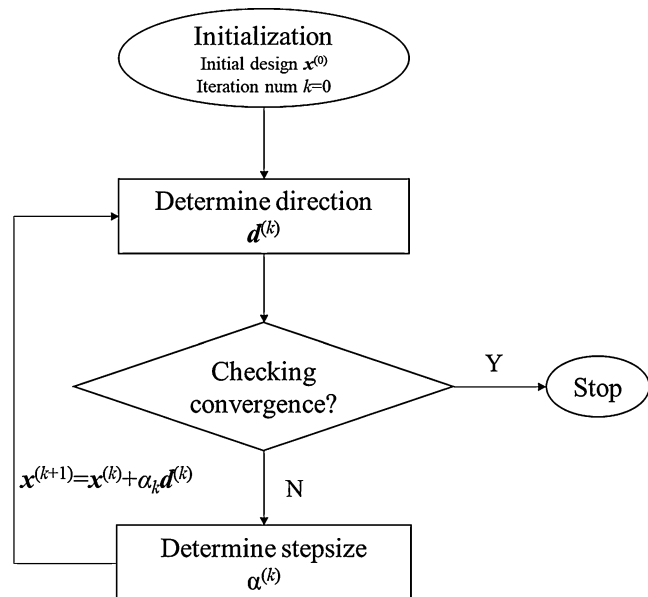
The overall steps of a numerical procedure are shown in Fig. 3.

Heuristic Optimization Methods Heuristic optimization algorithms are developed by observing different nature behaviors. They are different from classic optimization methods like gradient-

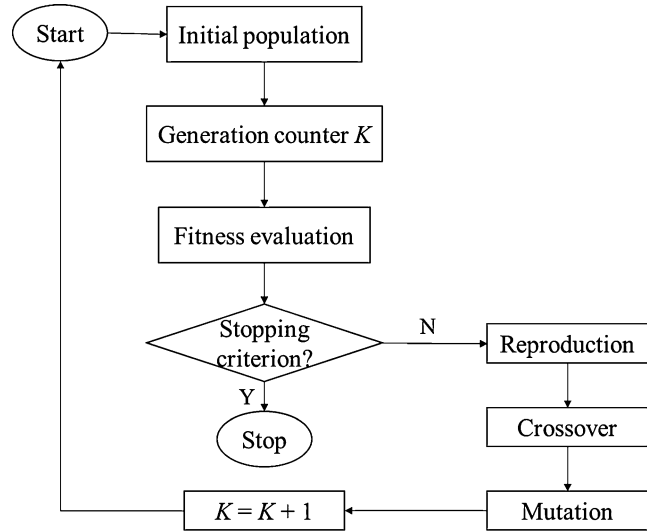
based approaches. The search process proceeds toward a solution(s) without caring how the functions are evaluated. Thus continuity or differentiability of the problem functions is not required. In addition, the methods inherently try to determine the global optimum. Since no gradient information is required, heuristic methods are easy to use and program. However, such algorithms usually require a large amount of calculations, and there is no guarantee of obtaining the global optimum.

Genetic algorithms (GA) and simulated annealing (SA) are two popular heuristic methods. GA are based on Darwin's theory of natural selection. The basic idea is to start with a set of randomly generated designs. A fitness value is assigned to each design by using its values of the objective function and penalty function for constrained problems. Processes (e.g., reproduction, crossover, and mutation) which can generate new designs with certain randomness are used to generate new designs. The size of the design set in one iteration is kept fixed, called the population. As more fit members of the set are used to create new designs, the successive sets of designs will have a higher probability of involving better fitness values. The process is continued until a stopping criterion is met. A flowchart of GA is shown

Optimal Design,
Fig. 3 General steps of
numerical method



Optimal Design,
Fig. 4 Flowchart of GA



in Fig. 4, representing a general procedure of this algorithm. The details of each step can be found in the literature (Arora 2004). The length of strings (used to represent each design) and population size are decided in the initialization stage. The fitness evaluation determines how “good” design points will be in the population. The three operations, reproduction, crossover, and mutation, make more copies of better designs and remove worse designs gradually. New designs with better quality are generated. They also provide GA the capability of jumping out from the local optimum. Designs will converge to the optimal solutions iteration by iteration.

SA is a computational stochastic technique for obtaining the global optimum. It is inspired by the thermodynamic process of annealing of molten metals to attain the lowest free energy state. When the metal is cooled slowly enough, it tends to solidify into a structure with the minimum energy. SA is a search strategy imitating the annealing process. The key of the method is to allow occasional worsening moves which may eventually help locate the design to the true global optimum. Details of SA can be found in (von Laarhoven and Aarts 1987). When setting up the problem, the appropriate initial temperature is required. New designs are generated by perturbation, and function values are evaluated. The new

design is accepted for sure if it is a better point, or it is accepted by probability if it is a worse design. The temperature is slowly reduced. Optimal solutions will be gradually reached through the SA procedure.

Multi-criteria and Multidisciplinary Design

If there is a set of criteria, $\mathbf{f} = (f_1, f_2, \dots, f_m)^T$, the design problem becomes a multi-criteria one:

$$\begin{aligned} & \min_{\mathbf{x}} \mathbf{f}(\mathbf{x}) \\ & \text{s.t. } g_j(\mathbf{x}) \leq 0, j = 1, \dots, J \\ & \quad h_k(\mathbf{x}) = 0, k = 1, \dots, K \end{aligned} \quad (6)$$

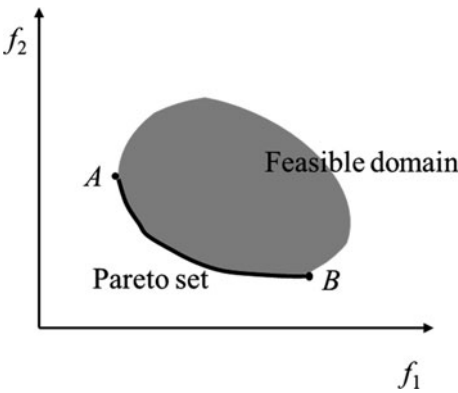
Several methods exist to convert the multi-criteria formulation into a substitute single-criterion problem which can be solved by single-objective optimization methods. The simplest scalar substitution is obtained by assigning subjective weights, $\mathbf{w} = (w_1, w_2, \dots, w_m)^T$, for multiple criteria, and the set of objectives are translated into a scalar $f = \sum_i w_i f_i(\mathbf{x})$. The weights can be considered as the design preference that can be adjusted according to designers’ preferences or different usage scenarios. For example, different combinations of preference parameters can represent different levels of trade-offs among those criteria. The solution set discovered by assigning different weights can form the

so-called Pareto set, which includes solutions that cannot improve one criterion without sacrificing others, as shown in Fig. 5. Other approaches suitable for multi-criteria design problems can be found in literature, like value-function method, goal programming (Ignizio 1976), game theory (Vincent 1983), and upper-bound formulation.

As discussed above, a system can be decomposed into different levels of subsystems. The designer must identify at which level the

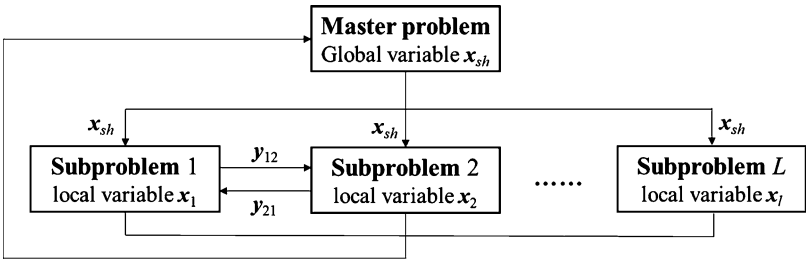
design optimal solutions are required, while it is possible that the design optima of subsystems (without considering other subsystems) may not be optimum for the entire system. For example, the GnC, vehicle body, and engine are three subsystems of an undersea vehicle system. The optimum design of the GnC may not be the best solution for the entire vehicle if not considering the design of the vehicle body or engine. For large-scale problems, treating the system as a single entity may encounter difficulties for reliable solutions. Instead, the problem can be modeled by a set of independent subproblems which are coordinated by a master problem (or called system problem). Design variables associated with the master problem are called *global variables*, while those associated with subproblems are *local variables*. *Linking variables* are used to represent the original couplings between subproblems. Hierarchical and nonhierarchical partitioning of a system are shown in Figs. 6 and 7, respectively.

Decomposition-based approaches for system design require both portioning and coordination. They are employed to improve the overall

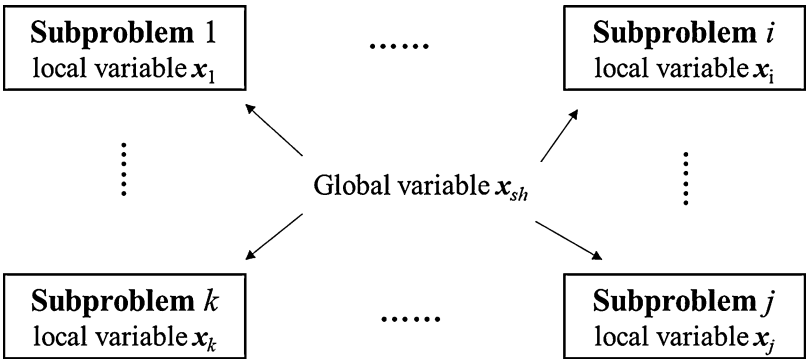


Optimal Design, Fig. 5 Feasible domain and Pareto set of bi-criteria optimization

Optimal Design, Fig. 6 Hierarchical decomposition



Optimal Design, Fig. 7 Nonhierarchical decomposition



solution efficiency. This branch of optimization design methods arises as multidisciplinary design optimization (MDO).

Robust Optimal Design

Robust design refers to the methods or procedures to develop system products whose performance is insensitive (or least sensitive) to the variations due to uncontrollable factors. As the pioneer of this research area, Taguchi first defined the loss function to evaluate the quality level of the products (Taguchi 1987). The smaller the loss, the more desirable the product. Taguchi method uses a special response variable, signal-to-noise ratio (S/N ratio), which encompasses both the mean and variation in one parameter. Design parameters that primarily affect the S/N ratio are identified, and the level of these parameters is found to minimize the response variations. Since lots of experiments are required in this parameter design, the fractional factorial design of experiments (DoE) is adopted instead of testing all combinations. Taguchi method is still popularly used in the practice although many drawbacks have been pointed out. Later, the robustness concept has been added to the conventional or so-called deterministic optimization. The objective and constraints functions are redefined with additional robustness indices. To distinguish the difference, the design variables determined by solving the conventional deterministic optimization problem

where variations (uncertainties) are not considered are called *nominal values* (or nominal solutions). *The robust optimal designs* come from the results of the robust optimization problem which takes variations into consideration. The robust optimization problem can be formulated as

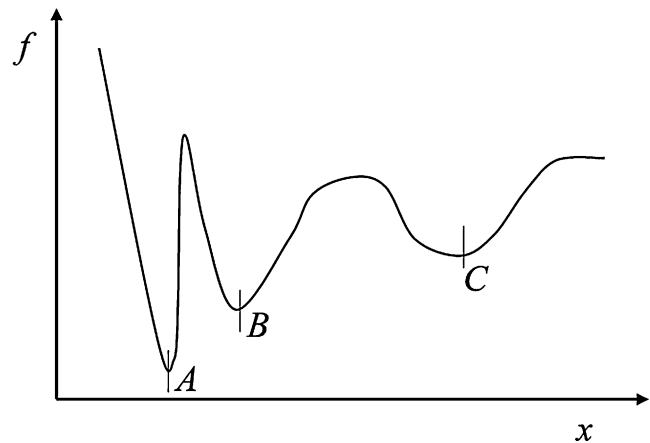
$$\begin{aligned} \min_{\mathbf{x}} & f(\mathbf{x}, \mathbf{p}) \\ \text{s.t. } & g(\mathbf{x}, \mathbf{p}) \leq 0 \\ & \mathbf{x}_{lb} \leq \mathbf{x} \leq \mathbf{x}_{ub} \end{aligned} \quad (7)$$

where $\mathbf{p}$ represents the uncertainties in design variables or parameters.

One way of defining the robustness is shown as follows. *The robustness of the objective function* is achieved by reducing the variation in the objective function due to the uncertainties. As shown in Fig. 8, the objective function reaches its minimum at the point A. However, the objective value at this point will be influenced a lot by the small variation in the design variable. On the contrary, Point B and C are less sensitive to the uncertainty in the design variable, thus being more robust. *The robustness of the constraint functions* is defined such that all constraints should be satisfied whenever design variables or parameters are changing within their variation range.

In robust optimization, objective functions can be formulated in different ways. When uncertainties are modeled by probabilistic distributions,

Optimal Design,
Fig. 8 Nominal point
versus robust point



robust optimization problems can be formulated with the mean and the standard deviations in different ways, as shown below (Park et al. 2006).

Weighted Sum with Normalization

$$f_{\text{new}} = \alpha \frac{\mu_f(\mathbf{x}, \mathbf{p})}{\mu_f^*} + (1 - \alpha) \frac{\sigma_f(\mathbf{x}, \mathbf{p})}{\sigma_f^*} \quad (8)$$

The new objective function in the robust optimization can be formulated as the weighted sum of the mean and standard deviation of the original objective function f . α is the weighting factor between minimizing the objective value and minimizing the variation of the objective function. A smaller α value will put more emphasis on the robustness. μ_f^* and σ_f^* are the base values for the mean and standard deviation, respectively, used for normalization.

Bi-objective Formulation of Robust Optimization

$$f_{\text{new}} = \{\mu_f(\mathbf{x}, \mathbf{p}), \sigma_f(\mathbf{x}, \mathbf{p})\} \quad (9)$$

In this formulation, the mean value of the objective and its variation are optimized as two objectives simultaneously.

Constraint Formulation of Robust Optimization The constraint formulation of robust optimization can be shown as Eq. (10).

$$g_{\text{new}} = \mu_g(\mathbf{x}, \mathbf{p}) + k\sigma_g^2(\mathbf{x}, \mathbf{p}) \quad (10)$$

where k is a constant value which reflects the confidence level on the variation of the constraints. For example, when $k = 3$, the constraints will be satisfied with a probability of 99.865% with normal distributions.

When uncertainties are modeled by intervals, robust optimization can be formulated by using the definition of the robustness index (Li et al. 2006). The parameters $\mathbf{p}$ with interval uncertainty are represented by their nominal value $\mathbf{p}_0$, and the corresponding variation range $\pm\Delta\mathbf{p}$. $\mathbf{p}$ can take

any value randomly in the interval $[\mathbf{p}_0 - \Delta\mathbf{p}, \mathbf{p}_0 + \Delta\mathbf{p}]$ without any assumption on the distribution. The robustness of objective and constraint functions is quantified by using the robustness indices η_f, η_g , respectively.

$$\begin{aligned} & \min_{\mathbf{x}} f(\mathbf{x}, \mathbf{p}_0) \\ & \text{s.t. } g_j(\mathbf{x}, \mathbf{p}_0) \leq 0, j = 1, \dots, J \\ & \eta = \max \{\eta_f, \eta_g\} - 1 \leq 0 \\ & lb \leq \mathbf{x} \leq ub \end{aligned} \quad (11)$$

The robustness of the objective function under interval uncertainties are defined by Eq. (12). It says that the design $\mathbf{x}_0$ is objectively robust if the worst objective variation induced by $\mathbf{p}$ is within a predefined acceptable variation range Δf_0 .

$$\begin{aligned} \eta_f \leq 1 \quad \text{where } \eta_f &= \max_{\mathbf{p}} \left| \frac{f(\mathbf{x}_0, \mathbf{p}) - f(\mathbf{x}_0, \mathbf{p}_0)}{\Delta f_0} \right| \\ & \text{s.t. } \mathbf{p}_0 - \Delta\mathbf{p} \leq \mathbf{p} \leq \mathbf{p}_0 + \Delta\mathbf{p} \end{aligned} \quad (12)$$

Eq. (13) defines the robustness of constraints, which represents that $\mathbf{x}_0$ is feasibly robust if the worst increase of the constraint function values is still bounded by the absolute constraint value at the nominal parameter value $\mathbf{p}_0$.

$$\begin{aligned} \eta_g \leq 1 \quad \text{where } \eta_g &= \begin{cases} \max_{\mathbf{p}, j} \frac{g_j(\mathbf{x}_0, \mathbf{p}) - g_j(\mathbf{x}_0, \mathbf{p}_0)}{|g_j(\mathbf{x}_0, \mathbf{p}_0)|}, \\ \quad \text{if } g_j(\mathbf{x}_0, \mathbf{p}) \geq g_j(\mathbf{x}_0, \mathbf{p}_0) \\ 0, \quad \text{otherwise} \end{cases} \\ & \text{s.t. } \mathbf{p}_0 - \Delta\mathbf{p} \leq \mathbf{p} \leq \mathbf{p}_0 + \Delta\mathbf{p}, j = 1, \dots, J \end{aligned} \quad (13)$$

After the treatment of the uncertainty using above strategies, the robust optimization problem can be solved through the optimization methods suitable for the deterministic case.

Cross-References

- [Concept Design](#)
- [Design of Submersibles](#)
- [Design Spiral](#)
- [Detailed Design](#)
- [Empirical Design](#)
- [Multidisciplinary Design Optimization \(MDO\)](#)
- [Preliminary Design](#)
- [Reliability Based Design \(RBD\)](#)
- [Structural Design](#)
- [Technical Design](#)

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Optimal Ship Operations

- [Human Factors in the Role of Energy Efficiency](#)

Optimization Algorithm

- [Multidisciplinary Design Optimization \(MDO\)](#)

Optimization of Propellers

- [New Technologies in Auxiliary Propulsions](#)

Optimized Design

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Orthogonal Frequency Division Multiplexing (OFDM)

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Outfit

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OWC – Oscillating Water Column

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Outfitting

- ▶ [Ship-Fitting Design](#)

OWT – Offshore Wind Turbine

- ▶ [Analysis of Renewable Energy Devices](#)

Painting Technology

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Definition

Ships are built on land and sail on rivers or oceans, so the main steel hull and numerous outfitting pieces will be corroded by the industrial atmosphere and the marine environment. Corrosion can cause great damage to ships and reduce the strength of ship structures (Xu 2005). When the steel ship is corroded to a certain degree, the strength of the hull will be reduced to be insufficient to withstand the huge impact of the ocean wind and waves on the hull, and a shipwreck is inevitable. When the various equipment of the ship is corroded to a certain degree, the equipment cannot work and operate normally, which will cause various accidents. In severe cases, the ship will lose control and its self-rescue ability in the ocean, causing disasters finally. Therefore, when the ship is corroded to a certain degree, it has to be scrapped and lose its use-value.

The corrosion of steel ships in the ocean is inevitable, but the rate of corrosion can be controlled. In order to prevent the corrosion of steel and prolong the service life of ships, steel, hulls, outfitting steel structural pieces, pipe system, and

cabinet must be rust removal and painted. This kind of surface treatment is called ship painting. In addition to anti-corrosion effects, ship painting also has the effects of exterior decoration and anti-foul (Liu et al. 2011).

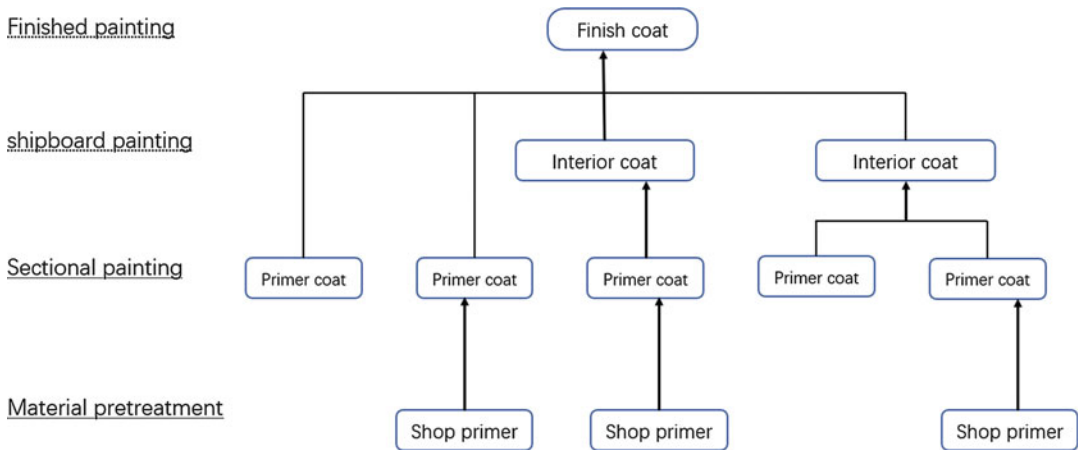
Introduction

Operation Method of Ship Painting

The ship painting method is coordinated with the block construction method of the hull construction and the regional outfitting method of ship outfitting. Therefore, ship painting operations include steel pretreatment, sectional painting, shipboard painting, and finished coat, as shown in the following figure. Among them, the painting of outfitting pieces, the painting of the brackets, bases, strong back, and other outfitting pieces connected with the hull belong to the scope of sectional painting; the painting of other outfitting units and outfitting pieces should be completed at the later stage of unit outfitting and outfitting pieces production, so they are all in-site operations (Fig. 1).

Steel Pretreatment

Steel pretreatment is the leveling, rust removal (including oxide scale), and protective paint on the steel. The process method and process of steel pretreatment have been discussed in detail above, and the automatic assembly line of steel



Painting Technology, Fig. 1 Painting stage and process

pretreatment has been briefly introduced, so there is no need to repeat it here (Li et al. 2006).

Sectional Painting

After the steel pretreatment is made into sections by component processing and structural pre-assembly welding, the surface of the steel structure will inevitably produce zinc salt, rust, oil stains, moisture, etc., so a second rust removal is required, and then spray primer according to the different requirements of the sections. The secondary rust removal and primer spraying operations for the sections are called sectional painting. In order to control the environmental conditions of the painting operation and ensure the quality of the painting, indoor operations should be used for sectional painting, so the second stage of rust removal is best to use shot blasting in the shot blasting room. It uses the pressure of high-speed flowing compressed air in the air duct to make iron shots impact the rust spots on the metal surface to achieve rust removal. In addition to the iron shot blasting device, the shot blasting room is also equipped with rails and section loading vehicles, iron shot recovery devices and rust dust suction devices to ensure shot blasting operating conditions, and rust removal quality. Outside the house is equipped with sectional dispatching equipment and indoor workplaces for sectional spray primer. The primer should be sprayed immediately after rust removal in sections,

otherwise yellow rust will easily occur and affect the quality of rust removal. Sometimes due to various conditions, it is necessary to carry out sectional painting operations on the field platform. At this time, most of the sectional rust removal uses pneumatic tools, such as pneumatic grinder, pneumatic control wire brushes, etc.; primer spraying uses portable airless spraying equipment (Wang et al. 2019).

Shipboard Painting

The painting work on the berth (in the dock) before launching the ship and at the dockside after launching is called shipboard painting. The painting operations onboard include rust removal, cleaning, touch-up painting, and spraying of the inner and top coats on various parts of the outer surface of the hull. If there is no subsequent outfitting work for the oil tanks, water tanks, and other liquid tanks inside the hull, painting operations can be completed on the hull (in the dock). However, due to the busy outfitting operation in the ship, the hull deck is easy to wear and tear the paint film and is left to be painted at the later stage of outfitting at the wharf (He et al. 2020).

Finished Painting

The painting operation before delivery is called finished painting. Finished painting is generally completed within a period of time from the end of the trial voyage to the delivery of the ship, so that

when that ship is hand over to the owner, it will look quite new. However, as long as the ship-owner agrees, the finished painting operation can also be carried out in advance. Finished painting operations include the cleaning and touch-up of the outer surface of the superstructure, deck, and deck machinery, as well as the spraying of the last topcoat on the outer surface of the hull; organic solvents are used to clean up grease and other stains before the paint is repaired. The last coat of paint on each part of the outer surface of the main hull is usually sprayed in the dock. Firstly, the bottom of the ship is washed with clean water to remove salt and soil. Secondly, the paint is repaired after drying (e.g., the first paint film is often damaged by the anchoring test). Thirdly, the finish coat is sprayed.

Common Paint for Ships

Paint is usually composed of binders, pigments, diluents, and additives. Among them, the binder is the main film-forming substance; the pigment is the secondary film-forming substance, which can color the paint film and improve the performance of the paint film; the diluent, a volatile substance is used to dissolve and dilute the paint for operation; the additive is used to make the paint film have specific properties, such as antifouling, corrosion inhibition, and plasticization. Paint can be classified according to binders. The commonly used binders are natural resin, mineral asphalt, and synthetic resin. Synthetic resin is widely used in modern marine paint, mainly including alkyd resin, ethylene resin, epoxy resin, chlorinated rubber, polyurethane resin, and inorganic zinc paint.

Cathodic Protection of the Hull

It is not enough for ships to rely on paint to prevent corrosion. For example, exposed copper propellers in water and contact with steel structures such as propeller shafts and stern posts exposed in the water will produce battery effects and cause corrosion of the steel components at the

tail. Another example is that the poor paint film of the bottom of the ship at the keel block, insufficient thickness of paint film at the weld seam of the outer plate, and local paint film scratches on the side outer plate will cause electrochemical corrosion in the marine environment (Cho et al. 2016). In order to make up for the deficiencies of anti-corrosion treatment, cathodic protection is generally used as an auxiliary means of anti-corrosion for the hull. According to the electrochemical principle, the metal with lower electroplating potential will be used as the anode in the corrosion battery to release electrons, thus causing it to be corroded. Cathodic protection is to polarize the protected metal as a cathode so that it can capture electrons and be protected. According to different polarization principles, there are two kinds of cathodic protection: sacrificial anode method and impressed current method.

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Parametric Roll Resonance

► [Parametric Rolling](#)

Parametric Rolling

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Synonyms

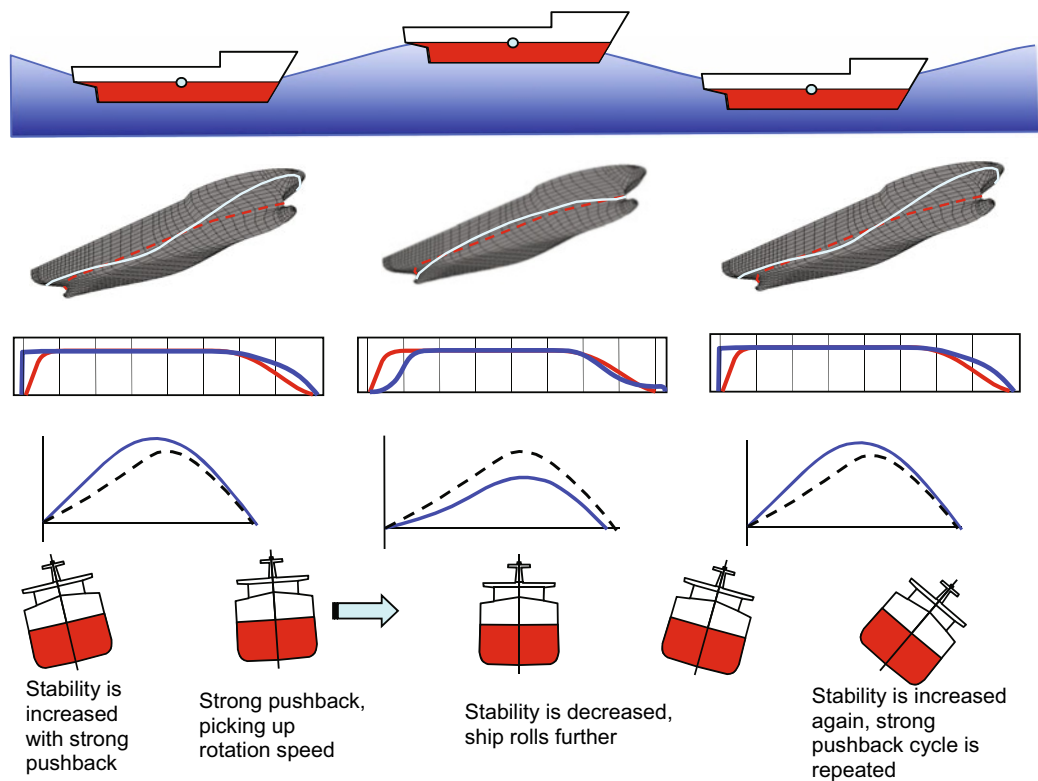
Low cycle resonance; Parametric roll resonance;
Parametrically excited roll

Definition

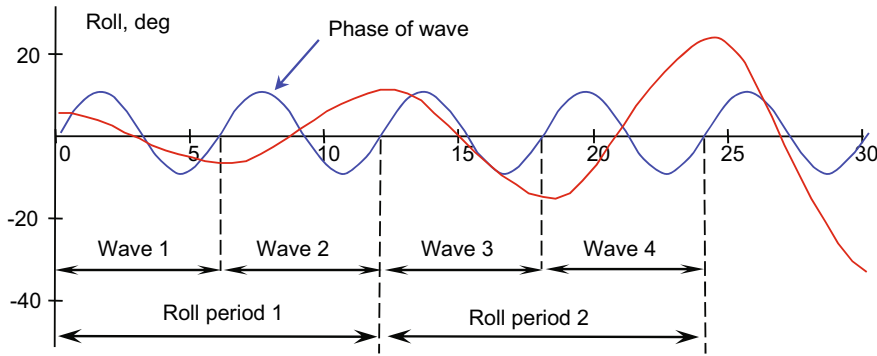
Parametric rolling is a dynamic stability phenomenon in which an amplification of roll motion is caused by periodic variation of transverse stability

in waves. The phenomenon of parametric rolling is predominantly observed in head, following, bow, and stern-quartering seas when the ship's encounter frequency is approximately twice that of the ship's roll natural frequency and the ship's roll damping is insufficient to dissipate additional energy (SDC 3/WP.5/Annex 4,2016).

Figures 1 and 2 shows the process by which parametric rolling develops. If the ship rolls while in the wave trough, increased stability provides stronger pushback, or restoring moment. As the ship returns to the upright position, its roll motion rate is increased, since there was an additional pushback from the increased stability. At that time, however, the ship will roll further to the opposite side because of the greater roll motion rate and less resistance to heeling. Then, if the wave trough reaches the midship section when the ship reaches its maximum amplitude roll, stability increases again and the cycle starts again.



Parametric Rolling, Fig. 1 Development of parametric roll resonance (Belenky et al. 2011; SDC 3/WP.5/Annex 4,2016)



Parametric Rolling, Fig. 2 Time histories plots of parametric roll resonance (SDC 3/WP.5/Annex 4,2016)

There are two waves that pass during each roll period (SDC 3/WP.5/Annex 4,2016).

Scientific Fundamentals

The roll motion in head or following seas can be expressed as follows:

$$\begin{aligned} & (I_{xx} + A_{44})\ddot{\varphi} + N_1\dot{\varphi} + W \\ & \cdot GM\left(1 + \frac{\Delta GM}{GM} \cos \omega_e t\right) \varphi \\ & = 0 \end{aligned} \quad (1)$$

where φ : roll angle, N_1 : linear roll damping coefficient, W : ship weight, I_{xx} : moment of inertia in roll, A_{44} : added moment of inertia in roll, GM : metacentric height in calm water, ΔGM : the variation of metacentric height in waves, ω_e : encounter wave frequency, and t : time. The dot denotes the differentiation with time.

The roll equation (1) can be rewritten as follows:

$$\ddot{\varphi} + 2\mu\dot{\varphi} + \omega_0^2\left(1 + \frac{\Delta GM}{GM} \cos \omega_e t\right) \varphi = 0 \quad (2)$$

where $2\mu = \frac{N_1}{I_{xx} + A_{44}}$, $\omega_0 = \sqrt{\frac{W \cdot GM}{I_{xx} + A_{44}}}$, ω_0 : natural roll frequency.

The roll equation (2) can be further rewritten as follows:

$$\ddot{\varphi} + 2\mu^*\dot{\varphi} + (\delta + 2\varepsilon \cos 2\tau)\varphi = 0 \quad (3)$$

where $\mu^* = \frac{2\mu}{\omega_e}$, $\delta = 4\frac{\omega_0^2}{\omega_e^2}$, $2\varepsilon = 4\frac{\omega_0^2}{\omega_e^2} \frac{\Delta GM}{GM}$, $\tau = \omega_e t/2$.

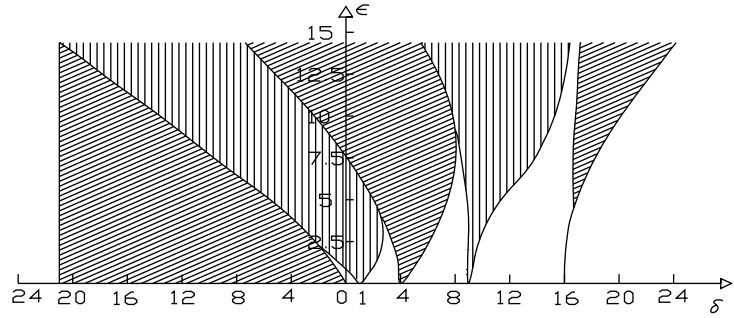
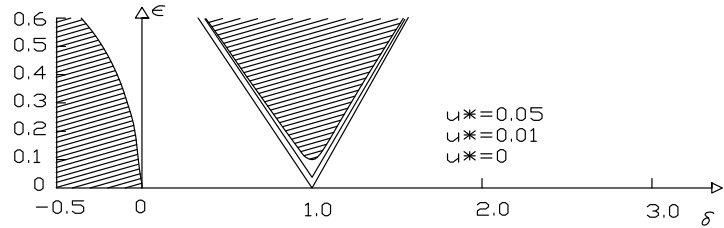
If the damping coefficient is ignored, the standard Mathieu equation can be obtained as follows:

$$\ddot{\varphi} + (\delta + 2\varepsilon \cos 2\tau)\varphi = 0 \quad (4)$$

The solution of the standard Mathieu equation is as follows, and the stability diagram of the Mathieu equation is shown in Fig. 3.

$$\begin{aligned} \delta_0 &= -\frac{1}{2}\varepsilon^2 + O(\varepsilon^3), \\ \delta_1 &= 1 - \varepsilon - \frac{1}{8}\varepsilon^2 + O(\varepsilon^3), \delta'_1 = 1 + \varepsilon - \frac{1}{8}\varepsilon^2 + O(\varepsilon^3) \\ \delta_2 &= 4 - \frac{1}{12}\varepsilon^2 + O(\varepsilon^3), \delta'_2 = 4 + \frac{5}{12}\varepsilon^2 + O(\varepsilon^3) \end{aligned} \quad (5)$$

From the stability diagram, we see that unstable motion can occur at several different ratios of ω_e/ω_0 . The shadow area means unstable region. The predominant value of this ratio is $\delta = 4\omega_0^2/\omega_e^2 = 1$, meaning that oscillation at the natural frequency occurs if the frequency of encounter is twice the natural frequency. This instability phenomenon is an example of a dynamic motion “bifurcation.” A small initial disturbance $2\varepsilon = \Delta GM/GM$ can trigger a growing oscillation, meaning that a small variation of GM in waves could trigger a growing roll motion, even capsizing in this unstable region. The unstable region becomes large as ΔGM increases. This phenomenon is called parametric rolling.

Parametric Rolling,**Fig. 3** Stability diagram of the Mathieu equation**Parametric Rolling,****Fig. 4** Stability diagram of the Mathieu equation for parametric rolling

The basic value of this ratio is $\delta = 4\omega_0^2/\omega_e^2 = 4$, meaning that oscillation at the natural frequency occurs if the frequency of encounter equals the natural frequency. This instability phenomenon is not discussed here.

If roll damping is considered as shown in Eq. (3), the solution of the Mathieu equation for parametric rolling is as follows, and the stability diagram of the Mathieu equation for parametric rolling is shown in Fig. 4.

$$\delta - 1 = -\frac{1}{8}\epsilon^2 \pm \sqrt{\epsilon^2 - (2\mu^*)^2} \quad (6)$$

The shadow area means an unstable region, and there is a threshold value $\epsilon = 2\mu^*$ for unstable region. If $2\mu^* \geq \epsilon$, it is stable. If $2\mu^* < \epsilon$, it is unstable, meaning that if $4\mu^* < \frac{\Delta GM}{GM}$ and $\omega_0/\omega = 1/2$, parametric rolling could occur.

Typical Parametric Rolling Accident

In late October 1998, a C11 class, post-Panamax containership, which is 262 m LBP, and 24.45m Depth with a loaded draft of about 12.5m, east-bound in the north Pacific from Kaohsiung to

Seattle, was overtaken by a severe storm. She carried a deck stow of some 1300 containers. As encounter with the storm became inevitable, the master rode out hove to the rough seas and winds. The storm encounter lasted for some 11 hours, mostly at night. During the period of severest motions, port and starboard rolling of as much as 35–40 degrees was reported simultaneously with extreme pitching. The master later described the ship as absolutely out of control. About 400 containers were lost overboard, and another 400 collapsed or crushed during these motions (SLF45/6/7,2002) (Fig. 5).

In February 2003, the Wallenius PCTC M/V Aida experienced sudden violent rolling in rough head seas southwest of the Azores. Roll angles as large as 50 degrees were read off the bridge inclinometer. When this incident was postanalyzed, it was found that the condition, in terms of the relation of wave encounter period and natural roll period, was such that parametric rolling was most likely the case. Partly due to this accident, M/V Aida was equipped with a Seaware EnRoute Live system in December 2003 for trial during the winter. In February 2004, on a voyage from Southampton to New York, Aida encountered heading sea with significant wave height 5–6 m and traveled with reduced speed 8–10 knots west

Parametric Rolling,

Fig. 5 APL China containership suffered parametric rolling (<http://marineinbox.com/marine-exams/parametric-rolling-in-container-ships/>)



of the Azores. At five different occasions, parametric rolling evolved, with roll angles up to 17 degrees. The strong pitch-roll coupling during head-sea-parametric rolling is clearly demonstrated by the recorded motion time series with the Seaware EnRoute Live system (SLF47/INF.5 2004).

Historical Development

Before the accident of the C11 class containership, parametric rolling is mainly dealt with in the cases of following waves (Kerwin 1955) or beam waves (Blocki 1980). In particular, clear experimental records of capsizing due to parametric rolling in following waves were published by Umeda (Umeda et al. 1995). In case of following waves, the encounter frequency is much lower than the natural frequencies of heave and pitch so that coupling with heave and pitch is not important. In addition, added resistance in following waves is generally small. Thus, several successful predictions of parametric rolling in following waves were reported (Munif and Umeda 2000). In case of head waves, however, prediction of parametric rolling is not so easy because coupling with heave and pitch is significant and added resistance cannot be ignored. The effect of dynamic heave and pitch motions on parametric rolling was investigated so far by many researchers and is well established: Restoring

arm variation in head waves depends on dynamic heave and pitch (Taguchi et al. 2006). On the other hand, effect of added resistance on parametric rolling seems to require further investigation. Jensen (Jensen et al. 2007) discussed the effect of surge motion on the probability of parametric rolling in irregular head waves, but their surge model does not include added resistance. Umeda (Umeda et al. 2008) and Umeda and Francescutto (Umeda et al. 2008) executed numerical simulation of parametric rolling in regular and irregular head waves with added resistance taken into account, but their hydrodynamic prediction method for added resistance is different from that for restoring variation. Lu et al. (2011) further developed a numerical prediction method for parametric roll in head waves in which both the restoring variation and the Kochin function for added resistance are calculated with a strip theory.

Key Numerical Method for Parametric Rolling

It is widely accepted that degrees of freedom (DOF) of a rigid body in a three-dimensional space are six but can be reduced if constraints are added. First of all, because of head waves, we can assume that a ship runs with the constant course so that there is no sway and yaw motions are two constraints so that the degrees of freedom become four. Then if the heave and pitch motions

are assumed to be independent from the surge and roll motions, the heave and pitch motions themselves can be regarded as two constraints so that the degrees of freedom become two. Further, if we assume that the ship runs with a constant speed, the degrees of freedom become finally one.

A coupled heave-roll-pitch-mathematical model which is based on a nonlinear strip theory called 3 DOF approach is widely used for predicting parametric rolling. The mathematical model is expressed as (7), (8), and (9).

$$\begin{aligned}
 (m + A_{33}(\varphi))\ddot{\zeta} + B_{33}\dot{\zeta} + A_{34}(\varphi)\ddot{\varphi} \\
 + B_{34}(\varphi)\dot{\varphi} \\
 + A_{35}(\varphi)\ddot{\theta} \\
 + B_{35}(\varphi)\dot{\theta} \\
 = F_3^{FK+B}(\xi_G/\lambda, \zeta, \varphi, \theta) \\
 + F_3^{DF}(\varphi)
 \end{aligned} \quad (7)$$

$$\begin{aligned}
 (I_{xx} + A_{44}(\varphi))\ddot{\varphi} + N_1\dot{\varphi} + N_3\dot{\varphi}^3 + A_{43}(\varphi)\ddot{\zeta} \\
 + B_{43}(\varphi)\dot{\zeta} + A_{45}(\varphi)\ddot{\theta} + B_{45}(\varphi)\dot{\theta} \\
 = F_4^{FK+B}(\xi_G/\lambda, \zeta, \varphi, \theta) + F_4^{DF}(\varphi)
 \end{aligned} \quad (8)$$

$$\begin{aligned}
 (I_{yy} + A_{55}(\varphi))\ddot{\theta} + B_{55}(\varphi)\dot{\theta} \\
 + A_{53}(\varphi)\ddot{\zeta} + B_{53}(\varphi)\dot{\zeta} + A_{54}(\varphi)\ddot{\varphi} \\
 + B_{54}(\varphi)\dot{\varphi} \\
 = F_5^{FK+B}(\xi_G/\lambda, \zeta, \varphi, \theta) + F_5^{DF}(\varphi)
 \end{aligned} \quad (9)$$

A_{ij} , B_{ij} , and C_{ij} are the coupling coefficients in

seakeeping theory. Nonlinear Froude-Krylov forces (FK) and hydrostatic force (B) are calculated by integrating the incident wave pressure around the instantaneous wetted hull surface by two-dimensional strip method or three-dimensional panel method. Radiation and diffraction forces are calculated for the submerged hull considering time-dependent roll angle with the static balance retained in sinkage and trim by two-dimensional strip method or three-dimensional panel method. The hydrodynamic forces in heave and pitch are calculated at the encounter frequency while those for roll mode are calculated at half the encounter frequency assuming parametric roll occurs. The linear and cubic roll-damping coefficients are used in the mathematic model which are obtained from the roll decay test in the experiment or Ikeda's method. The model used here also includes the body-nonlinear hydrodynamic coupling between roll and vertical motions, and the nonlinearity of vertical motions is ignored.

Equation (8) can be rewritten as below if one only considers Froude-Krylov components in the roll-restoring variation.

$$\begin{aligned}
 (I_{xx} + A_{44}(\varphi))\ddot{\varphi} + N_1\dot{\varphi} + N_3\dot{\varphi}^3 \\
 = F_4^{FK+B}(\xi_G/\lambda, \zeta, \varphi, \theta)
 \end{aligned} \quad (10)$$

The 3 DOF approach can also be simplified to 1 DOF approach if the heave and pitch motions are obtained by a strip theory applied to an upright hull.

$$\begin{aligned}
 \ddot{\varphi} + 2\mu\dot{\varphi} + \gamma\varphi^3 + \frac{W}{I_{xx} + A_{44}}GZ_{FK+R\&D}(t, X_G, \zeta_G, \theta, \varphi) = 0 \\
 GZ_{FK} = \frac{-F_4^{FK+B}(\xi_G/\lambda, \zeta, \varphi, \theta)}{W} \\
 GZ_{R\&D} = \frac{-F_4^{DF}(\varphi) + \left(A_{43}(\varphi)\ddot{\zeta} + B_{43}(\varphi)\dot{\zeta} + A_{45}(\varphi)\ddot{\theta} + B_{45}(\varphi)\dot{\theta} \right)}{W}
 \end{aligned} \quad (11)$$

FK: Only Froude-Krylov and hydrostatic components in roll direction are considered.

FK+R&D: The radiation and diffraction components in roll direction are also considered.

If the time-varied speed due to the surge motion is considered in 1 DOF approach, 2 DOF numerical approaches can be realized. If the time-varied speed due to the surge motion is considered in 1 DOF approach, 4 DOF numerical approaches can be realized. The time-varied speed due to the surge motion can be shown as follows.

$$\begin{aligned} (m + A_{11}) \ddot{X}_G &= \left(T \dot{X}_G, n \right) - R \left(\dot{X}_G \right) \\ &+ F_1^{FK} (X_G, t) \\ &- R_{AW} \left(\overline{X}_G, t \right) \end{aligned} \quad (12)$$

$$T(X_G, n) = (1 - t_p) \rho n^2 D_p^4 K_T \left\{ \frac{X_G (1 - w_p)}{n D_p} \right\} \quad (13)$$

$$\overline{X}_G = \frac{1}{T_e} \int_0^{T_e} \dot{X}_G(t) dt \quad (14)$$

where φ : roll angle, μ : linear roll-damping coefficient, γ : cubic roll-damping coefficient, W : ship weight, I_{xx} : moment of inertia in roll, A_{44} : added moment of inertia in roll, I_{yy} : moment of inertia in pitch, A_{55} : added moment of inertia in pitch, GZ : righting arm, t : time, ζ_G : heave displacement and θ : pitch angle, and X_G : instantaneous ship-longitudinal position. The dot denotes the differentiation with time.

Furthermore, T : propeller thrust, R : ship resistance in calm water, and R_{AW} : added resistance in waves.

Experimental Method for Parametric Rolling

Free Running Experiment Method

The ship model was driven by a propeller in regular head seas in the free running experiment. The pitch and roll amplitudes were measured by a MEMS (Microelectromechanical system)-based gyroscope placed on the ship model, and the

wave elevation was measured by a servo-needle wave height sensor attached to the towing carriage.

Partially Restrained Experiment

A multifunctional equipment was used in the partially restrained experiment. The equipment as shown in Fig. 6 consists of five components. The first component is a pedestal and four locking jaws for fixing the equipment to the towing carriage. The second component is a mechanism for keeping the ship model free in heave and measuring the displacement of heave motion and consists of a guide rail bracket, a linear guide rail pair, a displacement sensor using a guide wire, a heave rod, a block, and so on. The third component is three diverging combined sensors using rods for measuring the surge force, the sway force, and the yaw moment in which a strain gauge-measuring technique is applied. The fourth component is a complex movement mechanism for measuring



Parametric Rolling, Fig. 6 The new equipment for restrained experiment

pitch, roll motions with potentiometers, and roll moment with a double flange torque sensor by fixing a certain roll angle with a locking mechanism. The fourth component is fixed to the ship model, and its weight is regarded as one part of the ship model. The third component is fixed with the fourth component, and the second component is fixed with the third component, and their weights are not part of the ship model for the convenience of arranging and adjusting the center of gravity and moments of inertia. The fifth component is a mechanism for balancing the weight of the second component and third component and consists of a pulley block, a vertical guide rod, a counter-weight, and so on.

Surge, sway, and yaw motions are always restrained; heave and pitch motions are always free, and roll motion can be free or restrained by the equipment in partially restrained experiments. Therefore, parametric roll, pitch, and displacement of heave can be measured simultaneously. When roll is fixed with a certain angle, the roll moment and the sway force can be measured simultaneously by this equipment, and then the restoring moment can be calculated by roll moment, sway force, and vertical distance between the double flange torque sensor of the equipment and the center of gravity of the ship model.

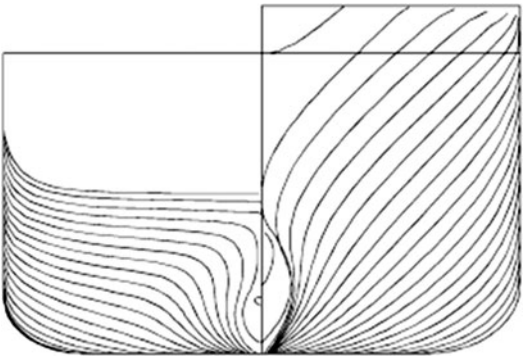
The principal particulars and body plan of the C11 class containership are shown in Table 1 and Fig. 7, respectively. The ship models the free running experiment and partially restrained experiment are shown Figs. 8 and 9, respectively.

Parametric Rolling, Table 1 Principal particulars of the C11 containership

Items	Ship	Model
Length: L	262.0m	4.000m
Draft: T	11.5m	0.176m
Breadth: B	40.0m	0.611m
Depth: D	24.45m	0.373m
Displ.: W	67508 ton	240.2kg
C_B	0.560	0.560
GM	1.928m	0.029m
T_ϕ	24.68s	3.05s
K_{YY}	0.24L	0.24L

The Effect of the Surge Motion on Parametric Rolling

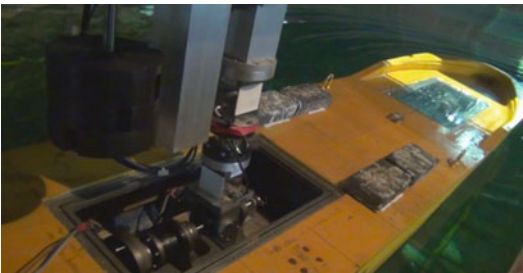
The uncoupled roll model (1 DOF) and the coupled surge-roll model (2 DOF) with added resistance taken into account are used to numerically investigate the effect of surge motion on parametric roll. An example of time series is shown in Fig. 10. Free running model experiments, which allow the surge motion and partially



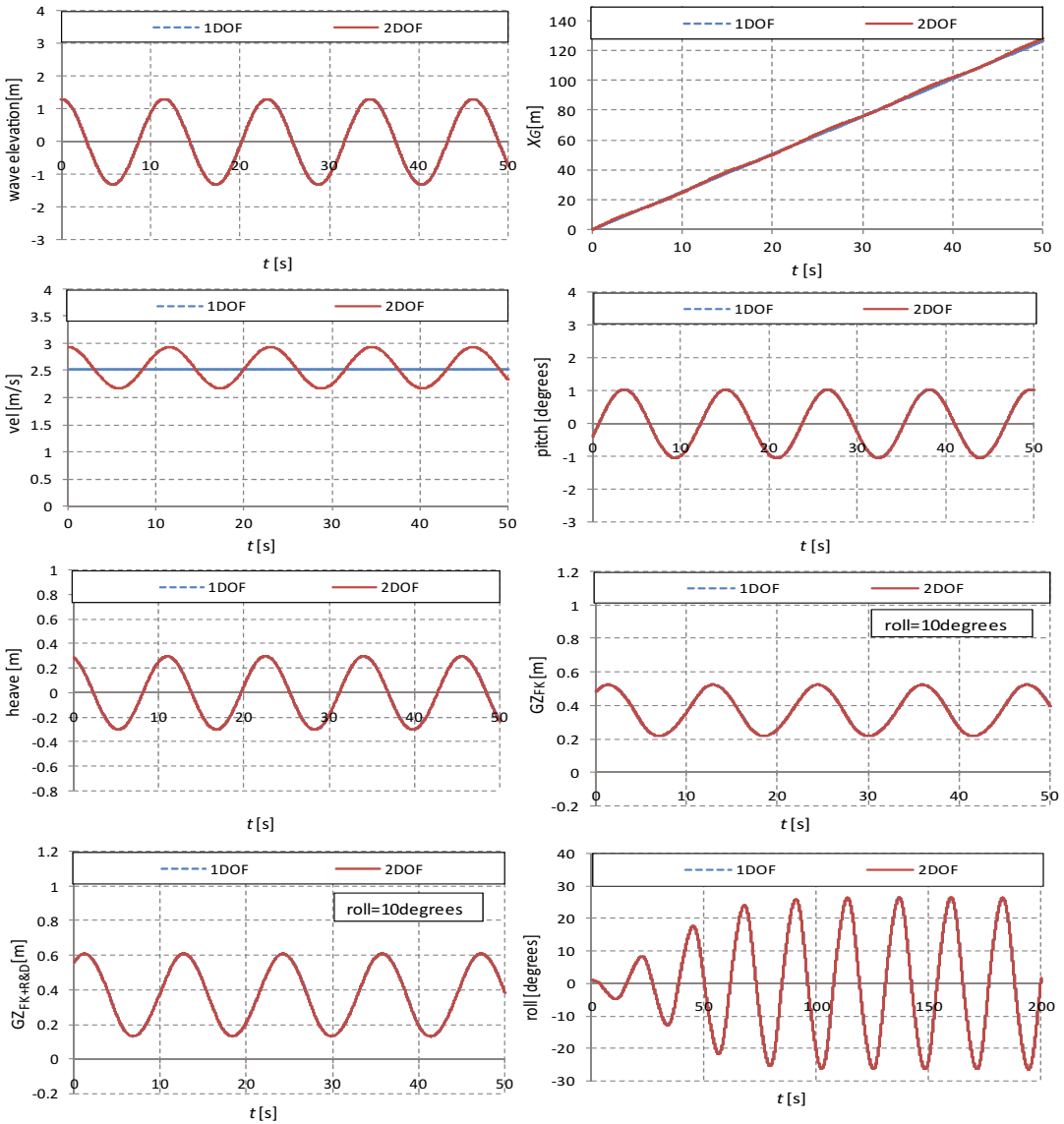
Parametric Rolling, Fig. 7 Lines of C11 containership



Parametric Rolling, Fig. 8 The ship model in free running experiment



Parametric Rolling, Fig. 9 The ship model in partially restrained experiment



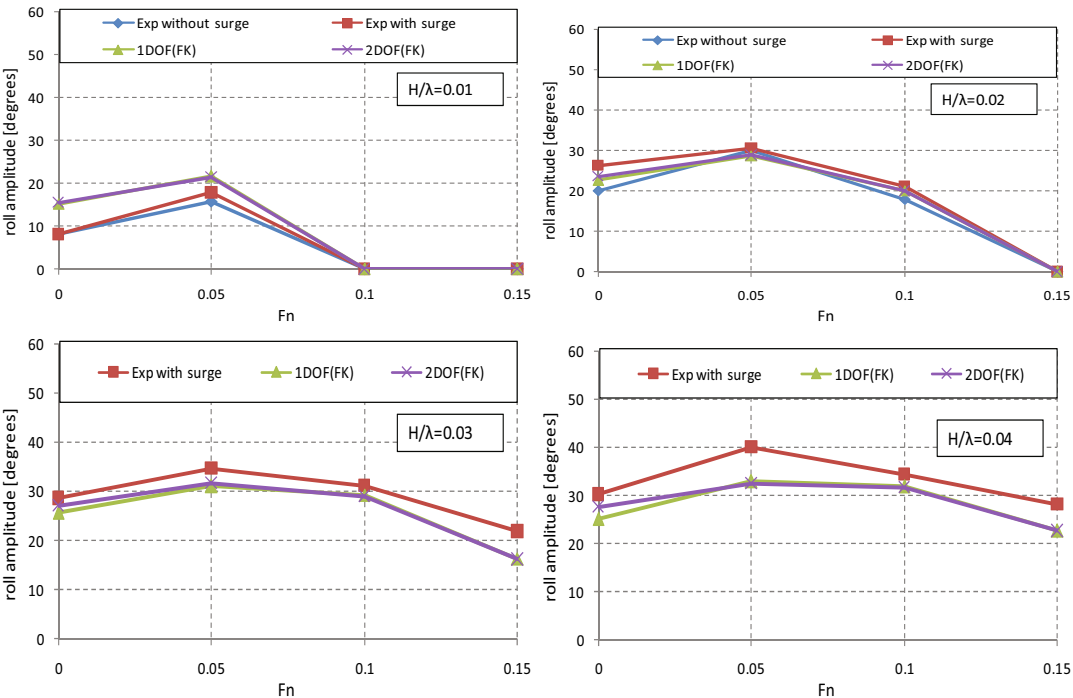
Parametric Rolling, Fig. 10 Comparisons between time series with surge motion and without surge motion, under the condition of the actual $F_n=0.05$, $\lambda/L_{pp}=1.0$,

$H/\lambda=0.01$, and $\chi=180^\circ$. (1DOF(FK+R&D) and 2DOF (FK+R&D) approaches)

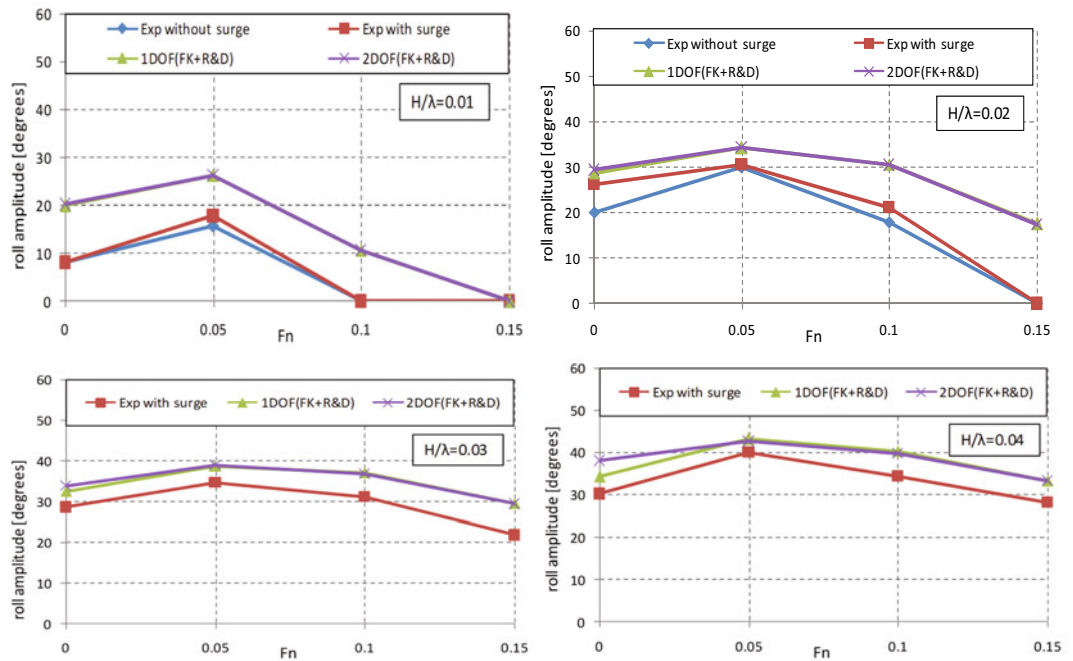
restrained model experiments without surge motion, were also conducted to investigate the effect of surge motion on parametric roll. More comprehensive comparisons are shown in Figs. 11 and 12.

The simulations indicate that the effect of the surge motion on parametric rolling is generally small in regular head seas. This is because the difference of the longitudinal ship position is

rather small between the calculations with and without surge motion; consequently, the difference between the relative wave profile around the ship and the change of GZ are very small too, although the ship's forward speed varies periodically due to the surge motion in regular head seas. This conclusion can be applied to the modeling of the roll-restoring variation with and without the radiation and the diffraction components.



Parametric Rolling, Fig. 11 The effect of surge motion on parametric roll as function of Froude number in experiments and simulations with $\lambda/L_{pp}=1.0$, $\chi=180^\circ$ (FK: Only Froude-Krylov components in roll direction are considered)



Parametric Rolling, Fig. 12 The effect of surge motion on parametric roll as function of Froude number in experiments and simulations with $\lambda/L_{pp}=1.0$, $\chi=180^\circ$ (FK+R&D: The radiation and diffraction components in roll direction are also considered)

The Effect of Parametric Rolling on Heave and Pitch Motions

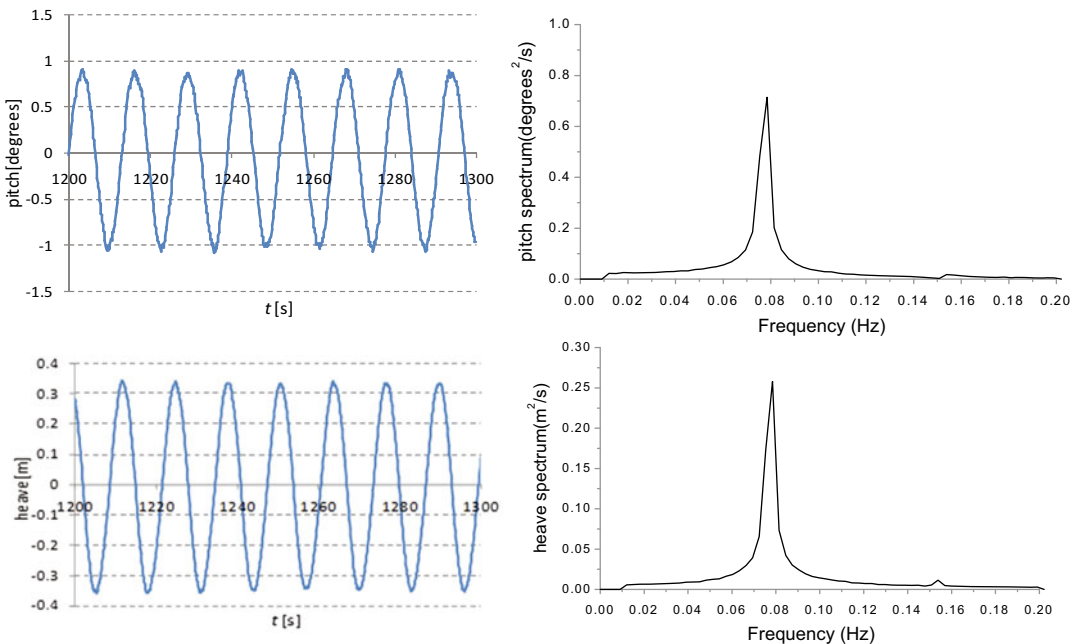
The effect of dynamic heave and pitch motions on parametric rolling was investigated so far by many researchers, and it is well established that restoring arm variation in head waves depends on dynamic heave and pitch motions. Coupling from heave, pitch to parametric roll is usually considered, but the effect of parametric roll on heave and pitch motions is ignored. Partially restrained model experiments free in heave and pitch but with a constant roll angle were conducted, and then, partially restrained model experiments free in heave, pitch, and roll for simultaneously measuring parametric roll, pitch, and heave were conducted, and the results are shown in Figs. 13 and 14. The spectra of the ship motions by the Fourier transformation are also shown in Figs. 13 and 14.

The results from the experiments indicate the frequency of heave and pitch motions is equal to the encounter wave frequency in the case without parametric roll as shown in Fig. 13, which is consistent with a linear seakeeping theory. When

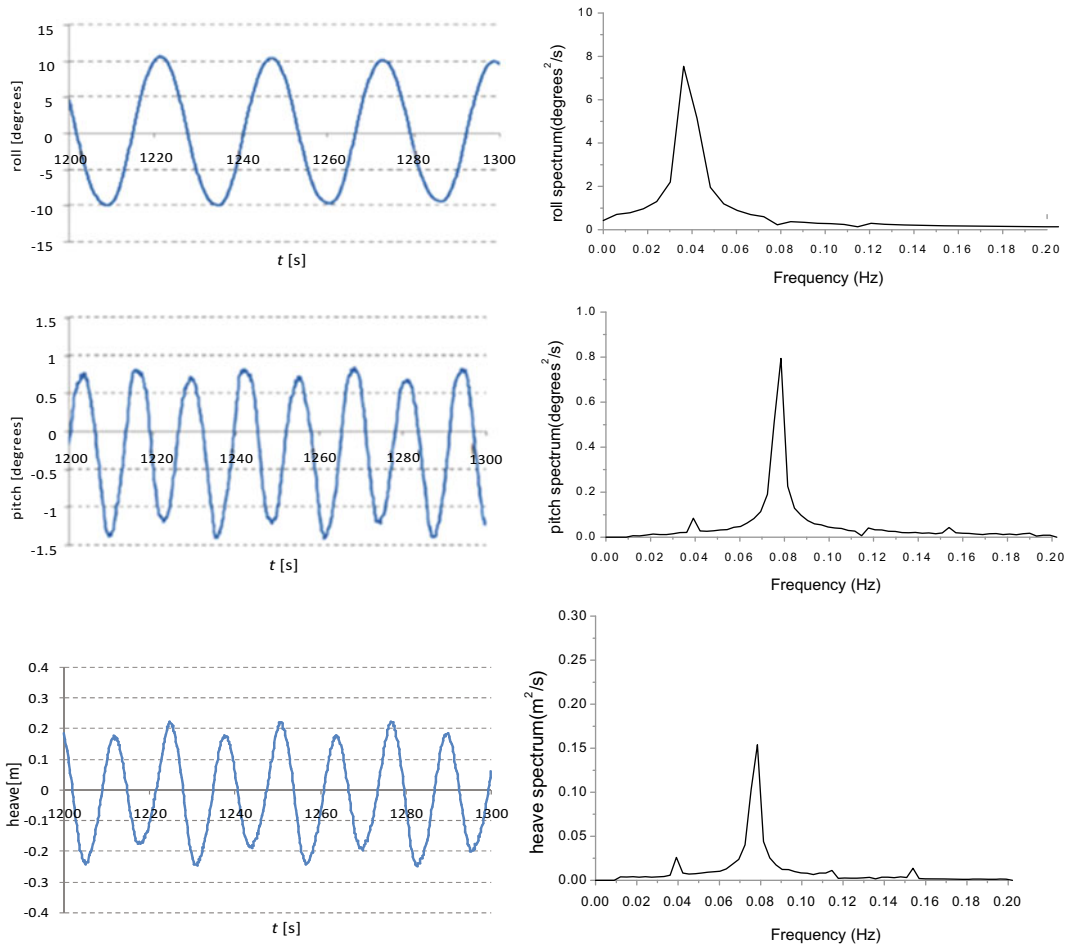
parametric roll occurs with the amplitudes of about 10 degrees as shown in Fig. 14, heave and pitch motions are affected by parametric roll: Their large and small amplitudes alternately appear. This phenomenon seems like “subharmonic pitch” and “subharmonic heave.” The heave and pitch motions are analyzed in the frequency domain by the Fourier transformation. One distinct observation was that pitch and heave motions in the experiments have both half the encountered wave frequency and the encountered wave frequency components when parametric roll occurs.

The Effect of Dynamic Roll Variation on Parametric Rolling

Parametric rolling in head seas is a nonlinear phenomenon due to roll-restoring force variation. The calculated roll motions with the 1 DOF approach and the 3 DOF approach are shown in Figs. 15 and 16 together with the results of free running model. The calculations of the roll-restoring variation are performed with and



Parametric Rolling, Fig. 13 Pitch and heave motions under the constant roll angle in the partially restrained experiment with $\lambda/L_{pp}=1.0$, $H/\lambda=0.01$, $\chi=180^\circ$, and $F_n=0.0$



Parametric Rolling, Fig. 14 Roll pitch and heave motions in the partially restrained experiment allowing the roll motions with $\lambda/L_{pp}=1.0$, $H/\lambda=0.01$, $\chi=180^\circ$, and $Fn=0.0$

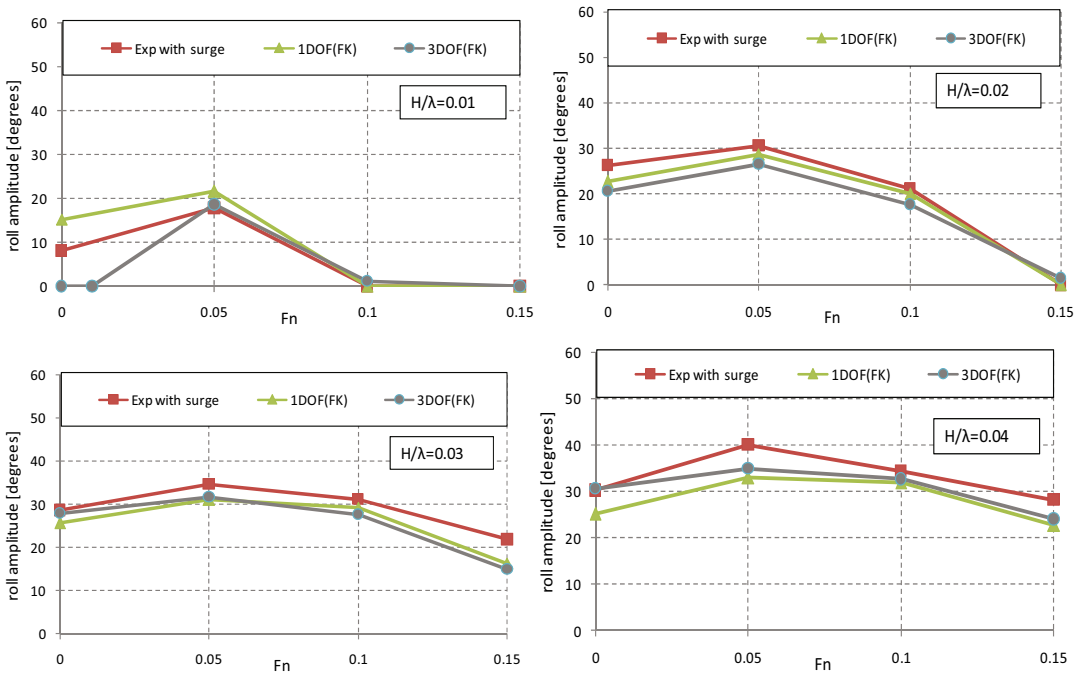
without the radiation and diffraction components as shown in Figs. 17 and 18.

The prediction of parametric rolling in the 1 DOF approach and 3 DOF approach with Froude-Krylov, radiation, and diffraction components is generally larger than that in the experiments while the prediction of parametric roll with the Froude-Krylov on its own is generally smaller than that in the experiments except for $H/\lambda=0.01$ as shown in Figs. 15 and 16. The prediction of parametric roll with the 1 DOF approach is generally larger than that with the 3 DOF approach as shown in Figs. 15 and 16, while the prediction with the 3 DOF (FK+R&D) approach is closer to

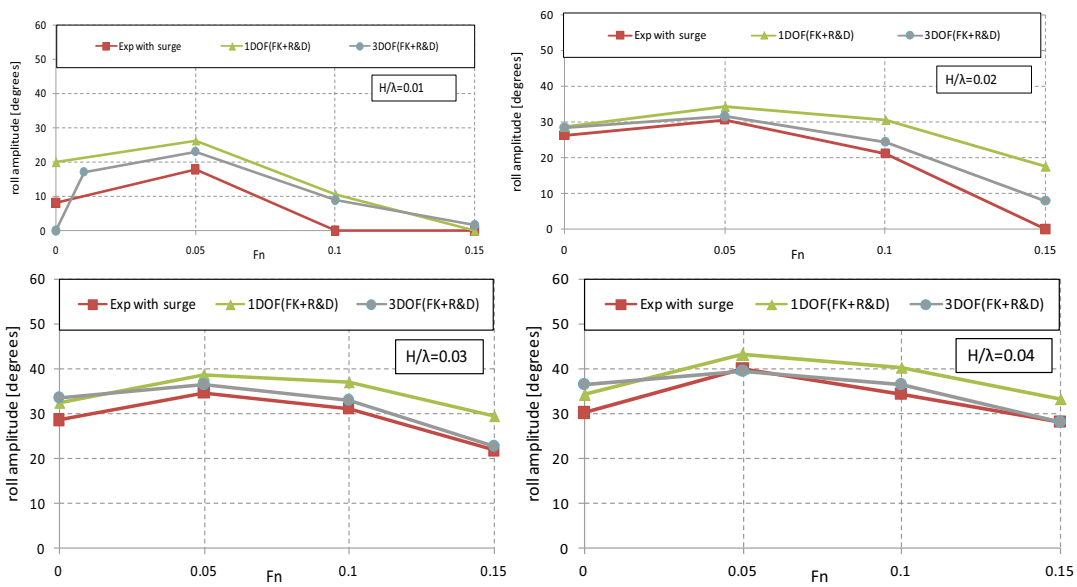
the results in experiments than that with the 1 DOF (FK+R&D) approach.

Key Applications

Parametric rolling is one of the five stability failure modes in the second-generation intact stability criteria which include Level 1 and Level 2 vulnerability criteria and direct stability assessment. The main purpose of these criteria is to enable the use of the latest numerical simulation techniques for evaluating the safety level of a ship from an intact stability viewpoint (IMO, Msc.1/Circ.1627 2020).



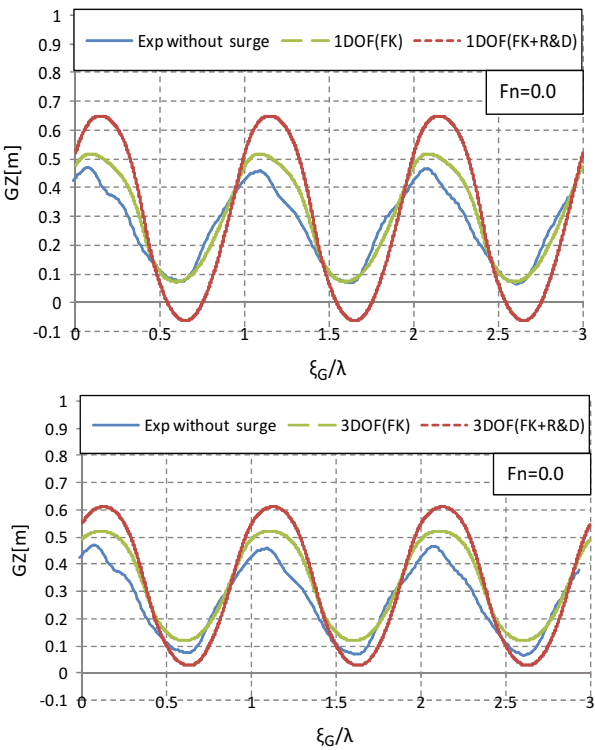
Parametric Rolling, Fig. 15 Comparisons of parametric roll as function of Froude number in experiments, 1 DOF approach and 3 DOF approach with $\lambda/L_{pp}=1.0$, $\chi=180^\circ$



Parametric Rolling, Fig. 16 Comparisons of parametric roll as function of Froude number in experiments, 1 DOF approach and 3 DOF approach with $\lambda/L_{pp}=1.0$, $\chi=180^\circ$

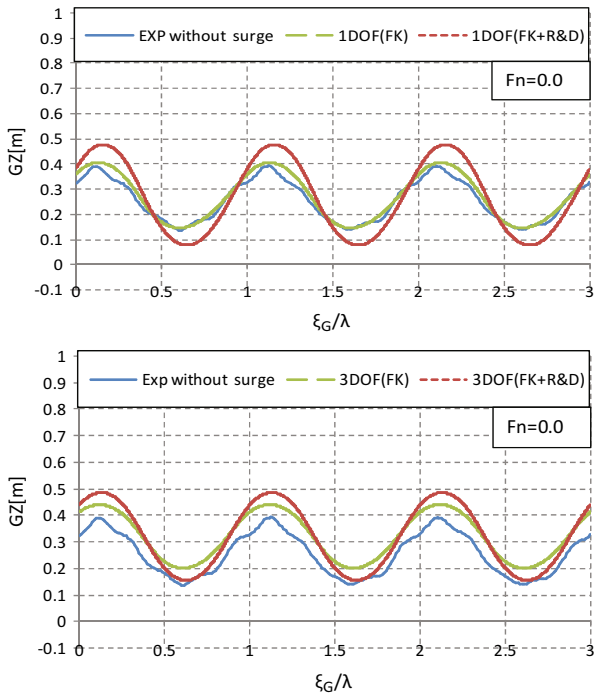
Parametric Rolling,

Fig. 17 Comparisons of roll-restoring variation in experiments and simulations with $\lambda/L_{pp}=1.0$, $H/\lambda=0.02$, $\chi=180^\circ$, and heeling angle 7.3 degrees



Parametric Rolling,

Fig. 18 Comparisons of roll-restoring variation in experiments and simulations with $\lambda/L_{pp}=1.0$, $H/\lambda=0.01$, $\chi=180^\circ$, and heeling angle 7.3 degrees



Parametric rolling is a complicated phenomenon when the ship is sailing in the seas, and the direct stability assessment is related to complex hydrodynamics. The above introduction on parametric rolling could be useful for shipbuilders, shipmasters, shipowners, ship operators, and shipping companies to understand parametric rolling.

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Parametrically Excited Roll

- [Parametric Rolling](#)

Passive Sonar

- [Sonar Technology](#)

PDU, Power Distribution Unit

- [Power Transmission and Distribution](#)

Phase Shift Keying (PSK)

- [Underwater Acoustic Communication](#)

Photo Multiplier Tube (PMT)

- [Photoelectric Detection Technology in Underwater Vehicles](#)

Photoelectric Detection Technology in Underwater Vehicles

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Tianjin, China

Synonyms

[Laser carrier intensity modulation \(LCIM\)](#); [Photo multiplier tube \(PMT\)](#); [Photoelectric imaging](#); [Polarization imaging](#); [Range-gated imaging](#); [Underwater device](#); [Underwater target](#)

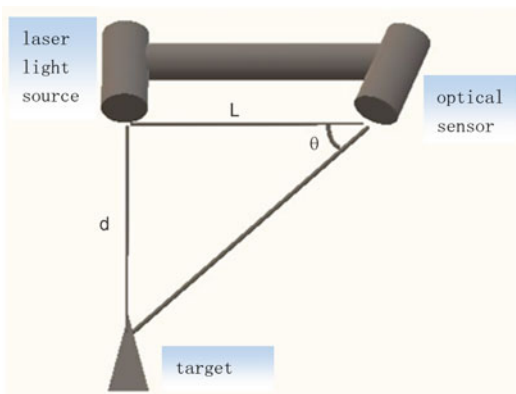
Definition

Underwater photoelectric detection technology is used for target detecting and imaging in the environment of underwater, using electro-optic detection method. Compared with acoustic detection techniques, photoelectric detection has the advantages of high resolution ratio, no near-field blind area, and fast and convenient for target identification. At present, the underwater photoelectric detection methods include range-gated imaging detection, laser line scanning imaging detection, polarization imaging detection, modulation/demodulation imaging detection, structured light imaging detection, etc. (Cao et al. 2011).

Scientific Fundamentals

Compared with land photoelectric imaging system, underwater photoelectric imaging and detection system work in special environment. There are mainly three parts in underwater photoelectric imaging and detection system, including underwater lighting system, underwater imaging lens, and detector (Yuan 2013).

Figure 1 shows the working principle. In Fig. 1, laser light source lights the underwater target, and the light reflected by the target enters the optical sensor through the aquatic medium and absorbed scattered by the water at the same time.



Photoelectric Detection Technology in Underwater Vehicles, Fig. 1 Underwater photoelectric imaging and detection system working principle

There are target light, scattering light and back-light showing on the image plane. Target image focuses on the plane of the optical sensor. The photoelectric device takes target image and sends it to the recording device after signal processing. The optical sensor's imaging quality can be affected by the absorption and scattering of light from the water.

Difficulties of Underwater Photoelectric Detection

Since 1963, S.Q. Duntley et al. found that the attenuation of blue and green light in 0.45 μm –0.55 μm band by sea water is much smaller than that in other bands (Zhao et al. 2014). The detection of underwater targets and environments such as oceans and lakes by laser irradiation has attracted more and more attention in the world and has also opened up a new way of underwater target detection. Due to the particularity of underwater environment, the key problems of underwater target detection are two aspects: one is how to suppress the backscattering effect of suspended particles in water and improve the imaging contrast; the other is how to overcome the attenuation effect of water on laser energy.

Attenuation Characteristics of Light in Water

Underwater working system and terrestrial imaging system are different, mainly because of the influence of water on the characteristics of light. The absorption and scattering of water to light are the main factors affecting the quality of underwater imaging. The attenuation characteristic of light in water refers to that the energy of light decreases exponentially in the course of transmission due to the absorption and scattering of water, which limits the propagation distance of light in water. And the scattering of water to light will result in the low contrast of the underwater image. If the light is only affected by the absorption of water in the process of transmission, then the absorption of water can be overcome by increasing the power of the light source. Thus, the imaging distance can be increased. However, due to the existence of scattering, increasing the power of the light source,

the power of the target-reflected light and scattered light are increasing at the same time. Then the image contrast obtained by the photoelectric receiver will not be improved. So simply increasing the power of the light source cannot improve the underwater imaging distance. It has been found that the underwater imaging distance is generally about more than 10 m, which is an urgent problem for the underwater optical imaging system, and also the main reason for the application limitation of the underwater optical imaging system.

Absorption of Light by Water

The absorption selectivity of water to light is obvious. The absorption rate of water is different in different spectral regions. The absorption of water to ultraviolet and infrared bands is stronger, but the absorption to blue-green bands is weaker. In the visible region, absorption of water is the largest in red, yellow, and light-green spectral regions. So color distortion will occur after transmitting light a certain distance underwater. When the underwater imaging distance is relatively long, the image is usually gray. In the water with low turbidity, the absorption in the blue-green band is the smallest, which is the so-called "blue-green window" of underwater imaging. But even in this blue-green window, the underwater transmission of light is greatly affected by the absorption of water. After a 1-m underwater transmission, the attenuation of light intensity reaches at least 4%. In other spectral regions, light is almost completely absorbed several meters away. So even in the blue-green window, the absorption of water will make the imaging distance very limited. Therefore, the influence of water absorption characteristics on the underwater imaging system must be fully considered in the research and analysis of the operating range of the underwater imaging system (Chen et al. 2011).

Light Scattering Characteristics of Water

Light scattering by water means that when light propagates in water, the direction of light propagation will change due to the action of various medium particles in water. The scattering of sea water is more complex than that of the

atmosphere. There are mainly two types of scattering: the scattering caused by pure water and the scattering caused by suspended particles. These include Rayleigh scattering, Mie scattering and the scattering caused by the refraction of transparent materials. The scattering functions of different sea areas and different water types are quite distinguished. Clean sea water is mainly scattered by water molecules, while turbid water along the coast is mainly scattered by large particles. The scattering mode can be divided into forward scattering and backward scattering according to the direction of scattered light. The scattering of water and particles to light is mainly forward scattering, which generally accounts for more than 90% of the total scattering, while backscattering only accounts for a small part. Forward scattering and backward scattering will affect the quality of the underwater imaging image. Although backscattering accounts for a small proportion, in underwater laser imaging applications backscattering affects image contrast and is the main factor affecting image quality.

Underwater Photoelectric Detection Technology

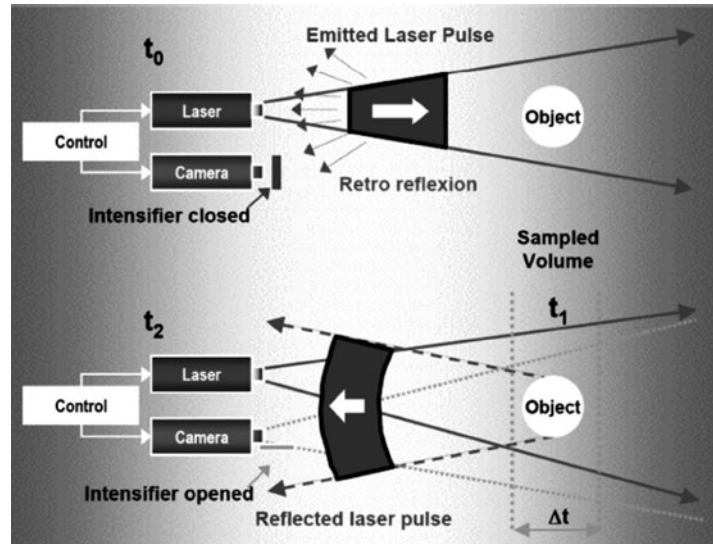
Range-Gated Imaging Detection Technology

Instead of using continuous laser, range gating technology uses the pulsed laser to emit the laser light and illuminate the target. The receiving end uses range-gated selection. When the target light reaches the receiver, the gate opens, so that the receiver receives the reflected light from all the scenery in the whole field of view. Before that the gating door has closed, thus eliminating the arrival of the background light during other periods. Its working principle is shown in Fig. 2.

The laser illuminates the target actively and then is reflected by the target. Before the light wave reaches the detection receiver, the gate is always in a "closed" state. Only within the very short time when the reflected light of the target reaches the receiver, the gate opens, and the reflected light enters the receiver for imaging. Thus, the target light and scattered light are separated from each other in time, and most of the forward and

Photoelectric Detection Technology in Underwater Vehicles,

Fig. 2 Principle diagram of range-gating synchronization control technology (Jin et al. 2011)



backward scattered light of the laser is eliminated from the optical receiver. And this reduces the background noise formed by the scattered light on the detector and improves the signal-to-noise ratio and contrast of the target image.

The laser underwater imaging system, which employs range-gated technology, uses the pulsed laser as the light source. In the process of light transmission underwater, the gate in front of the optical receiver is always closed and only opens at the time when the target-reflected light reaches the detector. At this moment, the receiver receives the reflected light from all the scenes in the whole field of view, including scattered light, target-reflected light, background light, etc. Therefore, the range-gated underwater laser imaging system can eliminate most of the backscattered light, but it cannot be completely eliminated. Moreover, the observation field of the laser underwater imaging system using range-gated technology is narrow. To improve the underwater imaging distance and make the laser irradiate the whole field of view at the same time, the high-energy laser pulse is used to focus on a relatively small area, so that the image observed under the condition of narrow field of view will be clear.

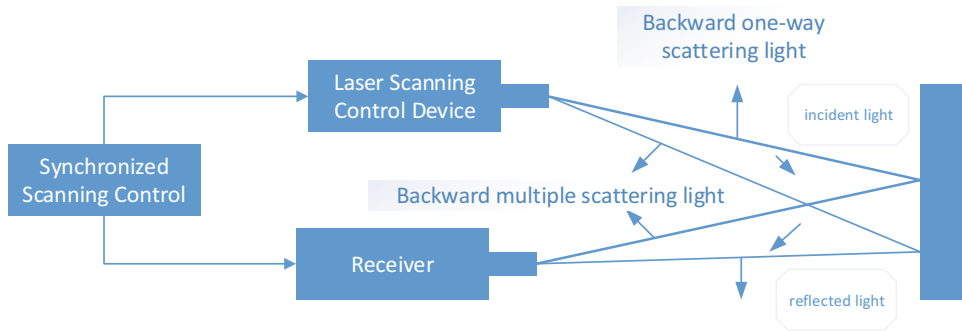
The performance of the range-gated imaging system depends on the gated pulse width and laser pulse width. If both are very narrow, only the light near the target can enter the receiver and the

scattered light and background light will be suppressed. This is conducive to the improvement of imaging quality.

Laser Line Scanning Imaging Detection Technology

Synchronized scanning technology is to synchronize the scanning beam and the receiving line of sight. The imaging technology utilizes the principle that the backscattered light intensity of water decreases rapidly relative to the central axis. It can suppress the backscattered light from space into the receiver, thus overcoming the influence of backscattered light and improving the imaging quality. The image shown in Fig. 3 is a typical optical path diagram of the system.

The scanning light and the receiving line of sight are synchronized. The laser and receiver are separated in space. The laser emits continuous narrow beams, and the receiver uses narrow field of view. There is only a small overlap between the two fields of view. The overlap area between illuminated water and receiver is minimized to reduce the scattered light entering the receiver and does not affect the target-reflected light entering the receiver. This effectively reduces the background noise without reducing the reflected light energy, thus effectively improving the imaging distance and image quality. Using synchronous scanning technology, point-by-point detection is



Photoelectric Detection Technology in Underwater Vehicles, Fig. 3 Principle of laser synchronous scanning imaging

used to form images. This method can scan a wide area up to 120° , and the effective field of view can reach 70° . The key to the image quality of synchronous scanning imaging technology lies in the tracking and receiving of highly sensitive detectors in the small field of view. The key to the realization of synchronous scanning technology lies in the synchronous control of the scanning beam and receiving the line of sight. The synchronous control methods are divided into mechanical synchronization and signal synchronization. Mechanical synchronization is used often.

Principle of Polarization Imaging Detection

Each object has a depolarization effect. The depolarization degree of sea water and object to light is different. If polarized light is used to irradiate underwater objects near Lambertine, the approximate depolarization of the object can be obtained according to the scattering theory. According to the knowledge of scattering, different objects have different scattering rates to the same light source. For objects approximate Lambertine, the scattering rate is generally less than that of backscattering light of particles in water. The depolarization of reflected light of objects is different from that of scattering light of particles in water. Therefore, according to the difference of depolarization between the backscattering light of particles and the object reflecting light in sea water, the backscattered light of particles can be suppressed by adjusting the angle of the polarizer in front of

the receiver, and the reflecting light of the object passes through the analyzer to enter the imaging system. Although the energy of backscattered light and target-reflected light are reduced by polarization technology, the degree of their reduction is different, and the backscattered light is reduced more, so the contrast of the image can be improved. There are two kinds of polarization technology: circular polarization technology and linear polarization technology. Both circular polarization technology and linear polarization technology can overcome backscattering to a certain extent and improve the contrast and imaging distance of underwater objects (Dubreuil et al. 2013).

Line Polarization Technology

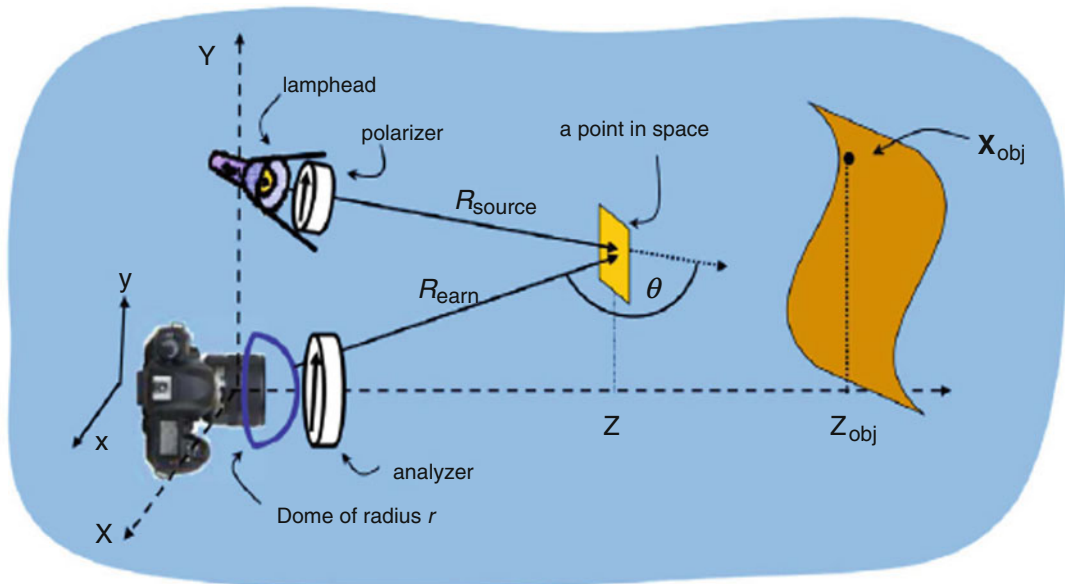
When the polarization direction of the illuminated target light (laser) and the polarization direction of the detector's front-line polarizer are both horizontal, the reflected light energy and scattered light energy of the object are approximately equal. And the contrast is too small to recognize the image. When the polarization direction of the linear polarizer is perpendicular to the polarization direction of the light source, the reflected light energy received by the detector is much larger than the scattered light energy. So the contrast of the image is related to the angle between the polarization direction of the linear polarizer and the polarization direction of the light source. When the angle is 0° , the image is blurred, and when the angle is 90° , the contrast is maximum, the image is clear.

Circular Polarization Technology

When the illumination source is right-handed circularly polarized light, the scattered light scattered by particles in water becomes left-handed circularly polarized light, that is, the scattered light becomes left-handed circularly polarized light. When the circular polarizer in front of the detector is right-handed, the backward scattered light of left-handed circularly polarized will be filtered out, and the incident right-handed circularly polarized light will be filtered out by the object. After reflection, the left-circularly polarized light and the right-circularly polarized light account for half of each, and the left-circularly polarized light will be filtered out by the right-circular polarizer in front of the detector. Only the right-circularly polarized light with the same polarization state can enter the imaging system through the polarizer for imaging, that is, the reflected light of the target entering the the detector is weakened and then backscattered light is basically eliminated. Using circular polarization imaging technology can reduce both backscattered light and target-reflected light entering the detector, but in different proportions. Most backscattered light is filtered out and most target-reflected light can

reach the detector, that is, the ratio of target-reflected light to backscattered light is improved, and the image quality formed on the detector is improved. To improve, the circular polarization technology can greatly improve the image sharpness.

The construction of the underwater polarization imaging system is simple and easy. Only a little modification is needed for ordinary underwater imaging instruments. Two polarizers are prepared. One polarizes the light source and the other analyzes the detected light signal. The light source modulated by polarizer emits fully polarized light, which illuminates the scene through the transmission of medium, and some light is scattered back to the detector by the medium particles. The sketch of underwater polarization imaging is shown in Fig. 4. The underwater polarization imaging system is connected with a circular polarizer on the outside of the digital camera and illuminated by a wide field of view polarization light. The analyzer detector is placed in front of the receiver. And two polarization images, which are perpendicular to each other, are collected. So the contrast and color of the underwater polarization image can be



Photoelectric Detection Technology in Underwater Vehicles, Fig. 4 Principle of underwater polarization imaging detection (Jin et al. 2011)

corrected. The contrast and color of the scene can be greatly improved.

Underwater Imaging Technology of Streak Tube

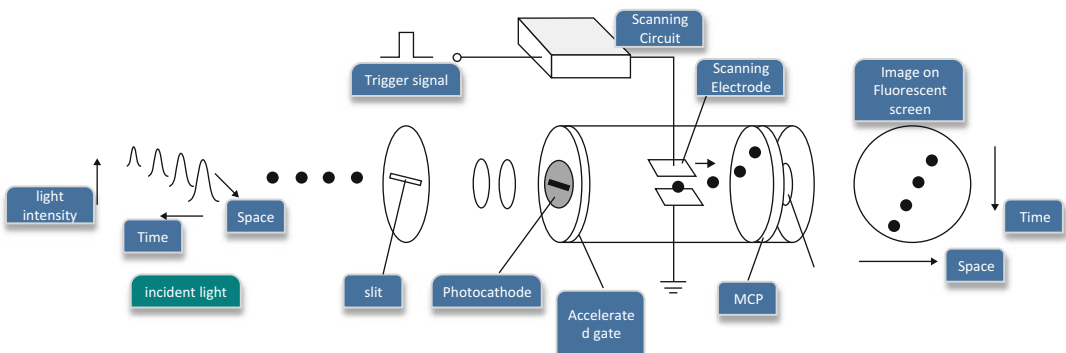
The principle of the streak tube imaging system is shown in Fig. 5. The receiving optical system imagines the incident light signal on the slit of the streak camera. The one-dimensional spatial light signal extracted from the slit is focused on the photocathode through the relay lens. The photoelectrons generated on the photocathode are accelerated by the accelerator and focused by the electrostatic focusing pole, and then enter the deflection system (deflection plate). There is a slope voltage that changes linearly at any time on deflection system. The photoelectrons entering the deflection system at different times are affected by different deflection voltage. When the photoelectron beam is enhanced by the image intensifier and reaches the fringe tube screen, it will spread along the direction of the slit and form a clear and visible fringe image on the screen. After the image processing system analysis, two-dimensional information of incident light signal can be obtained. That is, one-dimensional information of light intensity along slit direction changing with space and one-dimensional information of light intensity perpendicular to slit direction changing with time. A series of two-dimensional images of motion direction can be formed by sweeping with external mechanism, and three-dimensional data cube can be obtained by reconstruction. From the three-

dimensional data cube, the distance image of the observation direction dimension and the contrast image of the motion direction dimension can be obtained directly.

Modulation/Demodulation Laser Underwater Imaging Technology

Laser carrier intensity modulation (LCIM) technology is a feasible method for underwater range enhancement using coherent imaging detection. Early underwater coherent detection experiments usually used amplitude-modulated continuous laser to illuminate the target. And PMT was used to receive backscattered light and target-reflected light signals, and then the scattered light noise was partially removed by demodulating the amplitude modulated signal. Thus the detection distance could be extended to the detector scattered noise limitation. Subcarrier coherent detection technology is used to separate scattered light noise and reflected light signal in the time domain, thus generating target profile image or distance information.

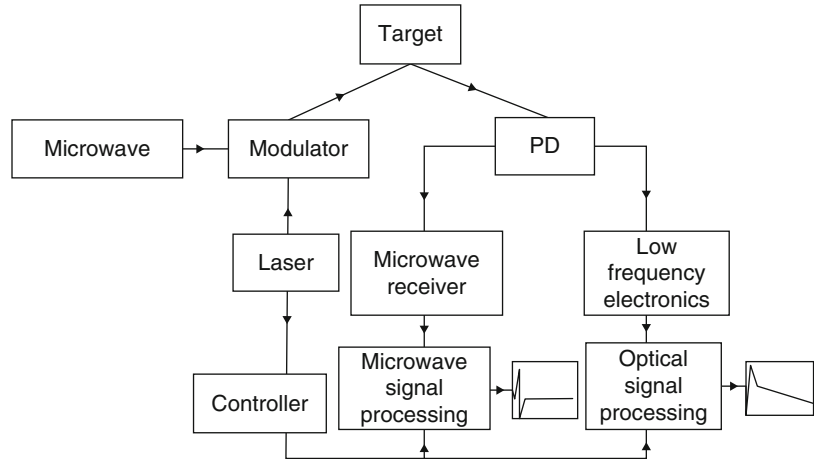
As shown in Fig. 6, a high-speed modulator is used to load the microwave carrier onto the optical pulse, so that the laser pulse can carry microwave signals underwater. After reflection from the underwater target, it is accepted by a high-speed photoelectric detector and a carrier-modulated Lidar system is formed. The backscattering signal of sea water is formed by the superposition of reflected light from a large number of scatterers in water. Due to the random distribution of scatterers, the phase delay of each reflection is



Photoelectric Detection Technology in Underwater Vehicles, Fig. 5 Principle of streak tube imaging

Photoelectric Detection Technology in Underwater Vehicles,

Fig. 6 Block diagram of the modulated Lidar on optical carrier system (Zhang et al. 2011)



uncertain and the coherence is lost. In this case, after the superposition, the information of back-scattering at the modulation frequency is largely lost and transformed into a low frequency signal. The phase information of the modulation frequency is not lost because of the coherence of the target reflection, and the high frequency characteristic is maintained. So at the receiving end, the received signal of the system is output by the photoelectric detector, which is divided into two channels. One path passes through a low-pass filter to filter out the signal retaining modulation characteristics. And the other passes through a band-pass filter, whose central frequency is equal to the modulation frequency and whose bandwidth is very narrow, to filter out the scattering noise. The traditional Lidar signal and the modulated pulse signal can be detected on two independent channels of the digital oscilloscope. Finally, the detection data are analyzed by the computer.

Design of Underwater Photoelectric Detection System

The underwater imaging system consists of the underwater lighting system, the underwater imaging system, the data transmission system, the storage and display system, etc. (Shirai et al. 2000).

Underwater Lighting System

It includes the underwater lighting source, transmitting optical system, lighting source mounting bracket, underwater lighting power supply, and controller. Its function is to illuminate underwater targets to meet the need for underwater imaging for light energy.

Underwater Imaging System

It includes underwater high-speed imaging system, camera power supply, zoom focusing and focusing controller, camera controller, fixed and waterproof seals, and other structures. It is mainly responsible for the photoelectric conversion of the target and clear imaging.

Data Transmission System

It includes power supply, control and signal transmission, and so on. The Cam Link interface is converted into optical fiber relay transmission mode to realize long-distance transmission. The waterproof and sealed structure is used for power supply, control, and signal transmission.

Data Storage Display Processing System

It includes host computer, corresponding software and image recording equipment (acquisition card) mainly completes the functions of image storage and playback and can convert data stream files into multiple formats for analysis and preservation after the event.

Key Applications

In the military field, the underwater photoelectric detection system can be installed on underwater carriers such as submarines, lightning extinguishers, underwater robots, etc. It can be used for underwater target investigation, detection, and recognition, and can be used for mine detection, submarine detection, anti-submarine detection, submarine navigation, and collision avoidance. In the civil field, the underwater photoelectric detection system can be used for underwater engineering installation and maintenance, underwater environmental monitoring, lifesaving salvage, submarine geomorphological exploration, oil exploration drilling location determination, biological research, and other marine development.

Cross-References

- [Photoelectric Imaging](#)
- [Polarization Imaging](#)
- [Range-Gated Imaging](#)

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Photoelectric Imaging

- [Photoelectric Detection Technology in Underwater Vehicles](#)

Photonic Crystal Fiber (PCF)

- [Fiber Optic Hydrophone](#)

Physical Ice Management

- [Ice Management in Offshore Operations](#)

Physical Layer

- [Underwater Acoustic Sensor Network](#)

Physical Properties of Sea Ice

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Definition

Physical properties of sea ice are the fundamental nature of sea ice, including ice crystal structure, salinity, temperature, and density, which affect other properties of sea ice, such as mechanical, thermal, and electromagnetic properties. Sea ice is

a mixture of pure ice, brine, solid salt, and gas. Additionally, the relative contents of phase components change with ice temperature to keep phase equilibrium, which, in turn, affects other properties of sea ice. Therefore, ice porosity (the sum of the brine and gas volume fractions) is another important sea ice physical property. Sea ice can be considered as a medium between atmosphere and upper ocean, which cuts off the direct interaction between the latter two. The changes in the physical properties of sea ice impact the state of atmosphere and upper ocean. Energy exchanges among atmosphere, sea ice and upper ocean in forms of radiation and conduction. Mass exchange also occurs between sea ice and upper ocean such as the rejection of salt from ice into sea water when ice freezes. Sea ice physical properties have a considerable effect on these exchanges.

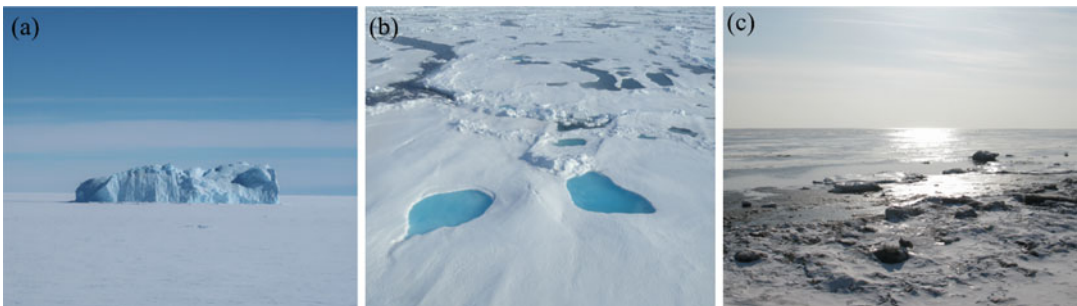
Sea ice exists in the polar regions perennially and further extends to subpolar sea areas in winter (Fig. 1). The production activities such as shipping and gas and oil exploration in the ice-covered water are affected significantly by sea ice. Structure in the ice zone must be designed to withstand extreme ice load. Sea ice mechanical properties are highly dependent on its physical properties. Given the vital function of sea ice on ocean engineering and global climate, a large number of field investigations have been carried out since the middle of the last century to study sea ice physical properties and their effect on ice mechanics. While most of them are conducted when air temperature is relatively high in late spring or summer

due to logistics, only a few of them have been carried out in winter or over a complete cycle.

Scientific Fundamentals

Sea Ice Growth and Crystal Structure

Generally, the seawater where sea ice forms has salinity more than 24.7 ppt leading to the freezing point being higher than the temperature corresponding to maximum density. Therefore, surface cooling of sea water produces an unstable vertical density profile in the upper water which further creates convective mixing. After the initial formation of continuous ice skim, the underlying ocean is isolated from the cold air so the subsequent ice growth is dominated by thermal conduction. Further crystallization occurs by direct freezing of seawater to the underside of the ice, which is called congelation growth. The above process is under calm conditions, while wind- or wave-induced turbulences in the open water areas can cause the ice skim over sea surface rafting and advection. The different grow histories result in varied grain textures. The observation of ice crystal texture is usually conducted by observing thin ice slices under crossed polarized light and photographed. An ice core is cut into several sections with thickness of a few centimeters. These sections are then attached to glass plates and thinned to a millimeter or less using a microtome. The preparation of ice slice should be cautious and is carried out under a low ambient temperature.



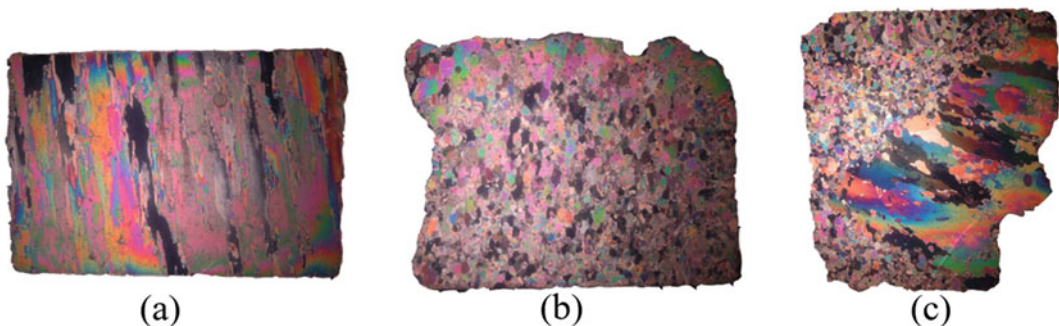
Physical Properties of Sea Ice, Fig. 1 Typical sea ice in the (a) Antarctic, (b) Arctic, and (c) subpolar region (Bohai Sea)

The most common ice crystal structures are granular and columnar grains (Fig. 2a, b). The granular grain usually has a sphere shape with size at the scale of $<10^{-2}$ m. Granular grain is usually associated with the formation of frazil ice which is mainly formed under a turbulent freezing condition. Four mechanisms have been identified responsible for turbulence (Jeffries et al. 1994). The first mechanism is the turbulence caused by wave and wind over open water. The second mechanism is double diffusion because of contact between two water masses with different salinity but both at their freezing point. The third mechanism is adiabatic decompression and resultant supercooling of water flowing from below ice shelves. The final mechanism is thermohaline convection initiated by surface cooling and freezing. In addition, frazil ice can also form by deformation processes. Some related processes are recognized (Lange and Eicken 1991). For example, newly formed frazil can appear at arbitrary places within ice because of ridging deformations. Rafting events can deposit newly formed frazil ice layer on the surface of lead underneath existing floes. Snow ice can also contribute the content of granular texture. Snow ice forms due to a heavy snow load submerging the ice surface below sea level, and sea-water floods and refreezes at the snow/ice interface. Snow ice is a common phenomenon in the Antarctic, but not very widespread in the Arctic. Recently, due to the thinning of Arctic sea ice and increase in total annual precipitation, the potential for snow ice is possibly increased. In

the Arctic, snow ice has a strong regional variability, and the largest potential is found in the Eurasian sector of the Arctic Ocean. Granular ice is typical isotropic of ice mechanics, i.e., the mechanical strength of granular ice does not depend on the loading direction. Columnar ice usually shows long vertically elongated crystals which are the result of thermodynamic congelation growth. Columnar grain has a larger size at a magnitude of $>10^{-2}$ m. Converse to the isotropy of granular ice, columnar ice usually shows anisotropy in ice mechanics. Such as there is an obvious difference between the uniaxial compressive strength in horizontal and vertical loading directions to ice surface, and the vertical strength is greater than horizontal strength. In the Arctic, first-year ice is often characterized as a thin granular ice layer at the surface underlain by a large fraction of columnar ice layer. While in the Antarctic, the ration between granular and columnar ice is reversed so that the granular ice accounts for more than columnar ice. While for multiyear ice, the crystal structure is much more complicated, which always shows discontinuities in crystal structure related to the changes in the size and shape of crystals. Apart from the common granular and columnar grains, there are also some other grains such as intermediate granular/columnar, mixed granular/columnar, inclined grains, and platelet grains (Fig. 2c).

Sea Ice Salinity

Ice salinity is usually expressed as the salts (in gram) contained in per kilogram of sea ice. It

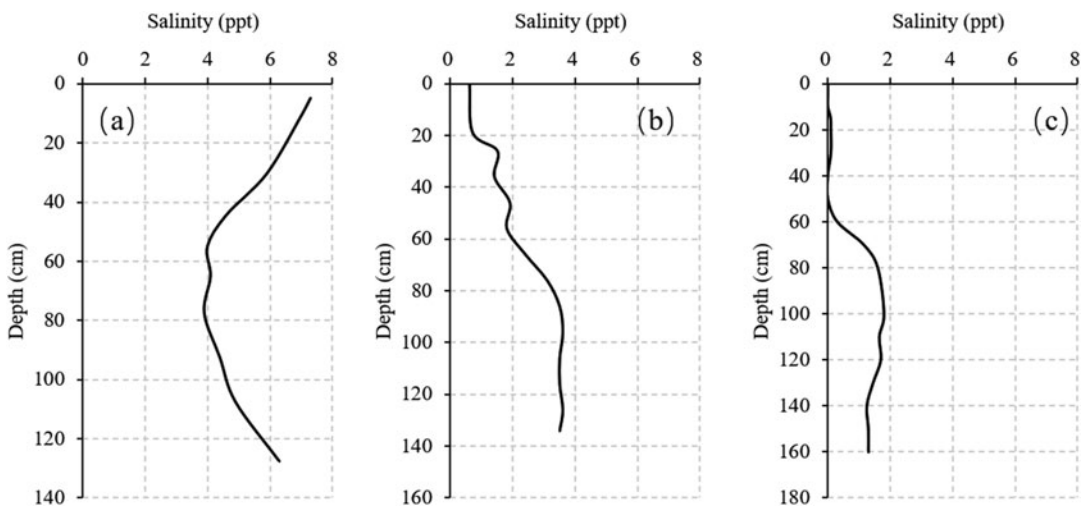


Physical Properties of Sea Ice, Fig. 2 Typical crystal structures of Arctic sea ice: (a) columnar, (b) granular, and (c) mixed columnar/granular grains

is quoted as a mass fraction in per thousand or practical salinity units (PSU). Ice salinity is usually measured by extracting an ice core, cutting the core into sections, and placing in containers to melt. Ice salinity is then obtained by measuring the electrolytical conductivity of melted samples. Quick operation is necessary because the loss of brine from an extracted ice core occurs as it is removed from ice cover, especially in the warm parts of a core. It is estimated that the salinity underestimation in the bottom layers of an ice core ranges from 1 to 20 ppt in bottom layers for growing sea ice using core-based measurements (Notz et al. 2005). For summer sea ice, the underestimation may be only a few tenths averagely, but it is the main contributor to the scatter of measurement (Eicken et al. 1995). Notz et al. (2005) have developed a nondestructive technique to measure ice salinity in field for growing sea ice based on the impedance measurements between individual thin horizontal platinum wires. However, this method cannot be used to measure salinity in preexisting ice, but must be deployed in an open water before freezing.

The salinity of sea ice is usually much lower than that of the seawater from which it has formed. Most of the salt is rejected by sea ice at the advancing ice-ocean interface as it forms. Such rejection can be described using an effective

distribution coefficient which is proportional to ice growth rate. The remaining salt within sea ice is not trapped into the ice lattice but becomes concentrated in interstitial liquid brine. After its initial formation, sea ice salinity varies during winter because of processes of brine diffusion, expulsion, and gravity drainage. Of these processes, gravity drainage is considered to dominate the salt loss from sea ice during winter. When an ice floe is cooled, it often has cold top layers and warm bottom layers, which can cause an unstable density profile. The brine pockets at the top with higher densities moving downward. But, this process is dependent of the permeability of the sea ice (e.g., the connectivity of brine channels). It is often assumed that sea ice is permeable for brine transport through connected brine channels as brine volume fraction is above 5–7%. With entering into melt season, the meltwater flushing caused by surface melt can drive brine out of sea ice, which further decreases the salinity. Therefore, for a first-year ice, the vertical salinity profile turns from a “C-shape” during grow season to a “?-shape” after the onset of melt during a cycle (Fig. 3a, b). Multiyear ice is much less saline and usually shows a nearly fresh surface layer (Fig. 3c) because large brine drainage has already occurred in previous seasons. The sea ice bulk salinity is linearly correlated to ice thickness. For



Physical Properties of Sea Ice, Fig. 3 Typical vertical salinity profiles of (a) growing first-year ice, (b) melting first-year ice, and (c) multiyear ice

cold ice, there is a pronounced decrease in bulk salinity with increasing ice thickness. While for warm ice, there is a positive relationship between bulk salinity and ice thickness.

Sea Ice Temperature

Ice temperature is affected by meteorological conditions such as air temperature and solar radiation. Snow cover over ice surface can also affect ice temperature acting as an insulation. The temporal variation of ice temperature can be investigated using thermistor strings. Core-based measurement is also usually carried out in field observations to measure ice temperature at a certain point of time. A general seasonal evolution of ice temperature is as follows: (1) a cold front propagates through the ice in the fall; (2) sea ice becomes cold and grows in the late fall, winter, and early spring; and (3) sea ice warms to melting point the next summer (Perovich and Elder 2001). The ice/ocean interface temperature is at around melting point. While the air/ice interface temperature varies with air temperature, solar radiation varies with long-wave radiation. Additionally, snow cover acts as an insulation for ice leading to a warmer ice temperature. There is a spatial variation for ice temperature with different conditions of ice and snow conditions. Typically, thinner ice and deep snow produce warmer snow/ice interface and internal ice temperatures, while thicker ice and shallow snow result in colder snow/ice interface and internal ice temperatures. The related contents of phase components change as ice temperature varies. The brine volume must decrease or increase to keep phase equilibrium with decreasing or increasing ice temperature. Gas bubbles are expected to form meanwhile. Therefore, ice mechanical properties have strong relationships with ice temperature. Sea ice mechanical strength decreases with increasing ice temperature.

Sea Ice Density

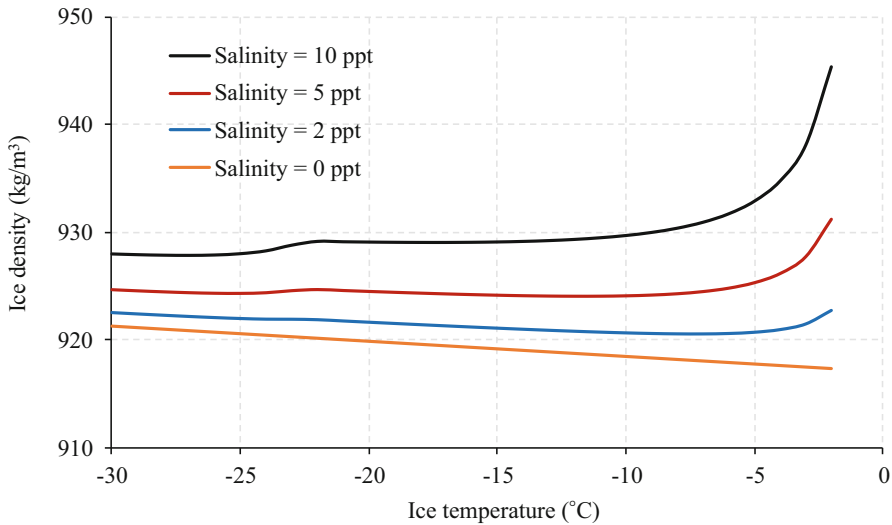
Sea ice is a multicomponent medium consisting of pure ice and inclusions such as brine solid salt, and gas. The density of sea ice is controlled by the densities and contents of the pure ice, brine, and solid salt as well as the air content in ice. Furthermore, the relative content of each component is

dependent on the ice temperature. There are mainly four different techniques to measure sea ice density including mass/volume, displacement (submersion), specific gravity, and freeboard and ice thickness methods. The advantage and disadvantage of each technique have been presented detailed in Timco and Frederking (1996). Of these techniques, mass/volume method is commonly used in most field measurements by measuring the mass and dimensions of ice samples. For the core-based density measurement, there are two sources of error. The first is because of the measurement uncertainty, and the second is the brine loss during core sampling. The former can be deduced through an error propagation analysis. An accurate identification of the error due to brine drainage is not easy. The underestimation is estimated to range from 5% to 20% (Hutchings et al. 2015; Pustogvar and Kulyakhtin 2016).

Sea ice density has been reported over a wide range from 720 kg/m^3 to 940 kg/m^3 , with an average of approximately 910 kg/m^3 (Timco and Frederking 1996). This wide range is caused by, on the one hand, the multicomponent nature of sea ice, and in the other hand, the measurement uncertainty of dimensions. In general, density in the ice section above sea level is less than below sea level. This is because the drained brine pockets in the section above sea level can be replaced by air. For first-year ice, density ranges from 840 kg/m^3 to 910 kg/m^3 in the section above the waterline, and from 900 kg/m^3 to 940 kg/m^3 below the sea level. For multiyear sea ice, density ranges from 720 kg/m^3 to 910 kg/m^3 in the section above the waterline; while below the waterline, density has a same range as first-year ice. Sea ice density decreases with the increasing gas content. Timco and Frederking (1996) have calculated the density of gas-free ice with four different ice salinities (0, 2, 5, and 10 ppt) according to equations in Cox and Weeks (1983). It can be considered as the upper-limit value of ice density for an ice with specific temperature and salinity (Fig. 4).

Sea Ice Porosity

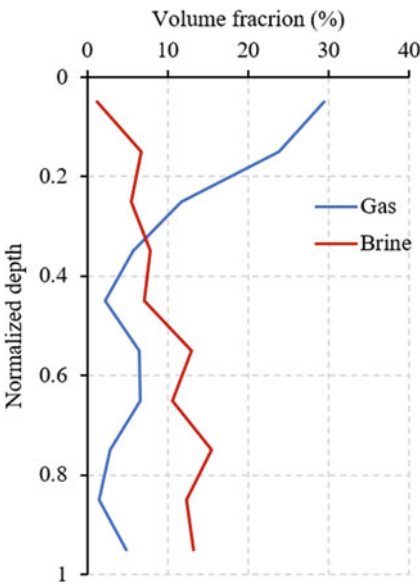
Porosity is defined as the sum of brine and gas volume fractions. Brine and gas inclusions are trapped when ice forms. Brine and gas volumes



Physical Properties of Sea Ice, Fig. 4 Calculated ice density versus ice temperature for four different salinities for gas-free sea ice in Timco and Frederking (1996)

vary with ice temperature. When ice is cooled, brine volume is reduced to keep phase equilibrium. Gas bubbles can also form by escaping from liquid brine because of decreased gas solubility. When ice warms, brine volume must be increased. At the same time, gas bubbles are expected to form in brine inclusions because lower density ice melting into higher density liquid forms a void within the inclusions. Brine and gas inclusions can be observed using the image process technique. Recently, computed tomography X-ray imaging method is employed to measure brine and gas inclusions. In addition, there are sets of mathematical formulae to calculate the volume fractions of brine and gas when the temperature, salinity, and density are known (Cox and Weeks 1983; Leppäranta and Manninen 1988). It is noteworthy that the calculation of gas volume fraction requires density with high accuracy. A small error of density measurement can produce a large error to calculated gas volume fraction.

Observations show that brine pockets are more vertically elongated and oriented than air bubbles. Brine pockets typically have mean major axis lengths of tenths of a millimeter, and air bubbles are much larger with mean major axis lengths of the order of millimeters. Measurements found that for multiyear ice, brine content ranges from 1% to



Physical Properties of Sea Ice, Fig. 5 Calculated brine and gas volume fractions against normalized ice depth for multiyear ice in the Arctic

15% by volume controlled by the thermodynamic equilibrium between the solid and liquid phases, and gas volume fraction decreases from >20% at the surface to <5% at the bottom (Eicken et al. 1995; Wang et al. 2020) (Fig. 5).

Applications

Because of the strong dependence of sea ice mechanical properties on its physical properties, the physical properties of sea ice are always used to determine the mechanical strength of sea ice. Several reliable formulae have been proposed between brine volume fraction or total porosity of sea ice and its mechanical parameters, such as tensile strength (Timco and Weeks 2010), shear strength (Frederking and Timco 1986), uniaxial compressive strength (Timco and Frederking 1990), and flexural strength (Timco and O'Brien 1994). It is then possible for a designer to estimate the ice strength parameters required by ice load evaluation based on these well-established relationships instead of performing mechanical tests.

In addition, sea ice physical properties, especially ice density, are directly involved in the ice load calculation for ice-structure interaction. For example, when an icebreaker ships in the level ice zone, its hull is covered by ice. Ice around the ship is lifted because of lighter density than water, which exerts additional resistance to the ship's hull. According to the commonly used ice resistance calculation method in Lindqvist (1989), ice density is a key input for determining the submerison component of total ice resistance. For ice interacting with a sloping structure, ice fragments or ice rubbles due to flexural failure of ice sheet will accumulate upon (for upward-slope structure) or below (for downward-slope structure) the slope. Accordingly, the weight or buoyancy of accumulated ice fragments exerts additional load on the structure. Therefore, ice density is involved in the calculation of ice load on sloping structures (ISO 2010).

For an ice-built structure, such as artificial grounded ice islands used for oil and gas exploratory drilling, it is important to monitor the ice temperature and density during construction (ISO 2010), as ice temperature and density largely affect its mechanical strength and thus the quality control of the entire program. The suggested measurement is mass/volume method for ice density and thermistor string for ice temperature. Ice strength decreases with increasing ice temperature. At high ice temperatures near melting

point, ice becomes soft and ductile; while at low ice temperatures well below melting point, it becomes strong and brittle. For an ice-built structure, it is usually used to support production equipment. Therefore, we must keep ice from warming up and make emergency measures before it warms. While, for an anti-ice structure, ice acts as the threat to structure safety, it is necessary to consider the ice strength at extreme low ice temperature during design.

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Phytoplankton Bloom

- [Harmful Algal Blooms](#)

Pigboat

- [Submarine](#)

Pile Anchor

- [Mooring Anchor](#)

Pile Capacity

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Synonyms

API – American Petroleum Institute; CPT – Cone Penetration test; DNV – Det Norske Veritas; ICP – Imperial College Pile; NGI – Norwegian Geotechnical Institute; UWA – University of Western Australia

Definition

The pile capacity refers to the maximum load that can be borne by a pile under load, to ensure that strength and stability of foundation soil and pile itself can be guaranteed and the deformation is within the allowable range.

Introduction

Pile foundation has been widely used in offshore engineering due to its high strength, large bearing capacity, and good durability (Kellezi and Hansen 2003). Like other foundation systems, strength and deformation are the two most important aspects that should be considered in design. That is, the pile foundation should be designed with high enough bearing capacity to carry the loads and without having excessive deformations at the same time. In this entry, bearing capacity of the pile foundation under both axial and lateral loads in cohesive and cohesionless soils is covered. The popular capacity design methods are introduced. The analytical methods for determining the soil reaction and pile performance under axial and lateral loading condition are presented in order to ensure the pile deformation, including the settlement, deflection, and rotation of the pile are within the acceptable serviceability limits and are compatible with the structural forces and movements.

Ultimate Axial Bearing Capacity in Cohesive Soils

Components of Axial Capacity

Pile is a typical deep foundation. The method to calculate the ultimate bearing capacity of deep foundation is thought to be suitable for estimation of bearing capacity of pile foundation in general.

The ultimate axial bearing capacity of deep foundation Q_u is generally composed of the end bearing capacity Q_b and the side friction resistance Q_s , that is:

$$Q_u = Q_b + Q_s \quad (1)$$

A_p is the gross end area of the pile,
 z is the depth below the original seabed.

The ultimate bearing capacity of the bottom of foundation is:

$$Q_b = q_u A_b \quad (2)$$

where

q_u is the unit bearing capacity of the bottom,
 A_b is the pile base area.

The ultimate shaft friction of the foundation is:

$$Q_s = f_s A_s \quad (3)$$

where

f_s is the unit shaft friction,
 A_s is the pile side surface area.

Several code methods are commonly used to calculate the pile axial capacity in cohesive soils, such as the API method (API 2000, 2008), the DNV method (DNVGL 2016), and the NGI method (Karlsrud et al. 2005; Karlsrud and Nadim 1990).

API Method

The API code is the specification of the American Petroleum Institute (API) for the planning, design, and construction of offshore fixed platforms.

The ultimate axial capacity of piles in compression Q_c in the API method is determined using the equation:

$$Q_c = Q_{f,c} + Q_p = f(z)A_s + qA_p \quad (4)$$

where

$Q_{f,c}$ is the shaft friction capacity in compression, in force units,
 Q_p is the end bearing capacity, in force units,
 $f(z)$ is the unit shaft friction, in stress units,
 A_s is the side surface area of pile,
 q is the unit end bearing at the pile tip, in stress units,

For open-ended pipe piles, the total end bearing capacity Q_p shall not exceed the sum of the end bearing capacity of the internal plug and the end bearing on the pile tip wall annulus. The weight and buoyancy of the pile soil-plug system should be considered when calculating the capacity of the pile foundation.

For pipe piles in cohesive soils, the unit shaft friction $f(z)$ at depth z can be calculated using the following equation:

$$f(z) = \alpha s_u \quad (5)$$

where

α is the dimensionless shaft friction factor, for clays,
 s_u is the undrained shear strength, in stress units.

The factor α can be computed as:

$$\alpha = 0.5\psi^{-0.5} \text{ for } \psi \leq 1.0 \quad (6)$$

$$\alpha = 0.5\psi^{-0.25} \text{ for } \psi > 1.0 \quad (7)$$

when $\alpha \leq 1.0$, $\psi = \frac{s_u}{p'_o(z)}$, where $p'_o(z)$ is the effective vertical stress at depth z .

For under-consolidated clays (clays with excess pore pressures undergoing active consolidation), α can usually be taken as 1.0.

The bearing capacity of unit pile end in clay can be calculated as:

$$q = 9s_u \quad (8)$$

The shaft friction, $f(z)$, acts on both inside and outside of the pile. The total axial resistance for pile compression is the sum of the external shaft friction, the end bearing on the pile wall annulus, and the total internal shaft friction or the end bearing of the plug, whichever is the lesser.

Details of the other code methods (DNV and NGI) can be referred to specific references.

Ultimate Axial Bearing Capacity in Cohesionless Soils

API Method

In the API method, the ultimate axial capacity of piles under compression is determined using the equation:

$$Q_c = Q_{f,c} + Q_p = fA_s + qA_p \quad (9)$$

For pipe piles in cohesionless soils, the unit shaft friction f is calculated using the equation:

$$f = Kp'_0 \tan \delta \quad (10)$$

where

K is the lateral earth pressure coefficient which is the ratio of horizontal to vertical effective stress,
 p'_0 is the effective overburden pressure at calculated points,
 δ is the external friction angle.

For unplugged open-ended driven pipe piles, it is usually assumed that the K values of tensile and compressive loads are all 0.8. For pile with soil plug or close-ended, the K value can be assumed to be 1.0. The capacity of open-ended driven pipe piles is usually lower than that of the closed-ended piles (Paik et al. 2003). If no further information is

available, Table 1 can be used to select the external friction angle. For long piles, f will not increase linearly with overlying pressure. In this case, restricted values of f are defined in Table 1.

The unit bearing capacity of pile end in cohesionless soil can be calculated as:

$$q = N_q p'_{0,tip} \quad (11)$$

where

$p'_{0,tip}$ is the effective overburden pressure at pile tip,
 N_q is bearing capacity coefficient, the values of which can be referred to Table 1.

CPT-Based Method

A simple method called CPT-based method is widely used for assessing pile capacity in cohesionless soils due to its better predictions of pile capacity and the advantages of simple, rapid, and relatively economic (Eslami and Fellenius 1997). This method is applicable for a wide range of cohesionless soils. The four commonly used CPT-based methods include the ICP method, UWA-05 method, Fugro method, and NGI method. More details of these methods can be found in the specific references (API 2011; Jardine et al. 2005; Clausen et al. 2005; Lehane et al. 2005a, b).

Pile Capacity, Table 1 Design parameters for cohesionless siliceous soil (API RP2A)

Density	Soil description	Soil-pile friction angle,	Limiting skin friction values	N_q	Limiting unit end bearing values,
		δ Degrees	kPa		MPa
Very loose	Sand	15	47.8	8	1.9
Loose	Sand-silt				
Medium	Silt				
Loose	Sand	20	67	12	2.9
Medium	Sand-silt				
Dense	Silt				
Medium	Sand	25	81.3	20	4.8
Dense	Sand-silt				
Dense	Sand	30	95.7	40	9.6
Very dense	Sand-silt				
Dense	Gravel	35	114.8	50	12
Very dense	Sand				

Soil Reaction for Piles Under Axial Load

The failure of the pile may be related to the bearing capacity, deformation, or settlement of the pile. Soil reaction is the result of interaction between the soil and the pile under load. It is necessary to analyze the soil reaction in order to ensure the pile settlement, deflection, and rotation of the pile are within the acceptable serviceability limits and are compatible with the structural forces and movements. The pile performance and deformation under both axial and lateral need to be predicted in design. The t - z curve is usually used to analyze the relationship between mobilized soil-pile shear transfer (t) and local pile displacement (z) at any depth. And Q - z curve is used to describe the relationship between mobilized end bearing resistance (Q) and axial pile tip displacement (z).

z is the local pile deflection,
 D is the pile diameter,
 t is the mobilized soil pile adhesion,
 $t_{\max}$ is the maximum soil pile adhesion or unit skin friction capacity.

The displacement value of 1% of the pile outer diameter (i.e., $z_{\text{peak}}/D = 0.01$) is generally employed to meet the design purposes. However,

Pile Capacity, Table 2 Data of the t - z curves for clays in API method (API RP2A)

Clays	z/D (in.)	$t/t_{\max}$ (lb/ft ²)
	0.0016	0.3
	0.0031	0.5
	0.0057	0.75
	0.008	0.9
	0.01	1
	0.02	0.70 to 0.90
	∞	0.70 to 0.90

Axial Load Transfer Analysis and (t - z) Curves

API Method

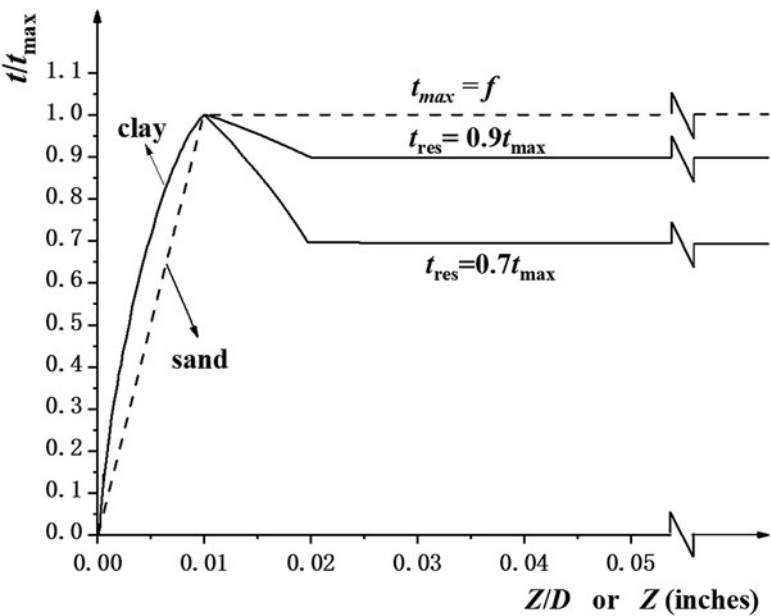
API method recommends a t - z curves for non-carbonate soils as shown in Fig. 1. The data to generate the curve are given in Tables 2 and 3.

where

Pile Capacity, Table 3 Data of the t - z curves for sands in API method (API RP2A)

Sands	z (in.)	$t/t_{\max}$ (lb/ft ²)
	0	0
	0.1	1
	∞	1

Pile Capacity,
Fig. 1 Typical axial pile load transfer–displacement (t - z) curves



there is great uncertainty for the case where the axial pile stiffness is critical for design. At this situation, it is feasible to take values of 0.25% to 2.0% of the pile diameter. It is necessary to pay attention to the shape of the t - z curve at displacements greater than $z_{\max}$ as shown in Fig. 1. Values of the residual adhesion ratio $t_{\text{res}}/t_{\max}$ at the axial pile displacement at which it occurs (z_{res}) are a function of soil stress-strain behavior, stress history, pipe installation method, pile load sequence, and other factors. Generally, the range of $t_{\text{res}}/t_{\max}$ is 0.70 to 0.90. Laboratory, in situ, or model pile tests can provide valuable information for determining values of $t_{\text{res}}/t_{\max}$ and z_{res} for various soils.

End Bearing Resistance-Displacement, Q - z , Curve

API Method

The ultimate bearing capacity of the pile foundation can be determined in accordance with the methods described in the previous sections. However, it usually requires a relatively large pile tip displacement to mobilize the full end bearing resistance. In sand and clay, a pile tip displacement as large as 10% of the pile diameter may be necessary for full mobilization. Definitive criteria and precise specifications are absent up to date.

API recommended a method as shown in Fig. 2 with data in Table 4 for both sand and clay. in which

z is the axial pile tip displacement,
 z_{peak} is the displacement to maximum soil pile adhesion or unit skin friction,
 Q is the mobilized end bearing capacity, in force unit,
 Q_p is the end bearing capacity, in force units,
 D is the diameter of the pile.

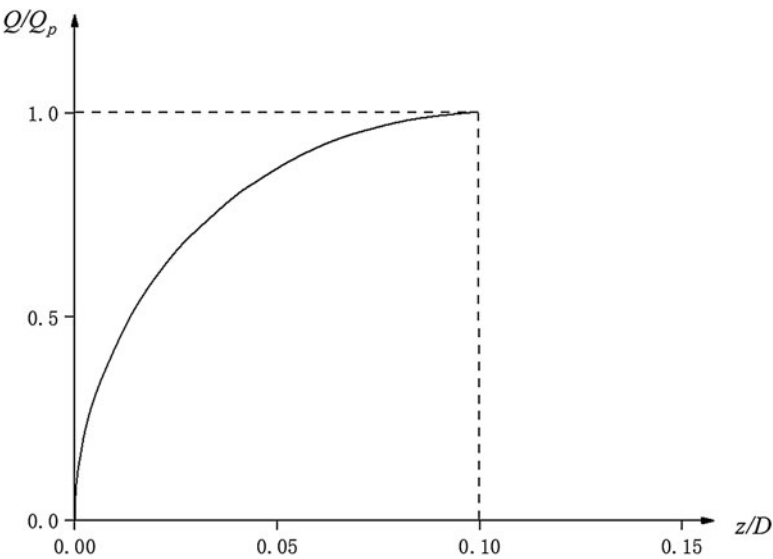
Lateral Bearing Capacity and Soil Response in Cohesive Soils

In addition to axial load, the lateral load also has an important influence on the pile foundation. For offshore piles, the significant lateral loads and moments are usually inevitable, especially for

Pile Capacity, Table 4 Data for pile end bearing capacity-displacement curve (API RP2A)

z/D	Q/Q_p
0	0
0.002	0.25
0.013	0.5
0.042	0.75
0.073	0.9
0.1	1
∞	1

Pile Capacity, Fig. 2 Pile end bearing capacity-displacement curve



foundation such as anchor piles and monopile supporting wind turbines and small platforms. Therefore, the lateral bearing capacity of the pile foundation and the lateral resistance of the soil near the surface should be considered in the design of offshore pile foundation (Gourvenec and Randolph 2011). The method commonly used to design piles under lateral and overturning loads is based on the Winkler modeling method, commonly known as the p-y method (Byrne et al. 2015).

Nonlinear load transfer analysis method can be used to analyze the capacity and response of laterally loaded piles using appropriate load transfer curves (p-y curves) for different soil types (Fonseca 2015).

The p-y curve method is recommended to calculate the horizontal bearing capacity of pile foundations in API and DNV (Jin 1993).

Lateral Capacity and Response for Soft Clay

The following equations are suggested to estimate the lateral capacity of the pile in soft clay:

$$p_u = 3s_u + \gamma X + Js_u X/D \quad (12)$$

$$p_u = 9s_u, \text{ if } X \geq X_R \quad (13)$$

where

p_u is the ultimate horizontal bearing capacity,
 s_u is the undrained shear strength of soil,
 D is the diameter of the pile,
 γ is the bulk density of soil,
 J is a dimensionless constant determined by field tests, with a range of 0.25–0.5,
 X is the depth below the seabed,
 X_R is the depth below seabed to bottom of reduced capacity zone for uniform soils.

For the soil of constant strength with depth, the following formula can be obtained:

$$X_R = \frac{6D}{\frac{\gamma D}{s_u} + J} \quad (14)$$

For the nonuniform soil of strength increase

with depth, X_R can be solved by drawing the curves of the two eqs. (12) and (13). The first intersection point of the two curves is X_R . This relation does not apply to situations where intensity changes are irregular. In general, the minimum value of X_R is about 2.5 times the diameter of a pile D .

The relationship between lateral bearing capacity and displacement of piles in soft clay is usually nonlinear. For short term static loads, the p-y curve can be generated according to the data in Table 5.

Unless further information is available, the initial slope of the p-y curve of clay can be calculated by the following equation:

$$k = \xi \frac{p_u}{D(\varepsilon_c)^{0.25}} \quad (15)$$

where

ξ is the empirical coefficient,
 ε_c is the vertical strain at one-half the maximum principal stress difference in a static undrained triaxial compression tests on an undisturbed soil sample.

For normally consolidated clay, $\xi = 10$ is recommended, and for over-consolidated clay, $\xi = 30$ is recommended.

where

p_u is the ultimate lateral capacity, unit of pressure,
 p is the mobilized lateral resistance, unit of pressure,
 y_c equals $2.5 \times \varepsilon_c \times D$,
 y is the local pile lateral displacement.

Pile Capacity, Table. 5 Mobilized lateral resistance – displacement data for short-term static loads (API RP2A)

p/p_u	y/y_c
0	0
0.23	0.1
0.33	0.3
0.5	1
0.72	3
1	8
1	∞

Lateral Capacity and Response for Stiff Clay

Hard clay has greater brittleness than soft clay, despite it has nonlinear stress-strain relationship (API RP2A). When establishing the stress-strain curves of hard clay and the p-y curves under cyclic loading, it is necessary to make a proper judgment on the rapid weakening of the bearing capacity of hard clay under large deformation.

Lateral Bearing Capacity and Soil Response in Cohesionless Soils

API Method

The ultimate lateral bearing capacity of pile in sand can be computed by the following equations. Eq. (16) is for the capacity of shallow soil, and Eq. (17) is for the deep soil. At the given depth, the ultimate bearing capacity should be the minimum value of the calculated p_u :

$$p_{us} = (C_1X + C_2X)\gamma'X \quad (16)$$

$$p_{ud} = C_3D\gamma'X \quad (17)$$

where

p_u is the ultimate resistance, units of pressure,

γ' is the submerged soil unit weight,

D is the pile outside diameter,

X is the depth,

C_1, C_2, C_3 are coefficients given in Fig. 3.

The lateral bearing capacity–displacement relationship (p-y) of sand is nonlinear. In the absence of reliable data, approximate values at any depth can be approximately determined according to the following equation:

$$p = Ap_u \tanh\left(\frac{kX}{Ap_u}y\right) \quad (18)$$

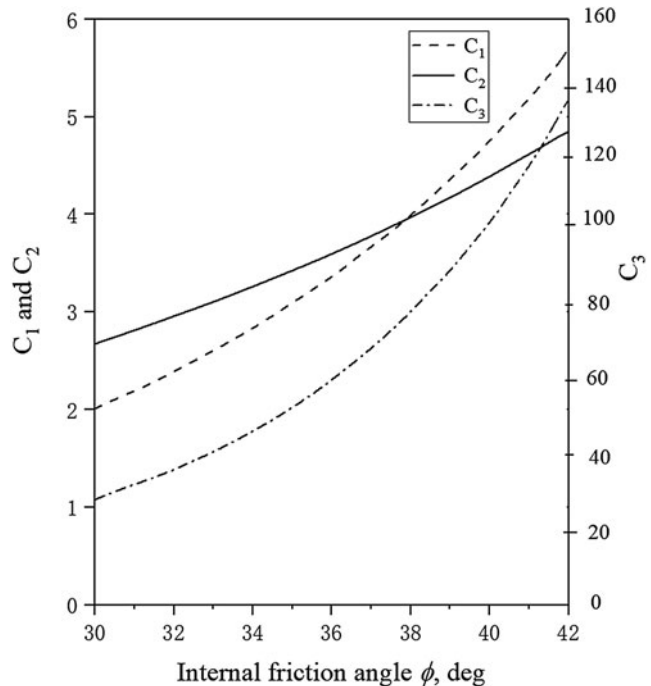
where

p is ultimate lateral bearing capacity at depth X ,
 k is the rate of increase with depth of initial modulus of subgrade reaction which can be got from Table. 6,

y is the lateral deflection at depth X ,

Pile Capacity,

Fig. 3 Coefficients C_1, C_2, C_3 , as function of ϕ in API method



Pile Capacity, Table 6 Rate of increase with depth of initial modulus of subgrade reaction k (API RP2A)

ϕ	k	
	MN/m ³	(lb/in ³)
25°	5.4	20
30°	11	40
35°	22	80
40°	45	165

X is the depth below the original seabed,

A is a factor to account for static or cyclic loading conditions as follows:

$$A = \begin{cases} 0.9 & \text{for cyclic loading} \\ \left(3 - 0.8 \frac{X}{D}\right) \geq 0.9 & \text{for static loading.} \end{cases} \quad (19)$$

Other Considerations

Pile foundation in offshore engineering is a complex system, from the installation process to loading stage. The contents of pile capacity in this entry only cover the methods for common and simple situations. Some methods, especially the code methods, are based on specific databases. Most of the databases may comprise piles of smaller length and diameter compared with piles in practical projects (Randolph et al. 2005). Some significant issues such as friction fatigue and progressive failure for long and large diameter pile may not be fully integrated in the methods. As a result, caution should be exercised when the selected methods are applied to some circumstances with special conditions. Another significant point is that cyclic loading is an important feature and has critical effects on the capacity of offshore pile foundations (Leblanc et al. 2010). But due to the complexity, cyclic loading is not covered here. Also, some other issues related to the capacity of foundation such as the pipe-pile plugging, pile in carbonate soils, grouted pile in rock, pile groups and so on are not included in this entry. Appropriate paper and books can be referred to address these specific issues.

Cross-References

- [Monopile Foundations in Offshore Wind Farm](#)
- [Offshore Pile Driving](#)
- [Shallow Foundations](#)
- [Short Pile and Uplifting Capacity](#)
- [Spatial Variability](#)
- [Suction Piles](#)

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Pipeline End Manifold

- [Shallow Foundations](#)

Pipeline End Manifold (PLEM)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Pipeline End Termination

- [Shallow Foundations](#)
- [Subsea Connector](#)

Pipeline Penetration

- [Pipeline Soil Interactions](#)

Pipeline Soil Interactions

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Synonyms

[Lateral buckling](#); [Pipeline penetration](#); [Pipe-soil interaction](#); [Steel catenary risers](#); [Submarine pipeline](#); [Touchdown point](#)

Definition

Submarine pipelines and risers are an important part of offshore oil and gas development engineering, and the seabed pipe-soil interaction is one of the key issues in the design of pipelines and risers. According to the laying state and operation state of pipelines and risers, the results show that the seabed as the foundation of supporting must provide sufficient support to ensure the stability of pipelines and risers. The research on pipe-soil interaction is to accurately predict the fatigue life of pipeline by describing the buried depth and internal force of pipeline in soil and correctly evaluating the effect of soil on pipeline resistance. This kind of research is of great significance to the stability, safety, and economy of pipelines and to the improvement of relevant design codes.

Seabed pipe-soil interaction affects various aspects of subsea pipelines and risers during installation and operation, which include the touchdown point (TDP) of the steel catenary risers (SCR) directions, bearing capacity of pipeline and lateral buckling, etc. In this chapter, the research status by scholars in recent years will be introduced in detail.

Soil Types and Classification

Soil parameters along the pipeline route are needed for modeling pipe-soil interaction, which can be obtained by carrying out geophysical and

geotechnical surveys. The soil can be classified by grain size or its plasticity and cohesion. Soils with particles larger than about 0.05 mm are called coarse grained (sands and gravels), while soils finer than this size are called fine grained (silts and clays).

Soil is a porous media material comprising solid particles with void spaces containing gas and water. In subsea environments, the water pressure is generally high enough that gas is forced into solution and the soils are saturated with water. Soil behavior depends on the rate at which load is applied to the soil, and generally can be categorized as “drained” or “undrained” behavior (Bai and Bai 2014). If the rate of load is greater than the rate at which pore water in the interparticle voids is able to move in or out of soil interparticle voids, then the soil is said to behave in an undrained manner. If the rate of loading is slower than the rate at which pore water is able to move in or out of soil interparticle voids, the soil is said to behave in a drained manner.

Dynamic Pipe-Soil Interaction at TDP

The use of SCR in deepwater developments is increasing, and most of the structural damage of SCR occurs in the so-called TDP where the pipe takes off and lands repeatedly on the soft soils generally encountered in deepwater (Fig. 1).

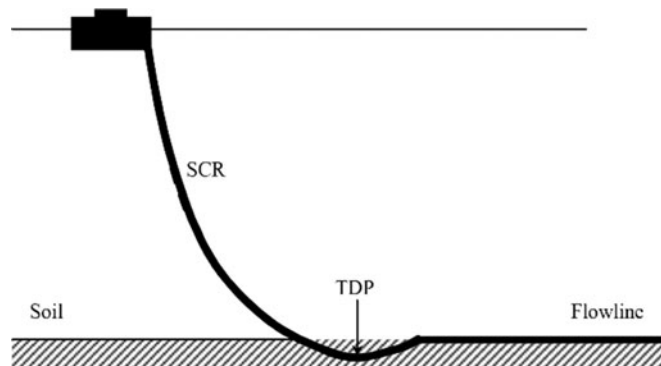
Different pipe-soil interaction models are used in TDP numerical analysis. The rationality and reliability of these models directly affect the accuracy and efficiency of calculation results. At

present, the most widely used pipe-soil nonlinear models are proposed by Aubeny and Biscontin (2009) and Randolph and Quiggin (2009), which are abbreviated as AB model and RQ model, respectively. Both of them are based on a large number of test results. AB model is mainly developed from the large-scale model test of Langford and Aubeny (2008), while RQ model is summarized from the centrifuge test results carried out by Hodder et al. (2008). Bridge summarized a nonlinear model based on the test results of STIDE and CARISIMA JIP. It was pointed out that the process of pipe-soil interaction can be divided into four stages: penetration, uplift, suction, and reintegration. RQ model refers to the full-scale field test results of Bridge et al. (2004). Different pipe-soil interaction models are used in numerical analysis. The rationality and reliability of these models directly affect the accuracy and efficiency of calculation results.

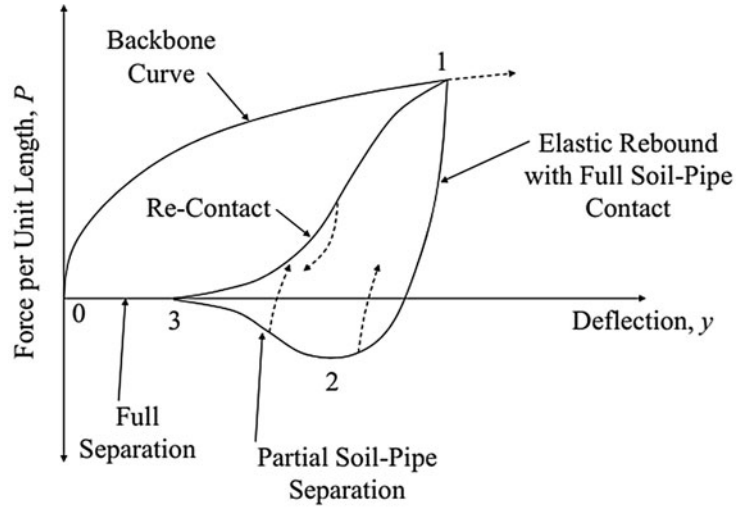
As shown in Fig. 2, there are four stages in AB model: initial penetration stage (power function backbone curve), upper pull-up stage (hyperbolic function elastic rebound curve), partial detachment stage (third order polynomial curve), complete detachment stage and repenetration stage (third order polynomial curve). The pipe-soil separation point in AB model can describe the formation of grooves under the pipeline, which is in accordance with the test results, but the effect of soil strength on cyclic weakening is not considered.

As shown in Fig. 3, RQ pipe-soil interaction model describes four different states of pipe-soil interaction: noncontact, initial penetration, uplift

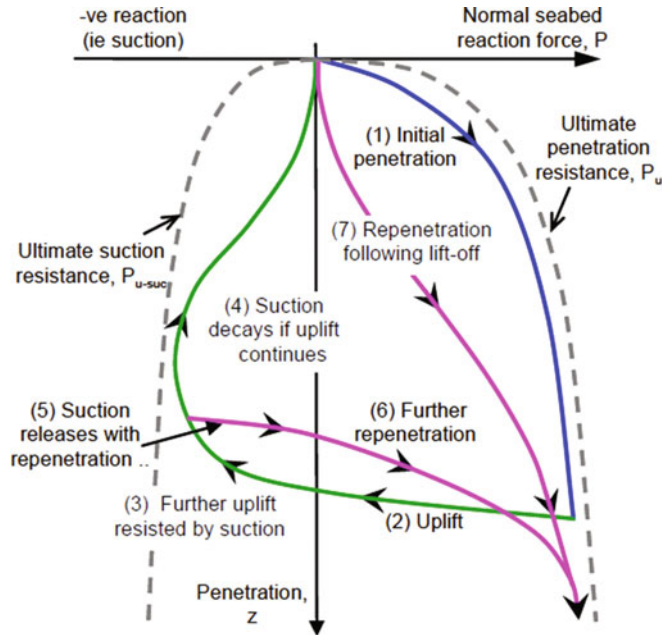
Pipeline Soil Interactions, Fig. 1 TDP of SCR (Fontaine 2006)



Pipeline Soil Interactions, Fig. 2 P-y curve in AB model (Aubeny and Biscontin 2009)



Pipeline Soil Interactions, Fig. 3 P-y curve in RQ model (Randolph and Quiggin 2009)



and repenetration. The curve describes the change of resistance of soil model with the buried depth of pipeline. Curve 1 is the initial penetration, curve 2 is the pull-up process, and curve 3, 4 are the process that the pull-up force increases first and then decreases. Curve 7 represent the repenetration process, which indicates that the soil stiffness decreases when repenetration occurs. As the burial depth continues to increase, the repenetration curve gradually merges into the

initial penetration curve. The variation of resistance in RQ model is related to the pipeline loading path, which can describe the disturbance degree of pipeline movement on the bottom soil to a certain extent. During the interaction between pipeline and seabed, plastic deformation occurs in the soil beneath the pipeline, which explains the reason why part of the soil is pushed by the pipeline to both sides to form a bulge and groove. However, the RQ model does not consider the

effect of pipe grooving when simulating upsetting.

You (2012) has improved the AB model by considering soil weakening with power function, incorporating the repenetration curve into the initial penetration skeleton line with a new formula, and adopting different mathematical formulas for each loading and unloading stage. Aubeny et al. (2015) based on the results of Langford and Meyer's tests (2010), a new weakening model was formed by considering soil weakening in AB model. The three stages of initial penetration, elastic rebound, and reloading were used to describe the nonlinear behavior of soil, and the effect of soil weakening was taken into account in the new model.

Vertical Pipe-Soil Interaction of Pipeline

Pipeline on the seabed will has preliminary embedment which called pipeline penetration. The pipeline penetration is affected by the following fundamental issues, such as geotechnical statement, pipeline laying operations, and scouring from the actions of waves and current. Figure 4 illustrates the definitions of parameters used in the model of pipe penetration (Bruton et al. 2008). To prevent the vertical movement of a pipeline in soil, the pipeline should be neither heavy enough to sink nor light enough to float. The pipelines to be buried should be checked for possible sinking or floatation. Sinking should be considered with maximum content density, such as water filled, and floatation should be considered with minimum content density, such as air filled.

The vertical pipe-soil interaction during pipeline penetration can be modeled by soil spring based on Winkler foundation beam theory

(Limura 2004) or shell model (Yatabe et al. 2002). The effects of deformation mode and the maximum strain of pipelines in different states can be obtained. A module used for assessing pore pressure accumulation in the seabed was designed by Bonjean et al. (2008). The module could assume both the uplifting force exerted on the pipeline and the corresponding soil resisting force.

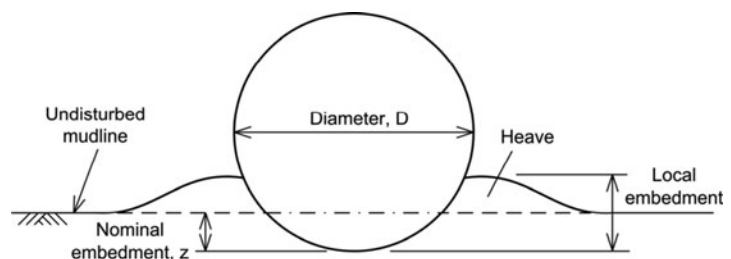
The process of pipeline penetration into the seabed was analyzed by Chatterjee, using the means of large deformation technology (Chatterjee et al. 2012). Considering the effects of strain softening, strain rate, and soil uplift, the formula for calculating the vertical resistance of pipeline was put forward, and a breakthrough in theoretical calculation was achieved.

Rossiter and Kenny (2012) examined pipe-soil interaction events during oblique lateral-vertical soil movements using plane strain finite element analysis. The results from this study provide a technical framework to assess complex loading events, such as coupled ice keel/seabed/pipeline interaction. Later, the discontinuous medium contact treatment method in discontinuous deformation analysis was used to simulate buried pipelines, and the feasibility and validity of solving pipe-soil contact with finite element method were discussed (Na et al. 2013).

White and Randolph (2007) described the relationship between penetration resistance and the vertical and lateral resistance of pipelines, using both in situ methods in the form of cylindrical (T-bar) and spherical penetrometers. The relationship taking account of the depth of burial and cycles of movement. Two new models of interaction between pipelines and seabed have been proposed: non-weakening model and weakening model (Jiao 2007). In both models, the main line

Pipeline Soil Interactions,

Fig. 4 Pipeline penetration in soil (Bruton et al. 2008)



is used to describe the pipeline penetration, the boundary curve is used to define the resistance-displacement relationship of the pipeline in large displacement, and a series of formulas are used to describe the resistance-displacement curve in the cyclic process.

A predictive model for estimating the compression and shear stresses around a buried pipeline in static conditions has been developed for determining stresses around a buried pipeline and verified using the Alyeska and Soil Box tests (Andrenacci and Wong 2012). Advanced pipe-soil interaction techniques that used to demonstrate benefits that can be realized on typical projects to reduce pipeline walking predictions and the associated mitigation measures were presented (White et al. 2015). Aubeny et al. (2015) formed a new weakening model considering soil weakening based on the test results of Langford and Meyer (2010). It used three stages of initial penetration, elastic rebound and reloading to describe the nonlinear behavior of soil. The interaction between soil and pipeline was studied by controlling the change of soil stiffness during uplift and repenetration.

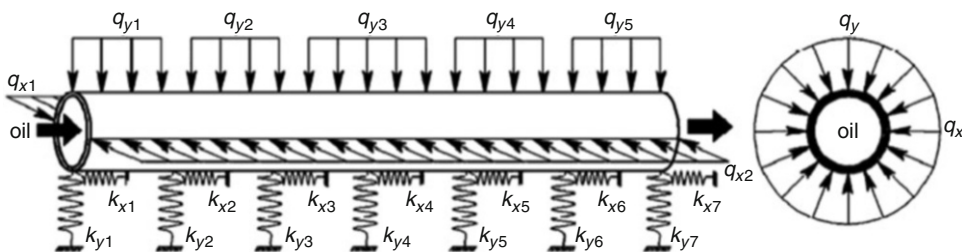
Computational fluid dynamics (CFD) was used to consider seabed soil as fluid and to establish a follow-up grid in a certain area around the pipeline (Hawlder et al. 2015). The computational method requires less computational resources than RITSS and CEL methods, and can also consider strength weakening and strain rate effect. Another research present details of a novel 2D pipe-soil-fluid (PSF) interaction algorithm, which has been developed to offer a more accurate model of the evolution of soil profiles around the pipeline (Griffiths 2012). A large number of parametric 2D CFD models have been run to generate

seabed shear stress profiles as a function of seabed and pipe geometry under different wave and current flow conditions. The PSF model has been designed to minimize computational cost but still enable the profile of the soil around the pipe to be tracked.

Lateral Pipe-Soil Interaction of Pipeline

Pipeline on-bottom stability design is conventionally based on the static balance between hydrodynamic forces and the soil lateral resistance forces, as shown in Fig. 5. Verley and Lund (1995) developed a soil resistance model applicable for pipelines laying on clay soils. Brennodden and Stokkeland (1992) performed some full-scale model tests for the Troll Phase 1 development. The results of these model tests revealed a significant increase in peak soil resistance (Bai and Bai 2014) (Fig. 6).

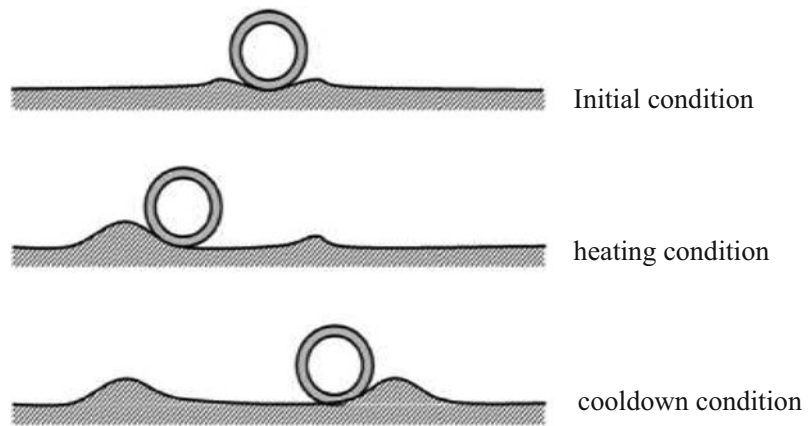
Based on theoretical analysis and experiments of lateral buckling of a pipe, Palmer and Baldry (1974) correctly interpreted the reason why lateral buckling can occur in a pipe and proposed an analytical formula for the critical pressure. Hobbs (1984) studied the lateral buckling of a perfect straight pipeline on the basis of related work on railroad track and proposed five buckling modes which may occur in the process of lateral buckling. Analytical formulas for critical force, wavelength, and amplitude corresponding to every buckling mode are also presented. Taylor and Gan (1986) considered the initial imperfections and proposed analytical formulas for critical forces of lateral buckling corresponding to mode 1 and mode 2. Finite element (FE) analysis was



Pipeline Soil Interactions, Fig. 5 Soil spring model

Pipeline Soil Interactions,

Fig. 6 Lateral buckling under cyclic thermal load (Bai and Bai 2014)



used to study the lateral buckling responses of submarine pipelines (Liu et al. 2014), some formulas were proposed to calculate the critical force of lateral buckling. The initial imperfection parameters (initial wavelength and maximum amplitude) are included in these formulas.

The stability of submarine pipeline during service is studied by hydrodynamic loading method (Fuping 2001). And the test reflects the coupling effect among wave, pipeline, and seabed. It is found that the lateral instability of submarine pipeline generally goes through four stages: complete stability, sand bed erosion, slight displacement of pipeline, and pipeline instability. It is also found that loading history, soil parameters, and initial settlement of pipeline all affect the stability of pipeline. On the basis of experiments, the empirical relationship of wave-tube-soil interaction for pipeline stability is preliminarily established.

Some results simulated the seabed by adding a nonlinear spring under the linear elastic pipeline, and put forward the corresponding analytical model of pipe-soil interaction (Aubeny and Biscontin 2008). The nonlinear spring takes into account the nonlinearity of the load-displacement relationship of the seabed and the separation of pipe and soil. A physical model was used to understand the mechanisms of interaction between pipe and soil through a lateral visualization, and then to simulate the pipe response under combined vertical and horizontal loads (Orozco-Calderon et al. 2012). This entry concentrated on

the results obtained using sideswipe tests with large and short horizontal displacements, and obtained an experimental yielding envelope of the pipe.

In the recent years PETROBRAS has developed several projects in deepwater areas of the Santos and Campos Basins, offshore Brazil, where soft clay is predominant. This has motivated the development of an extensive experimental test program for pipe-soil interaction, aimed at increasing the design reliability of subsea pipelines with lateral buckling (Cardoso and Silveira 2010). A new model presented was based on full-scale tests developed at the Institute of Technological Research (IPT) installations over the last 3 years. The experimental results were used to develop a model for the lateral residual friction factor based on dimensionless groups that govern the problem. The aim of this model is to improve knowledge obtained by the industry in recent years, mainly by the JIP SAFEBUCK I and II programs.

Another study concerned the lateral buckling critical force of a submarine PIP (pipe-in-pipe) pipeline with one symmetrical initial imperfection on soft foundation (Xinhu et al. 2018). On the basis of 3D beam elements, tube-to-tube interaction elements and pipe-soil interaction elements, 3D finite element model were built for a length of submarine PIP pipeline. The simple empirical formula was proposed to calculate the lateral buckling critical force of a submarine PIP pipeline based on dimensional analysis and finite element

results, and the case study was carried out to verify the obtained empirical formula and the results manifest its good accuracy.

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Pipe-Soil Interaction

► Pipeline Soil Interactions

Piping Technology

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Synonyms

High chromium duplex corrosion-resisting stainless steel (HDR); Numerical control (NC); Tungsten insert gas (TIG)

Definition

Ship piping system refers to the piping system of the ship, which includes all the pipelines, accessories, machinery, equipment, and appliances used to transport fluids. Ship piping system is key to ensuring the normal operation of the ship power plant, deck outfit, ballast systems, and living facilities and to the safe navigation of ships. It is also an important guarantee for the service life of ships.

For shipbuilding, ship piping is one of the largest engineering projects of both processing and manufacturing in the workshop and installing and debugging in the field (Rao et al. 2020). For example, the ship piping of a 10,000 DWT grade bulk carrier can account for about 25% of the total workload of shipbuilding. Another example is that piping system of a chemical product transport ship includes dozens of systems, nearly a 1000 types of pipe specifications, and tens of thousands of pipes. Therefore, the processing and assembly of ship pipes is an important part of the construction of the ship structure.

In modern shipbuilding, pipe system production is considered as an important processing step

in the manufacturing of shipbuilding enterprises. There are many types of pipes with different models and different uses. According to their different transmission media, pipe systems can be divided into steam pipe system, fuel oil and lubricating oil pipe system, compressed air pipe system, and cargo oil and liquefied gas pipe system. According to their different functions, pipe systems can be divided into high pressure pipe system, nuclear power pipe system, high temperature pipe system, low temperature pipe system, chemical medium pipe system, etc. (Xu 2011). According to their different materials, pipes can be divided into three categories: steel pipes, non-ferrous metal pipes, and nonmetallic pipes.

Introduction

Pipe Processing Technology

The pipe processing workshop is the place where the shipbuilding enterprise processes and manufactures ship pipes. In the modern shipbuilding model, pipe processing workshop is arranged with the principle of group technology, thus establishing a production operation system guided by the principle of manufacturing “intermediate products” of the ship. The pipe processing workshop first decomposes the pipe fitting production tasks according to pipe processing drawings and part lists issued by the pipe design department, together with the tray plan of the piping tray. Then the workshop distributes the decomposed production tasks to different teams, each of which, according to its task, receives the raw materials from the raw material warehouse of the pipe processing workshop and then completes the pipe processing according to the production process of the pipe processing drawing, including the processes of cutting, bending, reshaping and positioning, welding, polishing, testing, and surface treatment (Li 2016).

Cutting

Correct cutting is the first step in the pipe processing work. Before cutting, we obtain the theoretical calculation length of the pipe from the design drawing. The cutting length is considered

as the overall length of the bending pipe, which is theoretically the overall length of both the straight pipes and the elbow arcs. When the pipe is being bent, the arc length of the elbow part is slightly elongated due to plastic deformation of the material, so the actual cutting length should be slightly shorter than the theoretical length.

Elongation is an experience data, which depends on the pipe material, diameter, wall thickness, bending radius, bend size, and the choice of hot or cold bending. Generally, for a pipe of a certain specification and material, the elongation is proportional to the bending angle α . At present, the actual cutting length of a pipe in the factory is usually its theoretical cutting length, and the elongation will be cut off when the pipe is reshaped and positioned. However, if the "welding prior to bending" technology is adopted, the elongation must be considered, usually based on the experience data obtained from the test (Fan 2018).

Generally, pipes are cut by the sawing machine or through oxy cutting, but after oxy cutting, iron oxide residue and spatter must be removed. Non-ferrous metal pipes and HDR stainless steel pipes should be cut by machine saw or the handsaw.

1. Gas cutting

Gas cutting of metals (in reality, oxygen cutting) is currently the most commonly used cutting method for ship piping. Its process is described as follows: first, the metal at the cutting slit is heated with a preheating flame, so that the metal temperature at the starting point of the slit rises to the ignition point (the ignition point of the metal in pure oxygen); then the oxygen flow with high pressure is released and causes the metal to burn (violent oxidization); the slag generated by the burning is immediately blown away by the airflow; continue this process and a smooth slit will be formed in the metal to separate it.

2. Plasma cutting

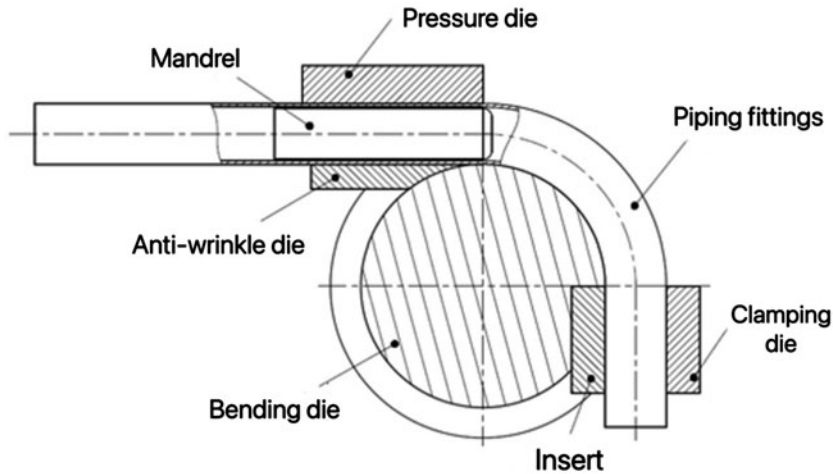
Plasma arc cutting is a processing method that uses the heat of a high-temperature plasma arc to locally melt (and evaporate) the metal at the slit of the workpiece and uses the momentum of the high-speed plasma to remove the molten metal to form a cut. Plasma cutting,

with different working gases, can cut all kinds of metals that are difficult to cut by oxygen, especially nonferrous metals (stainless steel, aluminum, copper, titanium, and nickel). Its main advantage is that it can reach a high speed when cutting the metal of limited thickness. Especially when cutting ordinary carbon steel sheet, the speed of plasma cutting can be five to six times that of oxygen cutting, with a smooth cutting surface, limited thermal deformation, and a smaller heat-affected zone.

Pipe Bending

Rotary draw bending is the main form of cold bend and is also used by most NC pipe bending machines. The pipe bending performed on the pipe bending machines is often divided into two categories: bending without mandrel and bending with mandrel. In order to prevent excessive flattening of the cross-sections of the bending part, the method of bending with mandrel is often used for pipes with relatively thin walls, while the method of bending without mandrel is used for pipes with relatively large wall thickness or with low or even no requirement for cross-section flattening. The schematic diagram of the processing by pipe bending machine is shown below (Fig. 1).

The bending forming process is completed with the cooperation of multiple molds. The bending radius of the pipe and the cross-section shape of the bending segment are determined and controlled by the bending die. The clamping die fixes the front end of the pipe on the bending die and gives the pipe a certain clamping force to ensure that the three-in-one rotation of the pipe, the bending die, and the clamping die completes the bending of the pipe. The pressure die supports the outside of the pipe and can move together with the pipe to boost the pipe, thus limiting the decrease in the thickness of the pipe outer wall and the increase in the thickness of the pipe inner wall at the bending segment. The main function of the antiwrinkle die is to reduce the wrinkles caused by pressure on the inner side of the bending position, so an antiwrinkle die must be used in the bending with higher requirements for inner wrinkling or in the bending of large-diameter thin-walled pipes. The mandrel is mainly used in



Piping Technology, Fig. 1 The schematic diagram of the processing by pipe bending machine

the bending of pipes with relatively small wall thickness, to prevent excessive flattening of the cross section of the bending segment, and the mandrel also has a great influence on the thickness change of the outer wall of the pipe and the rebound of the pipe (Wu 2016).

Pipe Reshaping and Positioning

The pipe reshaping and positioning by the pipe reshaping machine is a method of installing the slip-on welding flange on the main pipe. Based on the information of the pipe parts drawing, the pipe reshaping machine is moved to an appropriate position, and a flange is installed on the template, with a steel ring inside the flange, the thickness of which equals to the weld end distance. Then both ends of the pipe to be formed are set into the flange, and the position of the pipe is adjusted according to the drawing. Then the flange positioning work begins to be performed (Fig. 2).

Platform pipe reshaping and positioning is another method of assembling slip-on welding flange on the main pipe. The method of marking on the flange surface is used based on the flange bolt hole angle data in the piping parts drawing, and with the help of tools such as flange ruler, level bar, angle ruler, and plumb, the pipe is set on the platform according to certain requirements in order to perform flange positioning. General requirements for pipe reshaping and positioning are that

the end of the pipe should be flush; the surface of the pipe should be clean; the gap between the pipe and the accessories should be uniform and meet the relevant process requirements.

Welding

1. Workshop welding of ship pipe fittings

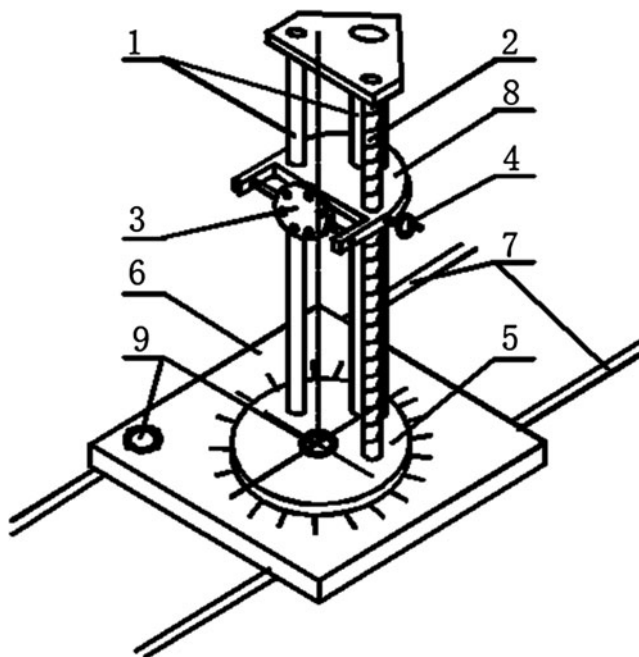
Ship pipe fittings refer to semi-finished pipe fittings of all types, batches, materials, and sizes that are welded and produced in the workshop and then sent to the hull block for group welding. The biggest advantage of preparing semi-finished pipe fittings under the conditions of workshop production is that the pipe accessories are easy to assemble, and the functions of various automatic pipe welding equipment and auxiliary welding equipment are given full play. The satisfactory welding quality could also be achieved in some occasions where manual welding is used.

As shown in Fig. 3a, an all-position automatic pipe welding machine is used in the workshop to perform pipe-pipe butt welding after preheating. As shown in Fig. 3b, a large all-position automatic pipe welding machine is used in the workshop to perform large-diameter pipe-pipe butt welding. As shown in Fig. 3c, a narrow-gap gas welding machine is used in the workshop to perform thick-walled pipe-pipe butt welding.

Piping Technology,

Fig. 2 The schematic diagram of the pipe reshaping machine.

- (1) Column; (2) screw rod;
(3) template; (4) hand grip;
(5) swivel plate;
(6) pedestal; (7) guide rail;
(8) guide frame; (9) brake



2. On-site welding of ship pipe fittings

On-site welding of ship pipe fittings (Fig. 4) refers to the welding of pipe fittings on the sites of hull block. On-site welding of ship pipe fittings is generally performed in the form of pipe-pipe butt joints, but various welding positions may be required (Figs. 5 and 6), so the automatic pipe welding machines are usually used to ensure high welding quality and reduce welders' labor intensity.

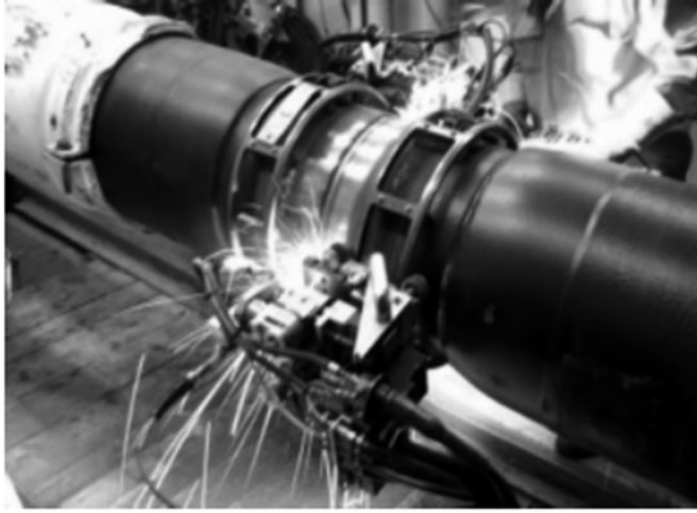
Welding of ship pipe fittings can be summarized into several types: the first type is pipe-pipe butt joint (Fig. 7), pipe-flange butt joint (Fig. 8), and pipe-pipe multipass butt joint (Fig. 9). This type of ship pipe fittings with butt welds are most widely used in ship piping system, and for this type, the standardization and serialization have been formed in the aspects of materials, models and sizes, pressure ratings, etc., for the pipes, pipe flanges, pipe elbows, three-way pipes, and other accessories within the common size range, as shown in Fig. 10.

The other type is the branch pipe welding, which is rare among ship pipe fittings within the common size range, because this type of joint can

generally be replaced by a three-way pipe joint; If the branch pipe welding becomes necessary, it must be used for the joint of a non-standardized pipe (or fittings) or a special thick-walled pipe (or fittings) with a tube, which is usually called saddle-shape weld joint in the industry.

The most widely used automatic arc-welding equipment in the on-site installation of ship pipelines (block pipe system) is the all-position automatic pipe welding machine. It is an automatic arc welding equipment which can weld fixed pipe joints (with pipe fittings not rotating) at all positions (the meaning of "all-position" is shown in Fig. 11).

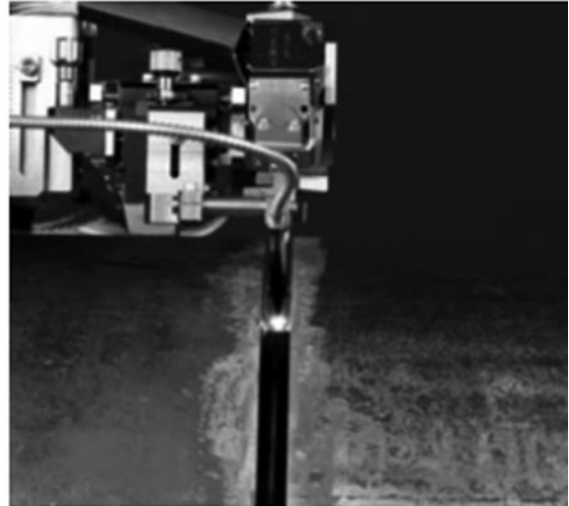
In Fig. 11, the butt weld pipe joint to be welded does not rotate, but the welding gun (usually a TIG welding gun, namely, an argon tungsten-arc welding gun) rotates more than 360° around the centerline of the groove of the pipe joint. In this way, a variety of welding positions on the pipe surface could be reached by the welding gun: flat welding position, vertical down welding position, overhead position, vertical upward welding position, and multiple welding positions among these four welding positions; because to complete the circumferential weld of a pair of butt-welded



(a)



(b)



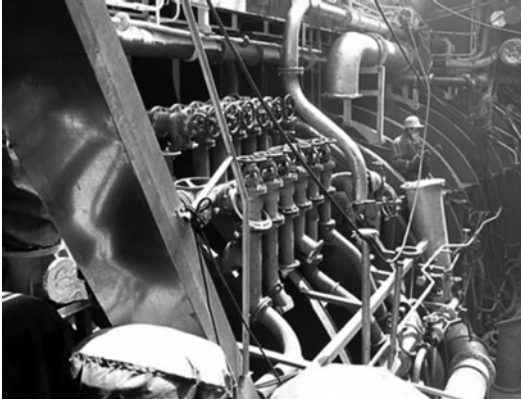
(c)

Piping Technology, Fig. 3 The automatic welding method of centralized pipe systems in the workshop

pipes, the welding gun will be at a variety of welding positions, so this arc-welding equipment is called an all-position automatic pipe welding machine. It is a highly automated mechatronic arc welding equipment, because the completion of the welding of the circumferential weld of a pair of pipes requires not only that the welding head rotates around the pipe at a uniform welding speed, but also that the welding current, shielding gas, filler wire, and cooling water are connected to

the corresponding parts of the welding head, as shown in Fig. 12.

In order to meet the process requirements of different arc welding and ensure excellent welding quality, sometimes the welding head is required to swing laterally during welding. Therefore, from the appearance, there are many wires, water pipes, gas pipes, and cables led to the pipe welding machine. In order to realize the smooth “three links” of water, electricity and gas, the



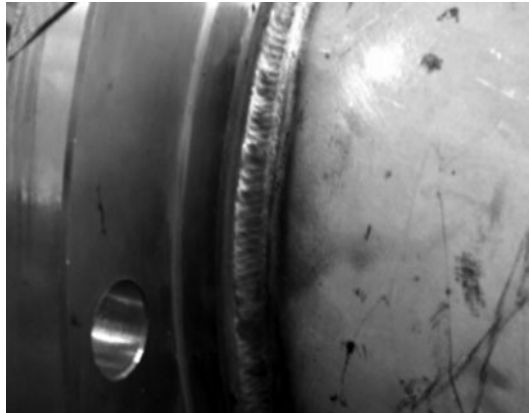
Piping Technology, Fig. 4 On-site welding of ship piping system



Piping Technology, Fig. 7 Unequal diameter pipe-pipe butt weld joint



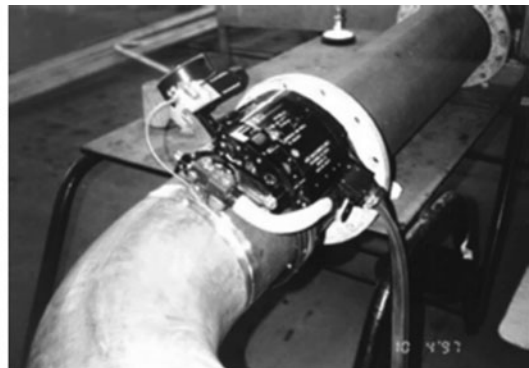
Piping Technology, Fig. 5 On-site automatic welding of ship piping system (horizontal welding)



Piping Technology, Fig. 8 Pipe-flange butt weld joint

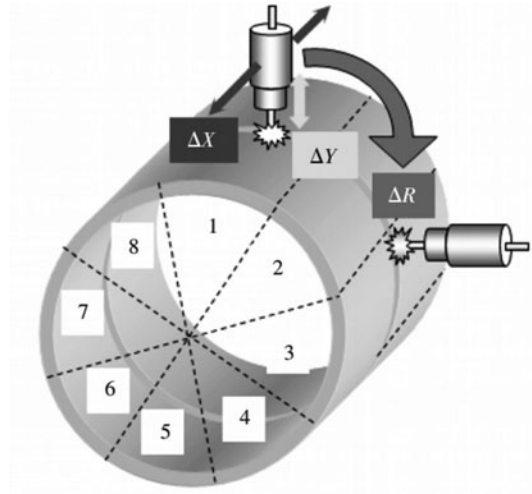


Piping Technology, Fig. 6 On-site automatic welding of ship piping system (horizontal position)





Piping Technology, Fig. 9 Pipe-pipe multipass butt weld joint



Piping Technology, Fig. 11 The schematic diagram of the all-position welding machine



Piping Technology, Fig. 10 Standard fittings of several pipe flanges

wires, water pipes, gas pipes, and cables of the pipe welding machine must be reasonably arranged. Obviously, the smaller the diameter of the pipe being welded is, the smaller the volume of the pipe welding machine is, but the higher the precision for the mechatronics system of the pipe welding machine is required to be, and the higher the flexibility of the pipe welding machine is required to be.

Pipe-Piece Family Manufacturing

The pipe-piece family manufacturing method is another successful application of group



Piping Technology, Fig. 12 The travel carrier pipe welding machine

technology by Ishikawajima Harima Heavy Industries of Japan. In an ideal situation, the various machines needed to manufacture a particular intermediate product family should be set into a group to form a production line. The group of machines can complete all processes as a predetermined process unit, and this method is called “process classification.” This method replaces the

traditional process design method required to manufacture parts. When this method is applied to a pipe workshop, it is called a pipe-piece family manufacturing method.

The pipe-piece family manufacturing method is a comprehensive method that simplifies the manufacturing process of the pipes of mixed varieties and mixed batches, such as vent pipe parts manufactured together with other pipe parts. This method requires more complicated plans and schedules than traditional, inefficient, and system-oriented plans and schedules. With this method, mass-manufacturing mode could be adopted for manufacturing complicated varieties of ship pipes, and piping part design, manufacturing technology, and production management could be reasonably arranged, so that the general shipbuilding productivity could be better improved by piping engineering, and the productivity of pipe fittings could be greatly improved, too (Xie 2011).

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Planar Motion Mechanism (PMM)

- [Maneuverability of Polar Vessel](#)

Plastic Design

- [Offshore Structure Design Under Ice Loads](#)

PLR (The Plug Length Ratio)

- [Offshore Pile Driving](#)

Polar Acoustics

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Polar acoustics (also known as Arctic Ocean acoustics) mainly studies the underwater acoustic environment in the polar region and adjacent sea areas and is a branch of hydro-acoustics. The significant research began during the Cold War (1947–1991) with the advent of the nuclear submarine. The main contents of polar acoustics include the marine environment noise under the ice cover, reverberation characteristics, the characteristics of underwater acoustic channels, and underwater acoustic signal processing. Based on these understandings, polar acoustics are well applied in the polar region, named polar acoustics technology.

The presence of ice and its spatial and temporal variability are the main characteristics of the Arctic Ocean that affect the underwater acoustic environment. This phenomenon is particularly prominent in Arctic Ocean, unlike the Antarctica that is covered by ice perennially. The growth and ablation of sea ice make the Arctic acoustic environment unique and interesting.

Ambient Noise

The ambient noise in the Arctic Ocean consists of many types and is different from any other sea

area. There is few shipping noise due to less water traffic. Sea ice movement, earthquake, and biological activity are the mainly sources of Arctic Ocean ambient noise.

The frequency spectrum range of the Arctic Ocean ambient noise is approximately 1 ~ 1000 Hz, and the peak occurs mainly at two specific frequencies of 15 Hz and 300 Hz (Dyer 1988). The peak at 15 Hz is believed to be generated by ice ridge activity as shown in Fig. 1, and 300 Hz is considered to be caused by temperature-induced thermal cracks. These two types are the main sources of Arctic noise. Zakarauskas summarized the Arctic ambient noise:

1. Pressure Ridging Noise

Floating ice is affected by the wind and ocean currents and squeezed to form ice ridges. Collisions, squeezing, and friction between sea ices produce low-frequency noise during the process (Greening and Zakarauskas 1994a).

2. Thermal Cracks Noise

As the season change, shrinkage or expansion occurs when the temperature is greatly different

between the upper and lower surface, resulting in thermal cracks (Greening and Zakarauskas 1994b). Thermal cracks are subtle noise phenomenon that takes place on the surface of the ice layer. It usually occurs during the night, and maintenance is 0.05 ~ 0.1 s. Different from the noise produced by the formation of ice ridges, the frequency spectrum of the thermal crack noise is relatively flat, and the noise produced by the ice ridge is mainly in the low-frequency range.

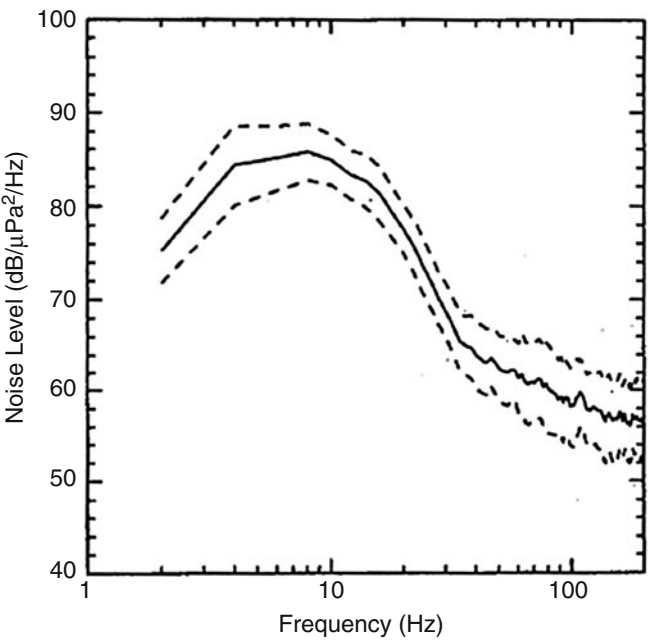
3. Snow Noise

The polar high atmospheric pressure exists for a long period over the Arctic Ocean, and the east wind prevails in winter. Snow noise become the main noise in the quiet ice area such as huge ice cap in winter and spring. The wind blows the snow particles on the ice cap to rub against it, and then the sound enters the water through the ice layer, creating snow noise. The frequency of this noise is between 2 Hz and 2 kHz.

4. Earthquake Noise

P wave generated by earthquakes have a frequency of 0.1 ~ 5 Hz and carry little energy. The

Polar Acoustics,
Fig. 1 Frequency
spectrums of pressure
ridging noise (Greening and
Zakarauskas 1994a)



T wave produced by the earthquake has more energy, and the frequency band is 5 ~ 60 Hz. Such waves propagate from the bottom of the sea to the water and reach the ice. This type of noise is more difficult to distinguish from long-range ice ridge forming noise, since the frequency bands of the two are similar.

5. Biological Noise

Take the whale noise as an example: the noise emitted by whales is about 15–45 Hz, and the duration is 1–20 s. Generally, there will be 50–100 repetitions, each interval being 1–20 s. There are some animal noise such as seals, but the impact of biological noise on the overall ambient noise in the Arctic is limited.

6. Ice Shore Noise

Ice shore noise is caused by the effects of ocean currents, sea breeze, and ice vortex on the coast. Noise levels in this area are generally higher. The main factors affecting the noise level are seawater conditions, water depth, and the size of the waves. T.C. Yang used sonar buoys in Greenland to observe ice shore noise and measured different shore ice conditions separately. The noise level is relatively high when the ice shore is compact and free of ice floes. In contrast, the noise level is relatively low when the shore has scattered floes.

7. Iceberg Melting Noise

There are a large number of tiny air bubbles in the water that forms icebergs. They are solidified in the ice due to the pressure. As the iceberg melts in the seawater, the gas in the ice suddenly released and made a sharp crack when the melting surface reaches the air bubbles. A sonar buoy is placed near the iceberg measuring the iceberg melting noise and found that the noise had a flat spectrum.

Propagation

The main features of the underwater propagation environment of Arctic Ocean were recognized: a

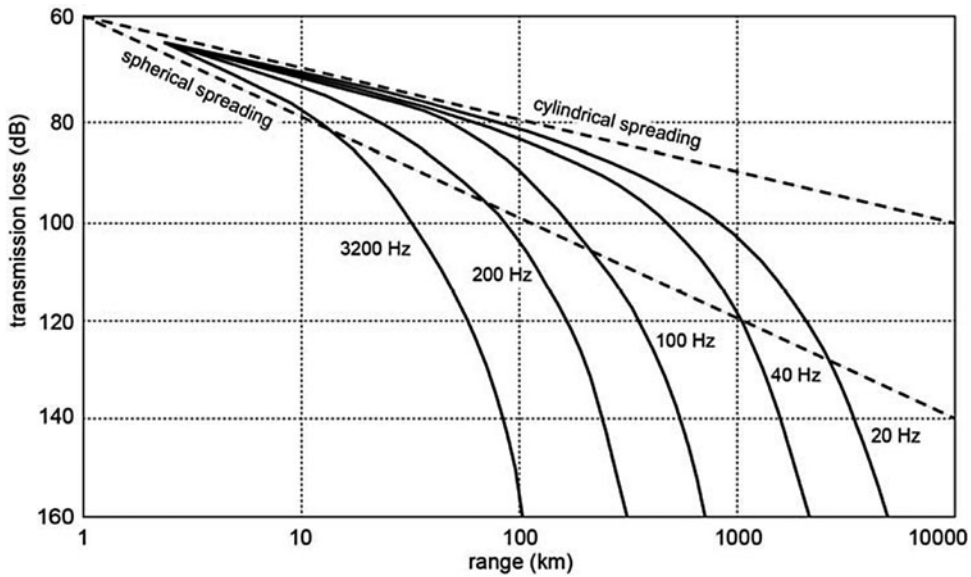
stable upward-refracting sound speed profile with an upper boundary of rough polar ice (Hutt 2012). Baggeroer reports that there are two positive spring layers in Arctic Ocean, halocline at about 40 ~ 50 m depth and syncline at 200 ~ 250 m depth, respectively.

When the source is upper the halocline, the sound is constantly reflected at the lower surface of the rough polar ice and can travel a long distance, which act like a surface duct. Acoustic waves with frequencies 15 ~ 30 Hz propagate best because of the rough polar ice scattering and energy conversion between flexural wave and polar ice. As the frequency increases, attenuation increases dramatically, because high-frequency components are scattered out by rough polar ice. Yet very low frequencies, below about 10 Hz, are not effectively trapped by the duct. Urlick summarized Arctic Ocean transmission loss shows that for frequencies less than about 100 Hz, propagation is approximately characterized by cylindrical spreading, which is an indication that the surface duct is an excellent waveguide for low frequencies (Urlick 1983). Characteristic transmission loss versus range based on numerous measurements is shown in Fig. 2.

When the source is near the spring layers, some grazing angle sound are trapped in another duct called “SOFAR” duct. In this duct, the sound can propagate without touching the bottom of the polar ice and the seabed so that travel a long range.

Reverberation

The sound waves emitted by the source impose the scatters and distribute the energy in all directions, i.e., result in scattering waves. The sum of the scattered waves returned to the receiving point constitutes the reverberation. The impact of polar ice variability is a major contributor in the reverberation of Arctic Ocean. The reverberation environment under Arctic ice is characterized by highly variable (and often large) scatter from the water-ice interface and low-volume scatter from the water column. The reverberation intensity is strongly correlated with the ice type and ice



Polar Acoustics, Fig. 2 Characteristic transmission loss versus range based on numerous measurements from the 1960s (Urick 1983)

roughness; sometimes the level of reverberation caused by Arctic ice is more than 40 dB higher than that of sea surface without sea ice (Marsh 1961).

J. R. Brown and A. R. Milne found that the scattering intensity of the covered sea ice region increases with the increase of the frequency and the grazing angle through the analysis of the observation data of two observation sites in the Arctic Ocean at different times of the year (Brown 1964). J. E. Burke and V. I. Twersky expressed the ice ridge as a semicylindrical elliptical rigid body and establish a Burke-Twersky model (Burke and Twersky 1966) that describes the reverberation under the ice in 1966, which is widely referenced and modified in following research.

In the 1990s, T. C. Yang and T. J. Hayward conducted research on low-frequency Arctic reverberation at different distances using CEAREX 89 test data from the Norwegian-Greenland Sea (Hayward and Yang 1993). For the short-range (less than 3 km) direct ice surface reflection reverberation, it is proved that the measured scattering intensity is consistent with the Burke-Twersky model law at the low frequency (24 ~ 105 Hz) when the grazing angle is less than

20°. For the mid-range (5 km ~ 20 km) mixing and reverberation under the ice surface and the seabed, the vertical array was used to separate the under-ice surface and the seafloor reverberation, and their degree of agreement with the model was verified respectively. For remote (no more than 200 km) very low frequency (10 Hz ~ 50 Hz) reverberation, an Arctic reverberation model based on a normal wave is established, and the model is verified by measurement data. It is pointed out that the effective reverberation intensity depends on the depth of the sound source in this case. Besides, the scope of the active sonar affected by reverberation can reach 200 km.

K. LePage and Schmidt find that Arctic reverberation has great spatial correlation, and half-wavelength correlation can reach 0.99 or more.

Polar Acoustics Technology

After the end of the Cold War, the interest in polar acoustics shifted from underwater surveillance to sensing of the Arctic Ocean and engineering application. Polar acoustics technology refers to the application of polar acoustics in the fields of

engineering implementation, environmental monitoring, shipping traffic, etc.

1. Acoustic Ice Thickness Measurement

According to different measurement principles, sea ice thickness measurement methods are mainly divided into direct measurement methods and indirect measurement methods. Indirect measurement methods are divided into physical detection methods and satellite remote-sensing methods.

Direct measurement methods mainly include sampling method of crushed ice, hole drilling method, and resistance heating method. The direct measurement method is to manually work directly on the ice, remove the ice sample, and measure the ice thickness directly. The method has advantages of high measurement accuracy and reliable data, but it is time-consuming and labor-consuming. It is only suitable for small areas and the thin ice thickness measurement.

Satellite remote-sensing uses satellite data for ice thickness measurement. This method can be used for a wide measurement range, but there are measurement errors at the intersection, and data reliability is poor.

Physical measurement methods include capacitance method, conductivity method, temperature method, electromagnetic induction method, ice-freezing degree method, on-site image measurement method, radar laser ranging method, and sonar measurement method. They can realize automatic monitoring of ice thickness for a long time without destroying the ice layer. In particular, in upward-looking sonar measurement method, the obtained ice thickness data has higher measurement accuracy and reliability and is less affected by external environmental factors.

Hudson and Wright tested the mooring fixation method in the Beaufort Sea, and in 1990s, Canadian Institute of Marine Fisheries Research develops Ice Profiling Sonar in order to measure the ice ridge thickness and underwater ice appearance of sea ice. Upward-looking sonar is also installed on a nuclear submarine and observes the thickness and morphological characteristics of ice ridges in the Arctic from beneath the ice

sheet. In 1996, ASL Corporation of Canada developed and commercialized the production of the top sonar. In 2008, it developed a new generation of model IPS5, whose ice thickness measurement accuracy in the vertical dimension reaches the order of centimeters, in the horizontal dimension, reaching the order of meters.

2. Acoustic Communication and Navigation

Acoustic communication is a useful method of transmitting information in the ocean, especially in the polar region covered with sea ice, as shown in Fig. 3. During the icing period, it is very difficult for vehicles to surface to communicate and obtain location information because of the uncertainty in the floating and thickness of sea ice. Furthermore, the harsh climate conditions in the Arctic are not suitable for long periods of activity. Acoustics solve this problem because the sound can travel a long distance. H. C. Song reported a successful acoustic information transmission over about 2720 km with a data rate of 2 bits/s (Song et al. 2014). More recently, Woods Hole Oceanographic Institution achieved underwater vehicle communication and navigation up to ranges of 70 ~ 90 km at a data rates 5 ~ 10 bit/s (Freitag et al. 2012).

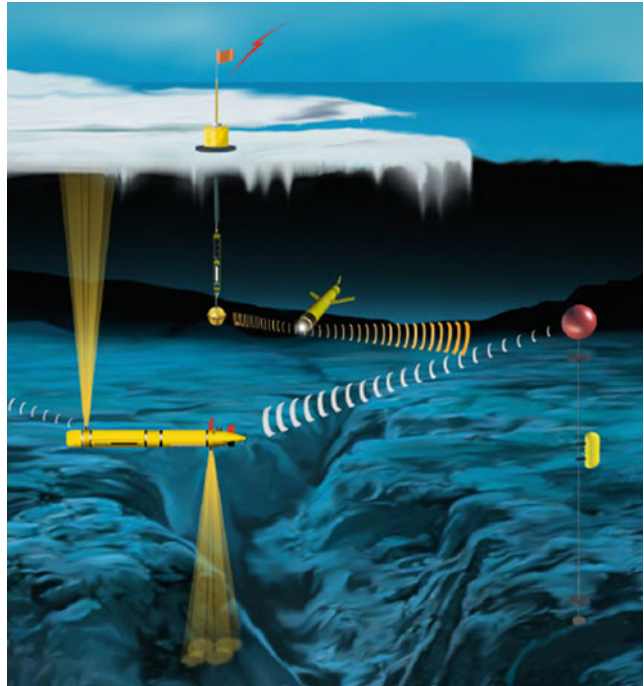
3. Acoustic Thermometry

The idea of using low-frequency pulses transmitted over long distances to measure small changes in average ocean temperature was developed in the 1980s. This technique is called acoustic thermometry. The first attempt to carry out acoustic thermometry in the Arctic was the 1993 international experiment called Transarctic Acoustic Propagation (TAP). By examining the modal structure of the received signals, which is affected by bathymetry, ocean structures, and ice thickness, it was deduced that the Atlantic intermediate layer was intruding further into the Arctic Ocean (Hutt 2012).

A follow-on project to TAP, and also a US-Russian collaboration, was the Arctic Climate Observations Using Underwater Sound (ACOUS) experiment in 1998. Since the success of ocean

Polar Acoustics,

Fig. 3 Gliders and powered AUVs performing oceanographic work beneath the Arctic ice (Freitag et al. 2012)



acoustic thermometry and large-scale ice thickness measurement, the ACOBAR project has been established as a long-term tomography project by the Nansen Environmental and Remote Sensing Center; the ACOBAR experiment consists of a triangle of acoustic transceivers approximately 200 km from each other with a vertical receiver array in the center.

New Arctic Acoustics

With the global warming, the Arctic sea ice is decreasing year by year, which makes the acoustic environment of the Arctic Ocean different from that of the 20th century. Henrik Schmidt points that the changes in the environment of the Beaufort Sea severely deteriorated the tracking performance compared to previous deployments (Schmidt and Schneider 2016), and the reason was associated with a previously observed neutrally buoyant layer of warm Pacific water persistently spreading throughout the Beaufort Sea, which severely alters the acoustic environment with dramatic effects for both long- and short-

range acoustic sensing, communication, and navigation. All kinds of changes require us to reinvest to understand the current polar acoustics.

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Polar Communications

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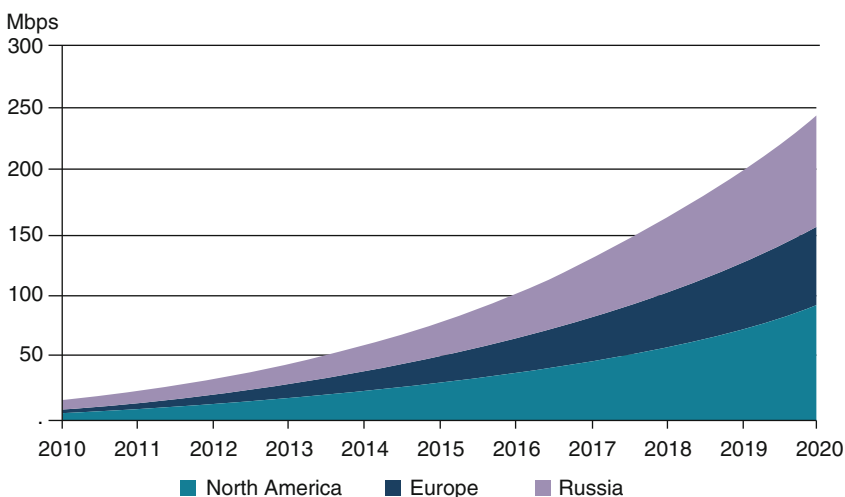
Introduction

In the twenty-first century, the Arctic and Antarctic regions, where the rich deposits of minerals have been found, are more and more

extensively being developed by humankind. For example, in the Arctic, gas deposits are estimated at 47.3 trillion m³, oil at 90 billion barrels, and gas condensate at 44 billion barrels. Their effective development is impossible without establishing reliable radio communication links with the Arctic and Antarctic regions.

According to the estimation by the European Space Agency, the stream of data in the Arctic region will increase tenfold over 2010–2020 (see Fig. 1). At the same time, the organization of communications in the Arctic region is hindered by the features of this region, the atmospheric and geospace structure, including the upper atmosphere, the ionosphere, and the magnetosphere. The Earth's magnetic field geometry is such that solar energetic particles penetrate into the polar regions, and the aurora, instabilities, geospace medium heating, a significant (by 1–3 orders of magnitude) increase in the electron density of the lower ionosphere, and other processes operate. All these events determine the instability of the atmosphere-ionosphere-magnetosphere communications links and result in considerable difficulties for organizing communications in the polar regions.

The instabilities in communications link significantly increase during major variations in space weather, the so-called geospace storms. As



Polar Communications, Fig. 1 Data rates required for the 75–90°N Arctic region (evaluated by European Space Agency)

is well-known, the geospace storm is comprised of the magnetic storm, the ionospheric storm, and the electric storm. The most prominent geospace storms in the polar regions occur when the ionospheric parameters and geoelectric fields vary by a factor of a few times to tens of times.

Conventional Communications

Communications in the polar regions provide radio waves in the frequency band from VLF (3–30 kHz) to VHF–UHF (30 MHz–3 GHz) when the near-Earth medium is used as a communications link.

VLF radio waves can propagate in the earth-ionosphere waveguide at distances, R , of 1,000–10,000 km. VLF communications are not disrupted even during the most severe geospace storms. The major drawback of VLF communication services is that the construction of large antenna systems with spatial sizes of about 100–1,000 m and high-power (of order of 1 MW) radio transmitters is needed and narrow (about 1 kHz) bandwidths are provided.

LF radio waves (30–300 kHz) can be reflected from the lower ionosphere over distances to a few thousands of kilometers. During geospace storms, the absorption of these radio waves drastically increases, and the link path length decreases. To provide stable communication services, large antenna systems with tenths of meters in height and high-power (~0.1–1 MW) transmitters are needed.

MF radio waves (300 kHz–3 MHz) can provide a ground-wave signal over distances of 100 km, and the signal reflected from the ionosphere can reach at distances of a few hundreds of km. The MF band has almost the same drawbacks than the LF band has.

HF radio waves (3–30 MHz) are particularly suitable for providing communication services in the polar regions. The radio waves in this wave band can travel as the ground wave at distances of no more than 50–100 km and as the ionospheric sky wave up to worldwide distances. When the distance reaches 4,000 kilometers, the HF radio wave can travel with a single reflection from

the ionosphere, while at greater distances, the HF wave can propagate in a waveguide with the walls which are usually made up by the ionospheric E and F layers. This waveguide is termed the interlayer waveguide, and the HF attenuation in this waveguide is significantly lower than on multi-hop links. The HF radio waves are mostly subjected to the variations in space weather.

The optimal frequency is given by

$$f_{opt} \approx 0.85 f_{MUF},$$

where f_{MUF} is the maximum usable frequency. This frequency does not exceeds $3f_{pmax}$, where

$$f_{pmax} = \frac{1}{2\pi} \sqrt{\frac{e^2 N_{max}}{\epsilon_0 m}}.$$

Here f_{pmax} is the plasma frequency, e is the electron charge, m is the electron mass, ϵ_0 is the permittivity of free space, and N_{max} is the F -region peak density. Under quiet conditions, the value of N_{max} is equal to approximately 10^{12} m^{-3} in the daytime and about $(2-3) \times 10^{11} \text{ m}^{-3}$ at nighttime, resulting in the diurnal variation of $f_{pmax} \approx 9-4 \text{ MHz}$ and of $f_{opt} \approx 23-10 \text{ MHz}$, respectively. During a positive ionospheric storm, the N_{max} values can increase by a factor of 2–2.5, resulting in $f_{opt} \approx 35-15 \text{ MHz}$. During a negative ionospheric storm, the f_{pmax} decreases by a factor of 3–4, resulting in $f_{opt} \approx 7-3 \text{ MHz}$. It should be noted that during ionospheric storms, the ionospheric D region electron density (50–90 km altitude) may be increased by a factor of 10–100 and the ionospheric E region electron density by a factor of 2–3. The integral absorption coefficient, K , may be increase by a factor of the same magnitude as given by

$$K = \frac{2\pi f}{c} \int_s \kappa ds,$$

where f is the radio wave frequency, c is the speed of light, ds is the radio wave path, s is the element, and κ is the absorption coefficient, given by

$$\kappa = \pi \frac{f_p^2 \nu}{f(4\pi^2 f^2 + \nu^2)}.$$

Here f_p is the plasma frequency, and ν is the collision frequency. Usually, $4\pi f^2 \gg \nu^2$ and

$$\kappa \approx \frac{f_p^2 \nu}{4\pi f^3}.$$

It is important that $\kappa \sim N\nu/f^3$ and $K \sim f^{-2}$.

The transmitter and receiver antennas are small and relatively simple, and the required transmitter powers are small (~ 1 – 10 kW) as well. This is the first advantage of HF radio wave service. The other advantage of HF service is that it permits two-way communications at large distances of $1,000$ – $10,000$ km.

The major drawback of the HF service is that it is profoundly affected by the state of the ionosphere and the space weather, and consequently, the optimal operating frequency should be adjusted almost continuously during the day and during ionospheric storms. During major storms, the HF service can be lost at all.

At frequencies greater than 30 MHz, the communications are possible within the straight line-of-sight horizon. The distance to the radio horizon does not exceeds the value given by

$$R_{\max} \approx \sqrt{2R_E} \left(\sqrt{h_1} + \sqrt{h_2} \right),$$

where $R_E \approx 6,400$ km is the Earth's radius, h_1 is the transmitter antenna height, and h_2 is the receive antenna height. For example, when $h_1 = h_2 = 9$ m, we find that $R_{\max} \approx 21$ km. For a service with an aircraft ($h_2 \approx 10$ km), $R_{\max} \approx 358$ km, and with a low-Earth-orbit satellite ($h_2 \approx 400$ km), $R_{\max} \approx 2,260$ km.

The communications with frequencies greater than 30 MHz are relatively weakly dependent on the state of the ionosphere and space weather, because the relative refractive index of the ionospheric plasma is given by

$$n = \left(1 - \frac{f_p^2}{f^2} \right)^{1/2},$$

and when $f^2 \gg f_p^2$, it differs from unity slightly. Furthermore, the absorption coefficient $\kappa \sim f^{-3}$ rapidly decreases with increasing radio wave frequency.

Communications Based on Radio Wave Scattering

The VHF (30 – 300 MHz) radio waves can be scattered by the irregularities of the refractive index in the troposphere and the ionosphere, as well as by meteor trails. Consider them in detail.

Tropospheric Scattering. Tropospheric scattering occurs from the irregularities in the refractive index, Δn , of the troposphere. Usually, the root-mean-square (RMS) refractive index fluctuations $\sigma_n \approx 10^{-4}$ – 10^{-3} , and they do not depend on frequency. Since $\sigma_n \ll 1$, the power of the signal scattered by the tropospheric irregularities is very small. To utilize this effect for radio communication, large aperture (~ 10 m) antennas and high-power (~ 10 – 100 kW) transmitters are needed. Such a service provides radio communication over ~ 100 – $1,000$ km, while this path incurs 50 – 120 dB of loss at 100 MHz. The maximum propagation distance can be estimated from the following formula:

$$R_{\max} = 2\sqrt{2R_E h_t},$$

where h_t is the altitude of tropospheric irregularities. For example, for $h_t = 10$ – 15 km, we have $R_{\max} \approx 720$ – 880 km.

The communication based on the tropospheric scattering suffers from seasonal and climatic factors. The major drawback to this kind of service is multipath interference caused by signal fading. As a result, the bandwidth of such a communications channel is limited.

Ionospheric Scattering. Over-the-horizon communications, usually at VHF, occur due to scattering by irregularities in the ionospheric D region. The root-mean-square electron density fluctuations are given by $\widetilde{\sigma_N} = \left((\Delta N/N)^2 \right)^{1/2} \approx 10^{-2} - 3 \cdot 10^{-2}$, and the RMS refractive index fluctuations are given by

$$\sigma_n \approx \frac{1}{2} \frac{f_p^2}{f^2} \widetilde{\sigma}_N.$$

It is important that the scattered power is proportional to σ_n^2 , i.e., $\left(f_p^4 \widetilde{\sigma}_N^2\right) / f^4$. The signal power rapidly decreases with increasing frequency, for example, the signal power decreases by a factor of 100 when the frequency increases about 3 times.

In the range of 30–70 MHz frequency, the attenuation varies from –90 dB to –120 dB. The maximum propagation distance can be estimated from the following relation:

$$R_{\max} = 2\sqrt{2R_E h_i},$$

where h_i is the altitude of ionospheric irregularities. For example, $h_i \approx 90$ km yields $R_{\max} \approx 2,160$ km, and the frequency band does not exceed 6 kHz.

Electron density irregularities also occur in the ionospheric E and F regions, where they are elongated along the geomagnetic field lines. Such irregularities provide aspect angle scattering and the maximum propagation distance of $R_{\max} \approx 4,000$ km.

Meteor Communication. Meteor communication systems operate in an intermittent mode and utilize only single meteor ionization trails that appear sporadically (not meteor showers or streams). The number of such meteors entering the Earth's atmosphere per day is estimated to be $n_0 \approx 10^{11}$, while their flux $n \approx 2 \times 10^{-3} \text{ km}^{-2} \text{ s}^{-1}$. Small meteors “burn up” in the height range of $h_m \approx 80$ –120 km and form an ionization trail with 10^{10} – 10^{18} m^{-1} electron density per unit of line length. The initial density per unit of volume is $N \approx 10^{11}$ – 10^{18} m^{-3} . The trails persist for ~0.1–10 s. The time of the SNR exceeding the threshold accounts for only 1–10% of the total running time of the system. Scattering from the meteor trails depends on the aspect angle. The maximum range of the meteor channel is 2,000–2,200 km, the channel losses attain 170 dB, and the channel has an instantaneous data rate of ~100–1,000 bit/s. The meteor channels have moderate information data rates

and require moderate transmitter power (~1–10 kW) and relatively simple antennas with an antenna gain of 15–20 dB. The major drawback of meteor channels is the complexity of the system design. The International Telecommunication Union Recommendation ITU-R F.1113, radio systems employing meteor-burst propagation, at https://www.itu.int/dms_pubrec/itu-r/rec/f/R-REC-F.1113-0-199409-1!!PDF-E.pdf describes the construction of two such systems at a realistic commercial cost.

Terrestrial Relay Links

These services can be provided by both conventional and new repeaters.

Conventional Repeaters. The system requires ~1–10 m antennas, 20–30 dB antenna gains, and ~1–10 kW transmitters. The maximum propagation distance is given by the relation:

$$R_{\max} = 2\sqrt{2R_E h_r},$$

where h_r is the relay station height. For example, substituting $h_r = 10$ m, we have $R_{\max} \approx 23$ km. The small values of $R_{\max}$ are the major drawback to the conventional repeaters.

New Repeaters. Airships provide a relay station with the height of $h_r \sim 1$ km, and stratosphere balloons stable in space can attain heights of $h_r \approx 20$ –25 km. These platforms can support communications at ranges of 226 km and 1,010–1,130 km, respectively.

Cable Links

The Russian Optical Trans-Arctic Cable System, ROTACS, (<http://www.polarnetproject.net/>) project intends to connect the UK, Russia, China, and Japan with optical cable. In the first stage, six pairs of optical fibers are planned to be laid out. The first phase of work is estimated at about \$1 billion.

Countries in the Arctic region advanced the Arctic Fibre project to connect the UK, Ireland, Canada, the USA, and Japan.

Canada has built Ivaluk Network to serve the needs of Northern Canadians with an optical cable loop of 8,000 km.

Arctic Communications Satellite Links

Satellites are practical for providing communications in the Arctic regions. Moreover, this kind of communication is preferable and advanced. The satellites used for these conditions include geostationary and low-Earth-orbit satellites, as well as satellites in highly elliptical orbit. Consider these opportunities in more detail.

Geostationary Satellites. In Alaska and the USA, high-quality video and broadband services are delivered by Intelsat, Telesat, and other providers of communication satellite service, when the satellites are at an angle of 9° over the horizon.

In Arctic Canada, broadband services and Internet connections are provided by the Telesat

operator via the AnikF1R, AnikF2R, and AnikF3R satellites.

In Norway, communications and broadcasts are delivered via the Thor satellite and, in Russia, via the Express satellite.

The major drawback of communications via geostationary satellites is that they become unstable at low angles of the satellites over the horizon.

Satellites in Highly Elliptical Orbits (HEO). Such a kind of project does not exist yet. Communications through these satellites lack the advantage that geostationary satellites have. Satellites on high elliptical orbit will be repeatedly immersed in the radiation belt, which can shorten the life span of spacecraft electronics components.

Low-Earth-Orbit (LEO) Satellites. The *Iridium NEXT* constellation operates in the L-band at 2.4–132 kbps data rates (Table 1).

Gonets System (Russia) is composed of 12 satellites in 1,400 km orbit at 82.5° orbital inclination. The data rate is 2.4–64 kbps, and the frequency used 200–400 MHz.

When completed, the *Kosmonet* satellite project will consist of 48 satellites. The total cost of this project is \$0.3 billion.

Polar Communications, Table 1 An overview of Iridium services. (Data from Iridium homepage (<http://www.iridium.com>))

Service	Iridium	Iridium NEXT
Voice	2.4 kbps	2.4 kbps
Circuit switched data	2.4 kbps	9.6–64 kbps
Short burst data	low	On demand
OpenPort	132 kbps	128–512 kbps
OpenPort aero	132 kbps	128–512 kbps
L-band high speed	N/A	512 kbps up–1.5 Mbps down
Broadcast	N/A	64 Mbps

New Missions Toward the Arctic

New missions are the following: (1) Antarctic Broadband (ABB); (2) Arctic Region Communications Small Satellites (ARCSS); (3) Polar Communications and Weather (PCW); (4) Arktika (A); and (5) CASSIOPE/Cascade (CC) (Table 2).

Polar Communications, Table 2 Proposed satellite communication systems (Birkeland 2014)

System	Frequency band	Capacity	Con't coverage	Arctic coverage
ABB	Ka	TBD	No	?
ARCS	S, X, UHF	3 Mbps	No	Yes
PCW	TBD, UHF	TBD	No?	Yes
A		Mobile	Maybe	Yes
CC		Gbit	No	Yes
Telenor HEO		Broadband	Yes	Yes

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Polar Engineering Antifreeze

► Winterization of Polar Engineering

Polar Materials

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Definition

Polar materials always refer to materials used in cold climate zones. Polar materials should

withstand the effect of the polar region climate without sacrificing other performance properties. Reliability, durability, trouble-free operation, and reparability are the requirements for the application of polar materials under polar region conditions. The exploration of the polar regions is largely determined by the availability of materials that are necessary to create technical devices and facilities and can ensure comfort for people who stay and work in severe climatic conditions.

Scientific and Engineering Fundamentals

Service Environment of Polar Materials

Polar Regions

The polar regions are the regions of the planet that surround its geographical poles (the North and South Poles), lying within the polar circles. These high latitude regions are dominated by polar ice caps: the northern resting on the Arctic Ocean and the southern on the continent of Antarctica. These regions include ice and marine sites with islands and straits and a part of the coast.

The polar regions are the last unspoiled natural treasures with huge reserves of minerals and resource on earth. In recent years, polar regions, especially Arctic region, have become regions interesting to the global community due to transportation, geopolitical, military, and raw material-related reasons. Today, only a small portion of the polar regions can be developed due to the huge infrastructure costs, the required reliable equipment, and the limitations of polar materials. The ability to explore and develop polar regions depends on the materials necessary to create technical installations and facilities for severe climatic conditions in the polar regions. Developing polar materials has become one of the keys to develop polar regions and utilize the polar region resources (Buznik et al. 2018b).

The characteristic features of the polar region are extreme natural and climatic conditions such as long-term low temperatures, significant annual temperature gradients (from $-60\text{ }^{\circ}\text{C}$ to $40\text{ }^{\circ}\text{C}$),

strong winds, high humidity, glaciation and snow buildup, the presence of ice caps covering in the seas, a high solar radiation during the polar day, and a low environmental stability. The simultaneous effects of all the factors mentioned above make the polar climate especially severe and the application of polar materials especially complex and strict.

Another important characteristic feature is that the polar region environment is sensitive to anthropogenic impacts. Therefore, industrial centers and densely populated settlements are far from the polar regions; the production and transport infrastructure are weak in polar regions; people, raw materials, and equipment need to be brought from the warm continents. Even small accident in polar regions can have serious consequences for humans and the polar region environment. Hence, there are strict requirements for polar materials: polar materials should retain their functional properties as long as possible.

Glaciations and snow buildup lead to additional mechanical loads. Sea water can cause severe corrosion. Moisture provides another negative climatic impact, which initiates metal corrosion and degrades the properties and structure of polymeric composites. Under the polar region conditions, this phenomenon is aggravated by negative temperatures and repeated transitions through the melting point of ice, which accelerates corrosion. The most important is that the ships used in polar regions and the related offshore structures may face risks associated with accidents such as iceberg collisions. Obviously, high-quality polar materials should withstand all the severe climate mentioned above, not solely individually or some of them.

Characteristic Features of Polar Materials

Cold is the main climatic characteristic feature of the polar regions. Long-term service in polar region low temperatures is the biggest challenge for polar materials. However, the impact of cold is not limited to geographical positions; one can identify many of the more severe cold continental areas. So polar materials always refer to materials used in cold climate zones. And polar materials should withstand the effect of the polar region

climate without sacrificing other performance properties. The important requirements on polar material are their reliability, durability, trouble-free operation, and reparability under field conditions. Therefore, structures used in polar regions are more rigorous, more precise, and more discreet than structures used in warmer regions.

Development History of Polar Materials

At the initial stages of the exploration and research of polar regions, natural materials (wood, wool, leather, coal, ice, and so on) were the main materials for the production of vessel, clothes, household, and building materials. However, the properties of these natural materials are very limited, which is not enough to further develop and explore the polar regions. Hence, from the nineteenth century, anthropogenic materials have replaced natural materials to become the most used materials for producing transport, household, and products for everyday life in polar regions. Recently, the performance requirements for polar materials increase rapidly with the requirements for exploring, utilizing, and developing polar regions. Today, there are two major ways to develop polar materials. The first is the specialized development of materials, and the second is adaptation of the existing materials, giving them additional performance for effective use in the polar regions (Buznik et al. 2017, 2018a, b).

Polar Steels

Steel is the main modern construction material. However, it is the most vulnerable material in the polar regions; the vessels used in polar region waterways and the related offshore structures are exposed to risks of brittle fractures due to accidents such as collisions with icebergs. The brittle fracture of steel in polar environments has caused many serious accidents. Moreover, the steels always manifest frost fracture under low temperatures, leading to the destruction of technical facilities. This phenomenon can be attributed to many factors: external impacts, traits of crystalline phases of iron, the microstructures, the presence of inclusions and textures in steels, etc. (Sychev et al. 2017).

Improving the toughness of steels under low-temperature conditions and preventing the occurrence of brittle fracture have become the main challenges for conventional steels used as polar materials. At present, the special alloying and thermomechanical processing have been used to solve the technical problem for alleviating the brittle fracture and frost cracking and meet the operation requirements in polar regions. The newly developed steels have already been tested in building pipelines, vessels, and the offshore oil platform. In addition, searches have conducted for techniques that would ensure an optimal combination of functional and economic indicators of cold-resistant steels. The properties of steels should be examined carefully and extensively before choosing the certain steel grade.

Carbon Steels Carbon steel is the most commonly used structural material in the world. They are cost-efficient in most applications and have good stiffness, high strength, high durability, and recyclability. Since the ductile-brittle transition temperature of carbon steel is generally between room temperature and -40°C , it must be strictly required for steel grade selection, manufacture, assembly, and inspection, when carbon steel is used in polar regions.

In general, the polar carbon steels can be divided into four different groups as normal strength steel, high strength steel, extra high strength steel, and ultrahigh strength steel.

The normal strength steels (e.g., P215NL pipe steel and grade E steel and steel EW steel certified by classification societies) have yield strengths up to 235 MPa, which are used in pipelines, offshore, and inland applications. For example, P215NL and grade E steel are tough at the temperature of -40°C , which suggests that these steels can be used in few polar areas. Grade EW steel has the transverse impact test energy up to 40 J at temperature of -40°C , and its weldability was improved. These suggest that it is more suitable for welding applications than the other two kinds of steel.

The high strength steels (e.g., pipeline steels L245-L390, 11MnNi5-3, 13MnNi6-3, 12Ni14, X12Ni5) have yield strengths exceeding

235 MPa up to 400 MPa. Their ductile-brittle transition temperatures reach -40°C or below. These steels are used in offshore pipelines, offshore fixed applications, and offshore vessels. 11MnNi5-3 and 13MnNi6-3 could work as offshore pipeline steel at -60°C . The Charpy-V impact energy 12Ni14 and X12Ni5 reach 40 J at -100°C and -120°C , respectively. Pipeline steel L320 and L360 could operate about 30 years as gas main.

The extra strength steels (e.g., E500, F420-690, X10Ni9 and 26CrMo4-2) have yield strength exceeding 400 MPa up to 700 MPa, and their transition temperatures also reach -40°C or below. Many steels in this group were used in nonredundant structures in polar regions around -60°C (Pavel et al. 2016).

The ultrahigh strength steels (e.g., L830, S890QL, S890QL1, and S960QL) have yield strength exceeding 700 MPa, and their transition temperatures reach -40°C or below. The pipeline steel L830 or X120 is standardized with impact energy of above 250 J at -30°C .

Polar Alloy Steels Polar alloy steels were also developed for offshore or general inland structural application. Steel 09Г2С (09Mn2Si), steel 10Г2, steel 10Г2С1, and steel 16ГС were developed for operating at pole region environments with temperature as low as -70°C in Russia (Panin et al. 2017). For example, steel 14Г2АФ is a ferrite-pearlite-based high-strength alloy steel with low corrosion resistance. It has yield strength and fracture toughness parameter of Charpy U impact of 540 MPa and 110 J/cm^2 at -60°C , respectively. Therefore it is recommended for manufacturing vessels and machinery operated at the temperature range of -50°C to 400°C .

High-strength low-alloy steels have good weldability and high yield strength (even over 1000 MPa). For example, steel 18Г2Ф is kind of high-strength low-alloy steel developed by Russian. Its yield strength and fracture toughness parameter of Charpy U impact test could reach 440 MPa and 29 J/cm^2 at -70°C , respectively. Therefore, it can work at variable temperatures ranging from -60°C to 450°C .

Polar Stainless Steels Polar stainless steels are typically used in applications where good corrosion resistance and low-temperature mechanical properties are obligatory. For example, they are used in cargo tanks, storage tanks, shafts, and pressure vessels, especially the pipelines. Polar stainless steels are often categorized to four different main groups: austenitic, ferritic, austenitic-ferritic (duplex), and martensitic polar stainless steels. Now, several stainless steels were developed in structures and pipes at low temperatures of polar regions. For example, AISI 304, 304L, 316, 316L, 321, and 347 can be used in offshore applications.

Polar Steel Welding The welding of polar steel is the most important factor to determine whether the steels can be used in polar region as modern structures, vessels, pipelines, and offshore (Pavel et al. 2018). Modern structures and vessels require the thick-welded steel plates with yield strengths of 690 MPa or more and exceeding plasticity and toughness. In the recent years, some high-strength cold-resistant steels with high plasticity and satisfactory weldability were developed for using as welded parts in the polar region conditions of -20°C . However, some irreversible structural changes are likely to occur due to the influence of the welding thermal cycle during the welding process, which reduces the cold resistance of the welding heat-affected zone in the heat-affected zone. Therefore, the welding techniques become a key problem for using these high-strength cold-resistant steels in polar regions. Research shows that reduction of the carbon content and alloying level could increase the mechanical properties of welding joint and lower the HAZ toughness, which implies that the development of high-strength low-alloy steel with good weldability and low-temperature mechanical properties may be an important way to obtain high-performance polar steels in the future.

Testing Methods of Polar Steels Steel and welded constructions can rupture rapidly under effect of low temperatures of polar regions. This phenomenon is due to the existence of the ductile-brittle transition temperature below which the

steels are brittle. The welding defects and fatigue cracks can occur unexpectedly in the stress-concentrated areas under low loads in polar regions.

Therefore, good toughness (impact toughness, fracture toughness, crack arrest toughness, and so on) is a desirable character for polar steels used as structural material. Several test methods were developed to analyze the toughness of metals; most common methods are Charpy-V impact, crack tip opening displacement (CTOD) or crack tip opening angle (CTOA), and Nil-ductility temperature (NDT) or drop-weight test (DT) or drop weight tear test (DWTT). Recently, temperature gradient type ESSO test and temperature gradient type double-tension test are developed for evaluating the brittle crack arrest toughness (Shimada et al. 2017).

By far the most common toughness test method is the Charpy-V impact test. As a traditional toughness test method, the Charpy-V impact test is easy to operate and clearly shows the toughness of traditional materials. The Charpy-V impact test uses a pendulum with certain weight and speed to impact a standard sized sample. After the collision, the energy absorbed by the sample is recorded to obtain the notch toughness of the test material.

The CTOD test was performed by placing the sample in a three-point bending machine and measuring the opening of the crack after applying the load. It obtains the stress intensity factor at the crack tip by calculating the open displacement near the crack tip and using the displacement extrapolation interpolation method. It has become more important in recent years for the CTOD test to have the ability to use a full-size specimen with a crack under the stress corresponding to the load actually used.

The DWTT is a method for evaluating the metallurgical quality and brittle fracture of test sample. It measures the impact energy absorbed in the test sample and observes the characteristics of metallurgical defects, fracture properties, and morphology. It can be used to compare welded parts and test “Nil-ductility temperature.” The “Nil-ductility temperature” is obtained by testing multiple samples at different

gradually decreasing temperatures until the specimen is broken.

Nonferrous Metallic Materials

Beside steels, many other metallic materials are also used in polar regions (Buznik et al. 2018a, b; Buznik and Kablov 2017). Among them, aluminum alloys are widely used in offshore structures for their lightweight and good corrosion resistance even in presence of sea water, such as deck structures. The other advantage of aluminum alloys used in polar region is that they are not suffered from the ductile-brittle transition at low temperature. In general, 5xxx and 6xxx series aluminum alloy, with different tempers, are classified for offshore use. 5xxx series products are valid in forms of sheet, strip, and plate and 6xxx series for extruded products. Other metals and alloys, such as titanium alloys, manganese alloys, and copper-based alloys, were also used for constructions and facilities and structural frames in polar regions.

High Molecular Compounds

The great interest in the applications of high molecular compounds in polar region is due to their great chemical diversity and broad spectrum as construction, functional, and auxiliary materials. High molecular materials, including oligomers, thermoplastics, thermosets, elastomers, etc., have been widely applied to produce products used in polar regions, such as glues, sealants, coatings, fillers, anti-icing coatings, hydrophobic polymer and oligomer materials, frost- and fire-resistant rubbers, weatherproofs, hydrophobic, and tribological coatings. High molecular materials are also used for low-temperature oil- and gasoline-resistant elastomers, ultrahigh molecular weight polyethylene with a high toughness and crack resistance, and high-performance radiation-modified polytetrafluoroethylene and polymer electrolytes. Low temperatures are harmful for high molecular compounds: polymers and rubbers vitrify, crystallize, and lose their serviceability and destruction (Petrova et al. 2016). Various fillers are being sought actively to increase the cold resistance of materials within acceptable cost.

Ceramics Materials

Ceramics materials, including glasses and cements, have been used to produce the heat-resistant glasses (used for alarm systems and navigation systems of rescue vessels), various coatings, porous fire-resistant heat-insulating materials, electrode materials for chemical current supply sources operating at low temperatures, and radiation-resistant ceramic materials.

Composites

Metallic, ceramic, and polymeric composite materials have been used to produce structural frames, composite protector hydro- and ice-phobic coatings, high-strength hydrophobic CMs with a low coefficient of friction, heat and noise insulation, and composite paint-and-lacquer systems.

Other Materials

In addition to the above materials, natural materials were also developed and used in polar regions (Schulson 2015), such as ice, liquid fuels and low-temperature lubricants, electrochromic and heatproof coatings for glasses, frost-resistant textile materials, etc.

Cross-References

- [Ice Breaking Vessel](#)
- [Polar Offshore Engineering](#)

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Polar Merchant Ship

► Polar Merchant Vessel

Polar Merchant Vessel

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Synonyms

Polar merchant ship; Polar merchantman; Polar trading vessel

Definition

The polar merchant vessel is a boat or ship sailing in the polar region in order to transporting cargo or

carrying passengers for hire. This excludes pleasure craft that do not carry passengers for hire. Warships are also excluded. As to the unique natural environment of the polar region, some icebreakers are also belonged to this region.

The Reason for the Emergence of Polar Merchant Ships

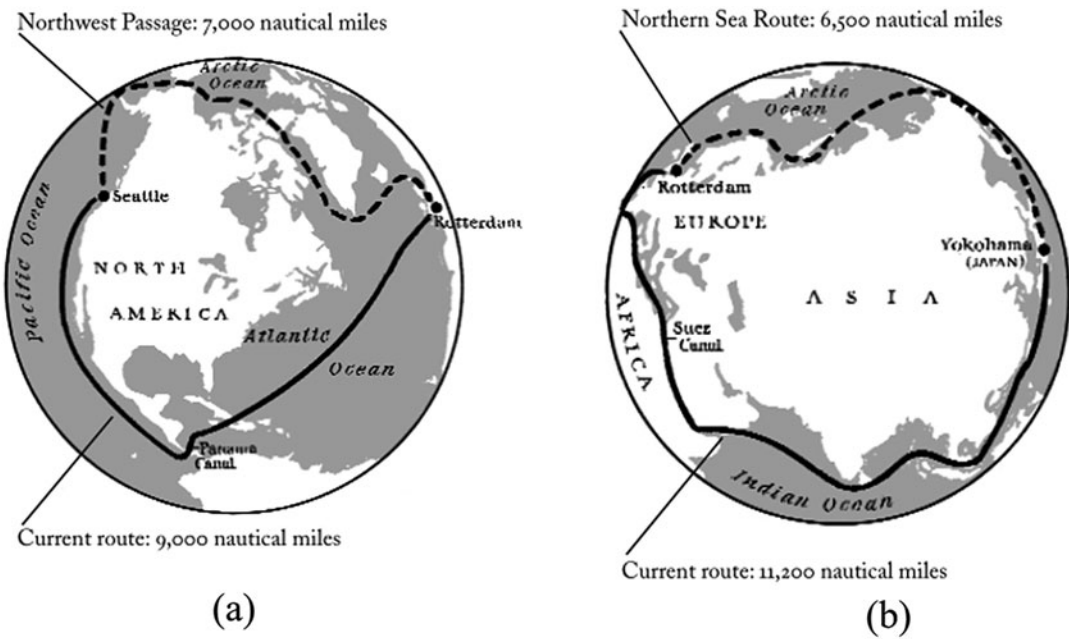
With the continuous exploration of the polar region, the abundant resources of the polar regions are gradually known by human beings. The arctic alone holds 30% of the world's undiscovered gas reserves and 13% of the world's undiscovered oil reserves, according to resource exploration (Gautier et al. 2009). Besides, the arctic is rich in beryllium, indium, niobium, platinum, graphite, rare earth and other minerals. In addition, by using the arctic route, the distance between East Asia, Europe, and North America will be greatly shortened, as shown in Fig. 1. For example, the distance from Rotterdam to Yokohama via the Suez Canal is 20,742.40 km. If it goes through the northeast channel, its distance is only 12,038 km, which is reduced by 40%. Likewise, from Seattle to Rotterdam, the northwest passage is 25% shorter than the panama canal route (Borgerson 2008).

Considering the cost of canals, fuel, and so on, a single trip would save nearly a million dollars in transportation costs. More and more people's yearning for polar scenery has also promoted the development of polar mail ship.

Polar Regulations and Influence of Polar Merchant Vessel

It is we all known that the geographical environment of the polar sea is different from that of other sea. Cold water, lots of floating ice, and icebergs make it extremely difficult for polar merchant vessel. Therefore, polar ships often need some unique design and navigation routes.

In 2006 IACS has issued a unified requirement for polar Marine, *Requirements Concerning Polar Class*, which is formulated by the nation



Polar Merchant Vessel, Fig. 1 Illustrations for Northwest and current routes (a) from Rotterdam to Seattle; (b) from Rotterdam to Yokohama (Borgerson 2008)

classification societies jointly. The requirement for the design and construction of the polar ships provides a unified standard on the hull structure, machinery, etc. Using additional signs, PC1 ~PC7, to show polar classification (IACS 2011). This requirement should be used by major classification societies in their classification codes as a basis of classification for polar ships. However, this requirement only provides indicative requirements, which cannot be used as a guide document for polar ship design. Therefore, the implementation and verification levels of the requirements vary from country to country.

International maritime organization (IMO) issued a guide *International Code for Ships Operating in Polar Waters* in December 2002 (IMO 2013). The main contents include: hull structure, subdivision, stability, electrical equipment, fire safety, rescue device, emergency equipment, environmental protection, and so on. Indeed, it is suitable for vessels in the arctic waters. On the base of the code, ships sailing in Antarctic waters are also supplemented. The *Polar Code* was adopted in 2014 as a mandatory rule for polar

sailing ships, which took effect on January 1, 2017.

In terms of the typical design of ice ships, the ship's plate and deck which is closed to the outside should avoid cold brittle damage at low temperature. In order to restraining the expansion due to ballast water translating to fresh water ice, ballast tank and some similar structures in polar ship should have the appropriate excess volume (10%). Especially, the piping system (including the air permeability pipe) serving the tank should prevent being frozen.

Polar Merchant Ship Categories

Polar merchant vessels can be divided into polar bulk carrier, polar container ship, polar tanker, and polar cruise ship. Based on the special environment of the polar regions, polar tanker accounts for a significant proportion of polar ship. Moreover, Polar cruise vessels are mainly used for sightseeing.

Polar Bulk Carrier

Bulker carrier is a merchant ship specially designed to transport unpackaged bulk cargo, such as grains, coal, ore, and cement, in its cargo holds. Today’s bulk carriers are specially designed to maximize capacity, safety, efficiency, and durability. Polar bulk carrier should pay attention to the cold resistance of their cargo and their own thermal insulation.

Yong Sheng

“Yong Sheng” is a polar bulk carrier of Chinese COSCO shipping company, which is shown in Fig. 2. It started from Tai Cang port in 2013 and reached Rotterdam, the Netherlands, via the arctic route. It is the beginning of the arctic voyages of Chinese merchant ships. From then on, Yong Sheng crossed the arctic route four times from 2014 to 2017 and arrived in Europe from China. During this period, several vessels belonged to Chinese COSCO shipping company undertook more than ten commercial voyages.

Main Dimension of Yong Sheng Ship

Classification	LR, +100 A1, Ice Class 1A Strengthened for heavy cargoes Equipped with container securing arrangement
GRT/NRT	14,357/6,985 mt
Deadweight (summer)	19,561 mt

Max draft (summer)	8.42 m
LOA	159.99 m
Breadth	23.70 m
Depth to main deck	11.95 m
Height above keel	41.10 m
Max speed	14.80 kn

Polar Container Ship

Container ships are cargo ships that carry all of their loads in truck-size intermodal containers, professionally called containerization. They are a common means of commercial intermodal freight transport and now carry most seagoing nonbulk cargo. So far, polar container ships have not been the mainstream of polar merchant Vessel. In other words, polar container ships do not have enough travel times as much as polar oil tankers and polar bulk carriers. Zapolyarniy, a polar container ship, sailed between Shanghai, China, and Dudinka, Russia, sailing through parts of the northeast passage.

Polar Tanker

A tanker is a ship designed to transport or store liquids or gases in bulk. Due to the abundant natural gas and oil resources in the polar and circumpolar regions, polar tanker accounts for the largest proportion of polar merchant vessel.



Polar Merchant Vessel, Fig. 2 Yong Sheng ship (<http://www.coscol.com.cn>)

UIKKU

In 1997, UIKKU became the first merchant ship under non-Soviet flag to navigate the entire Northern Sea Route, which is shown in Fig. 3. In the following year, she took part in Arctic Demonstration and Exploratory Voyage (ARCDEV), a research project funded by the European Union to determine the feasibility of year-around navigation in the Northern Sea Route (Transport Research Knowledge Center 2003). UIKKU accompanied by a Russian nuclear-powered icebreaker NS Rossiya to open the way and a research icebreaker Kapitan Dranitsyn to provide facilities to 70 researchers from different countries carried a cargo of gas condensate from Ob river estuary to Europe.

General Characteristics (Russian Maritime Register of Shipping)

Tonnage:	11,290 GT 4,937 NT 16,038 DWT
Displacement:	22,654 tons
Length:	164.40 m (539.37 ft) (overall)
Beam:	22.22 m (72.90 ft)
Draught:	9.55 m (31.33 ft)
Depth:	12.00 m (39.37 ft)
Ice class:	1A Super RMRS UL
Installed power:	2 × Wärtsilä Vasa 12V32E (2 × 4,920 kW)

1 × Wärtsilä Vasa 12V22D-HF (1950 kW)

Propulsion:	Diesel electric propulsion 11.4 MW Azipod unit
Speed:	17 knots (31 km/h; 20 mph)
Capacity:	8 cargo tanks, 16,215 m ³ (98%)

Polar Targeting Design

In order to adapt to arctic navigation, people have made a series of changes to UIKKU. Designers have given UIKKU so much power that tankers usually only need to keep one of the two engines working. In addition, hull structures being similar to icebreakers are given to UIKKU. To reduce the friction between the hull and the ice, the ships were also equipped with an air bubbling system (Nathan et al. 1996). Besides, according to IMCO’s nonmandatory requirements, the hull was designed with double hulls to avoid grounding (Gallin et al. 1981).

New Polar Tanker – Double Acting Shuttle Tanker

A shuttle tanker is a ship designed for oil transport from an off-shore oil field as an alternative to constructing oil pipelines. Both double acting shuttle tanker’s bow and her stern can break the ice continuously. It implements that the sailing bow forward is suitable for open water, and the sailing stern forward is suitable for ice. It settles an

Polar Merchant Vessel,

Fig. 3 Towing of tanker UIKKU By icebreaker (From Final public report of the ARCDEV project)



economic problem which results from the difference of sailing environment.

We should be clear that making the ice broken is not a simply collision when the double acting shuttle tanker sails stern forward, but using full-revolving propulsor breaks the ice. In this way, we can collapse the ice and at the same time also rotate the ship stern, broaden the channel. However, it is a slow process.

Timofey Guzhenko

She is completed at 24 February 2009, as shown in Fig. 4. Her sister ship Vasily Dinkov delivered 2007 is the first double acting shuttle tanker.

General Characteristics

Type: Shuttle tanker

Tonnage: 49,597 GT 21,075 NT 72,722 DWT

Displacement: 93,515 tons

Length: 257 m (843 ft)

Beam: 34 m (112 ft)

Draught: 14.2 m (47 ft)

Depth: 21 m (69 ft)

Ice class: RMRS Arc6

Installed power: $2 \times$ Wärtsilä 16V38B

($2 \times 11,600$ kW)

Wärtsilä 6L38B (4,350 kW)

Propulsion: Diesel-electric two ABB Azipod

units (2×10 MW)

Speed: 15.7 knots (29.1 km/h; 18.1 mph)

Polar LNG Carrier

An LNG carrier is a tank ship designed for transporting liquefied natural gas. Likewise, the polar LNG tanker is a kind of polar tanker. Because of the polar region and its environs owning huge gas reserves, LNG ship is becoming a super star of polar merchant ships with the launching of the Russia Yamal project.

Christophe De Margerie

Christophe De Margerie, as shown in Fig. 5, made its maiden voyage in mid-august 2017. Its owner is Russia's shipping company, Sovcomflot. In one voyage, she can carry $172,600 \text{ m}^3$ of LNG.

The ship is 299 m long, 50 m wide and has a draft of 8.42 m. The ship loaded LNG from Hammerfest Norway on August 6, in the absence of icebreaker's escort, going through the north pole route, across the Bering strait, arriving in South Korea on August 17th. The voyage set a new record on the shortest route across the arctic. In addition, 14 polar LNG carriers will be built for Russia's Yamal LNG project. This will help customers in Japan, South Korea, and China to speed up cargo turnover and help develop polar route in the arctic.

As far as we know, her ice-class was assigned as Arc7, the highest ice class among existing polar LNG carrier. The ship is able to go through ice area independently until the ice is over 2.1 m thick. It is attribute to her bow and stern are

Polar Merchant Vessel,

Fig. 4 Double acting shuttle tanker Timofey Guzhenko (<https://rs-class.org/>)



Polar Merchant Vessel,
Fig. 5 Polar LNG carrier
Christophe De Margerie
(<http://www.sovcomflot.ru/>)



covered with 70 mm steel and can withstand -52°C . Furthermore, strong motivation also guarantees that she can travel in the ice area. The total power of this vessel’s propulsion system is 45 MW. Compared to the world’s first nuclear-powered ice-breaker, Lenin, owning 32.4 MW propulsion system, only accounts for around two thirds of Christophe De Margerie’s power.

Polar Cruise Ship

Unlike most polar merchant ships, polar cruise vessel often is designed as low ice-class ship. Some of them only stiffen their bow simply, and some of them are modified by scientific research ships. There are two main reasons lead to this phenomenon. Firstly, polar cruise ships choose to sail in summer usually, and their routes are safer than others’. Moreover, polar environmental groups, especially Antarctic ones, have imposed strict controls on commercial shipping to protect the polar environment. For example, International Association of Antarctica Tour Operators (IAATO) divides Antarctic travel vessels into four categories depended on the number of passengers. The number of tourists is from little to large, followed by YA, C1, C2, and CR, respectively. Carrying the more tourists, received the more restrictions.

Ushuaia

Ushuaia is a cruise ship operated by Argentina’s Antarpplly Expeditions, which is shown in Fig. 6. She was built for the National Oceanic and Atmospheric Administration. She served the NOAA for 20 years under the names “Researcher” and “Malcolm Baldrige.”

CR- Prinsendam

General characteristics (shown in Fig. 7)

Length	427 ft (130 m)
Capacity	350 passengers
Crew	200

C2-Ocean Diamond

Ocean Diamond is a cruise ship operated by Quark Expeditions, which is shown in Fig. 8. She was previously named Song of Flower. Many kinds of entertainment facilities on board are ready.

General Characteristics

Tonnage: 8,282 GRT	Displacement: 3,433 DWT
Length: 124.19 m (407.4 ft)	Beam: 16.03 m (52.6 ft)
Decks: 5	Ice class: 1D
	Speed: 15.5 knots
Installed power: 2 Wichmann Engines, 7375 horsepower	



Polar Merchant Vessel, Fig. 6 Polar cruise ship Ushuaia (<https://antarply.com/>)

Polar Merchant Vessel, Fig. 7 Polar cruise ship CR- Prinsendam



Le Lyrial

With the increasing popularity of Antarctic travel in recent years, more and more shipping companies are building polar exploration cruise ships specially. These designed polar cruise ships offer more luxurious facilities on board. This makes Antarctic travel not only an adventure, but also an unparalleled enjoyment elsewhere. Le Lyrial is a new cruise ship built by Fincantieri in Ancona, Italy, as shown in Fig. 9.

50 Let Pobedy

There is a special kind of polar cruise ship whose customers are enthusiastic explorers. These explorers' pursuing is nature's pole and the limit

of human will. This kind of polar mail ship's mission tends to be held by heavy icebreakers. The Arktika grade atomic icebreaker, 50 Let Pobedy, is the most famous ship, which is shown in Fig. 10.

Since 1989 the nuclear-powered icebreakers have also been used for tourist purposes carrying passengers to the North Pole. In 2008 Quark Expeditions hired this ship to finish expeditions to the North Pole. In October 2013, the vessel carried the Olympic Flame to the North Pole, in the run-up to the 2014 Winter Olympics. In August 2017, Let Pobedy set a new record for transit time to the North Pole.

Polar Merchant Vessel,

Fig. 8 Polar cruise ship
Prinsendam and Ocean
Diamond (http://www.sohu.com/a/192993268_606811)

**Polar Merchant Vessel,**

Fig. 9 Polar cruise ship Le
Lyrial (<https://us.ponant.com>)

**Polar Merchant Vessel,**

Fig. 10 Polar cruise ship
Let Pobedy (https://www.sohu.com/a/107182071_383632)



Main Characteristics

Tonnage: 23,439 GT 3,505 DWT
 Length: 159.6 m (524 ft)
 Beam: 30 m (98 ft)
 Draught: 11 m (36 ft)
 Depth: 17.2 m (56 ft)
 Endurance: 7.5 months
 Crew: 189
 Installed power: Two OK-900A nuclear reactors (2×171 MW)
 Two steam turbo generators (2×27.6 MW)
 Speed: 18.6 knots (34.4 km/h; 21.4 mph) (maximum)

Cross-References

- [Ice Breaking Vessel](#)
- [Navigation of Polar Vessel](#)
- [Special Marine Vehicle](#)

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Polar Offshore Engineering

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Synonyms

[Floating production storage and offloading \(FPSO\)](#); [Health, safety, and environmental management system \(HSE\)](#); [Mobile offshore drilling unit \(MODU\)](#); [Semisubmersible drilling and production rig \(SSDP\)](#); [The Global Maritime Distress and Safety System \(GMDSS\)](#)

Definition

Polar offshore engineering refers to the equipment and technology to carry out scientific investigation, commercial navigation, oil and gas resources development, and tourism and leisure activities in polar regions. It is an important carrier to understand, develop, and utilize polar regions. At present, the international polar equipment is mainly divided into polar scientific equipment, polar ship equipment, polar resource development equipment, and so on.

Introduction

In general, the polar offshore engineering is such kind of engineering operated in the Arctic including exploration, development, and utilization of resources in the polar ice area, as well as the detection, protection, or recovery of the polar ocean environment.

The Significance of the Polar Offshore Engineering Development

The arctic region is rich in oil, gas, minerals, and fishery resources. With global warming and the arctic glacier melting, the arctic route has started to become the most efficient energy and materials

Polar Merchantman

- [Polar Merchant Vessel](#)

channel which connects Europe, Asia, and America. Because polar ocean environment is unique and fragile, environment protection of the polar ocean is significant for the whole human beings.

However, human activities, such as commercial activities, scientific investigation, exploitation of resources, adventure tourism, and fishing in the Arctic which is a remote area with harsh climate (low temperature, strong wind, huge waves, floating ice, etc.) and fragile environment, rely on economical, safe, and environmental polar offshore engineering equipment which are suitable for the polar environment.

The polar offshore engineering equipment are important tools and platform for us to explore, understand, protect, and exploit the polar region. The polar offshore engineering is an important subject that studies and solves the key technologies involved in the whole life cycle of research, design, construction, operation, and discarding of the polar offshore engineering equipment.

Key Technologies of Polar Offshore Engineering

- The Environment and Load of Ice in Polar Region

In extreme environment, the impact of ice floe on the structure of the polar marine equipment is significant. The perennial floating ice will cause continuous and periodic extrusion and influence on the structure of the offshore platform. The influence will cause great ice load and possibly form intense ice exciting vibration.

How to confirm the proper design ice load is a challenging technical problem. On the one hand, the work experience of the marine engineering in the polar environment is limited. On the other hand, the interaction between ice and the platform is complex, as well as the physical mechanism of the ice breakage.

In order to meet the customer's demand of ice load forecasting for marine equipment,

ABS has explored a series of new methods for numerical simulation of ice load by cooperation with universities and enterprises in recent years. For example, they use the particle-in-cell method to predict the ice load of the dispersed floating ice on the sea structure, the discrete element (DEM) method to explore the consequences of the ice load produced by the whole ice floe on the sea structure, and the traditional finite element (FEA) method to simulate the ice load with the special physical properties of the ice.

- Cold and Ice Prevention of Mechanical Systems, Electrical Systems, and Equipment

Based on the particularity of the low temperature of the polar environment and the influence of different temperature on the equipment and the function of the system at low temperature, the design and use requirements of the core equipment and system at different temperature grades are selected according to the temperature grade of the low-temperature environment.

According to the low-temperature conditions of the polar regions, the reasons for the formation of the platform ice in the low-temperature environment, the antifreezing and protection of the equipment in the open area, the indoor antifreeze, and the deicing of the platform are studied to ensure the safety of the polar operating platform and the operation of the equipment.

- Development and Application of Low-Temperature Materials

Low temperature is another major challenge for polar offshore operations. Extreme low temperature will not only test the physical condition of the staff on the marine equipment but also have a great impact on the structure, equipment, and materials of the marine equipment. As we acknowledge, winter of the Arctic region usually starts from November to April next year. The average temperature in January is -20°C to -40°C , and the average temperature in the warmest month is only -8°C in August.

The toughness of ordinary materials is decreased in the low-temperature environment,

and the possibility of brittle failure is greatly increased. The brittle failure is difficult to find out in the early stage, so it is more likely to cause serious consequences, especially in the extreme low-temperature sea conditions of the polar region. The safety problems such as material fracture, pipe and valve plugging, control system failure, and so on should be paid more attention. Therefore, research and development of low-temperature-resistant materials will become the focus of future development.

- Communication and Navigation Technology in Polar Regions

The global maritime distress and safety system (GMDSS) has been listed as a special A4 sea area because the polar regions cannot be actively covered by INMARSAT's satellites due to its far distance. In addition, the electromagnetic environment of the polar region is more complex than that of the Earth's magnetic field. Moreover, the development of polar resources depends on the development of underwater communication under ice, which is also a key technology for polar marine engineering equipment.

- Ocean Environmental Protection, Monitoring Technology, and Standards in Polar Region

The problem of polar environmental protection is a serious problem. In order to protect the fragile ecological environment in the polar region and reduce the chain reaction caused by the development of the polar ocean as far as possible, the International Maritime Organization and other organizations have established the "polar rules" and formally entered into force in January 1, 2017. In spite of this, some scholars believe that the existing pollution prevention measures in the rules are not sufficient to protect the special ecological environment of the polar region. In order to further improve the pollution prevention measures of the rules, the specific provisions and proposals are deeply analyzed with the method of fish bone map analysis and comparative analysis, and the corresponding revisions are put forward suggestions on this basis.

Main Equipment of Polar Offshore Engineering

In the polar, the temperature is very low, and the climatic condition is very poor; polar offshore engineering development is facing great challenges. The main offshore engineering equipment currently operating in the polar field includes drilling platforms, FPSOs, drilling vessels, supply vessels, subsea production facilities, and so on.

1. Polar Icebreaker

At present, the countries with polar icebreakers in the world are mainly distributed in the near polar region, including Russia, Finland, Canada, the United States, China, and so on. Among them, Russia is the country with the most polar icebreakers.

- "NORTH" Level Nuclear-Powered Icebreaker (22220 Type)

For the icebreaker as shown in Fig. 1, the length is 170.3 m, the width is 34 m, and the draft is 10.5 m. It is equipped with two new generations of RTM-200 nuclear reactors. Each reactor has a power of 170 MW and can break 3 m of ice. The icebreaker joined the Russian nuclear fleet in December 2017. It can travel in the shallow waters of the Arctic Ocean or in the deep waters and can guarantee the passage of ships with a load of 100,000 tons.

2. Polar Drilling Equipment

At present, there are relatively few drilling platforms specially aimed at the polar operations; those that have been put into use are from Norway and Russia, and some of the platforms are made in China. The details are as follows in Fig. 2:

- CS50

CS50 is a semisubmersible rig which is registered in Russian Maritime Register of Shipping. For the operation areas in the Barents Sea and in the Kara Sea, drift ice on the sea may occasionally occur. The SMODU shall be designed to operate in occasional first-year drifting thin ice. However, the SMODU shall not have any ice-class notation. The length of pontoon

Polar Offshore Engineering,

Fig. 1 “50th anniversary of victory.” (Image from the network: http://www.sohu.com/a/190672530_99968045)



Polar Offshore Engineering,

Fig. 2 Russian offshore – CS50. (Courtesy of Yantai CIMC Raffles Offshore Ltd)



is 118.56 m. The breadth of pontoon is 17.20 m. The spacing of pontoon is 58.00 m. The length of deck is 84.48 m. The breadth of deck is 72.72 m.

- North Dragon

GM4D North dragon is a semisubmersible drilling rig (the unit) built by YANTAI CIMC Raffles. The rig is equipped and suitable for year-round operations worldwide, inclusive operation on Norwegian Continental Shelf (NCS). Therefore, the design process and

the technical facilities of the unit shall comply with relevant Norwegian shelf rules (PSA Regulation). The pontoon length of the rig is 106.75 m. The breadth of pontoons is 16.20 m. The length of deck is 88.15 m. The breadth of deck is 73.10 m. The height from the bottom of pontoon to double bottom of deck is 42 m, as shown in Fig. 3.

- Cat-I Polar Drilling Ship

Norwegian Inocean's Cat-I drilling ship concept design for Statoil, based on

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Fig. 3 GM4D – North Dragon. (Courtesy of Yantai CIMC Raffles Offshore Ltd)



Polar Offshore Engineering, Fig. 4 Cat-I Polar Drilling Ship. (Image from the network: <http://www.offshorelm.com/cn/>)



its INO-80 drilling ship, added cold-resistant devices, etc., which is designed to be suitable for the work in the polar area, which minimizes the impact on the environment and can maximize the concern of the health, safety, and environmental management system (HSE), as shown in Fig. 4.

The ship length is 232 m, the width is 40 m, and the depth is 19 m. The working displacement is 89,800 tons, and the effective load is 22,400 tons. The hull of the ship is reinforced for passing through the ice zone. It can resist the 16-m ice ridge when it is positioned by dynamics and can resist the 8-m underwater ice when anchoring.

Polar Offshore Engineering,

Fig. 5 “Sevan 1000”

FPSO 平 (https://www.eniday.com/en/technology_en/goliat-norway-offshore-oil-production/)



- “Sevan 1000” FPSO

The “Sevan 1000” FPSO was designed by Sevan Marine in 2013. The FPSO is specifically designed for Eni’s Goliat oil field in the Barents Sea in the northern Arctic of northern Norway. The climate is cold, and there is a need for better performance FPSO. The equipment was built by Hyundai Ulsan Shipyard in Korea and completed in 2015, as shown in Fig. 5.

Compared with conventional ship types, the cylindrical structure can better withstand the ice drift pressure in different drift directions while avoiding the drift ice floes causing damage to the semisubmersible hull. The cylindrical platform lower space can be used to store more oil than a typical platform unit.

Polar Offshore Engineering Development Trend

At present, most countries are actively participated in the establishment of international polar

rules and accelerate the study of polar rules and related supporting equipment.

Independent research and development of heavy icebreakers, polar drilling equipment, polar scientific research equipment, polar expedition travel equipment, and related technologies to meet the strategic needs of polar seas and the construction of ice pool laboratory, low-temperature material laboratory, low-temperature electromechanical system verification experiment, the laboratory verification conditions, and the actual ship verification system are the key development directions of the major base equipment strength countries.

In addition, we should energetically cultivate the experienced polar ship handling and management personnel, realize the safe and reliable operation of the equipment in the polar sea area, fully verify the guidelines for the operation of the polar navigation, and continue to improve.

At the same time, we should speed up the development of meteorological and insurance services in our Arctic waterway so that the polar region can truly become a “treasure trough” and a “golden waterway.”

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Polar Propulsion

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Nomenclature

V	Velocity of ship
n	Propeller rotational speed
D	Propeller diameter
r	Radius of the section
t	Time
ϕ_r	Pitch angle of the section
F	Force
M	Moment
F_x	Force in the direction of x-axis
F_y	Force in the direction of y-axis
F_z	Force in the direction of z-axis
M_x	Moment in the direction of x-axis
M_y	Moment in the direction of y-axis
M_z	Moment in the direction of z-axis
EAR	Disc area ratio
Z	Amount of blades
H_{ice}	Ice thickness
S_{ice}	Ice intensity index
F_b	Maximum backward bending load
F_f	Maximum forward bending load
α_A	Apparent angle of attack for the blade section

Introduction

Ice-going vessels operate in the ice conditions that present various challenges to the propulsion system performance. Compared with conventional propulsion, the focus and difficulty of polar propulsion are that the conventional hydrodynamic performance, noise, cavitation, and excitation force of the propeller become more complicated due to the addition of ice. What is

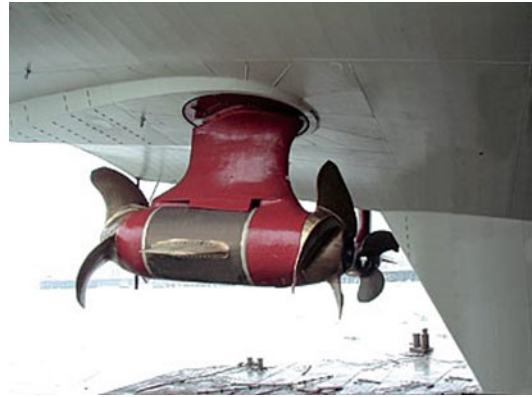
more complicated is the propeller will have a great chance to be in contact with the ice body directly during the ship navigation in the ice area, which makes the propeller suffer ice load directly. This will bring new challenges to the strength verification and the design work of the propeller. Therefore, polar propulsion is not simply an extension of a single discipline. It is a comprehensive science with direct value of engineering application. At present, the research focus of polar propulsion mainly contains the measurement and prediction work of propeller load, the propeller strength verification, and the design work.

Polar Propulsion Requirements and Selection

Icebreakers are typically meant to operate in more severe ice conditions than normal ice-going vessels, which means that strength and thrust requirements for the propulsion system need even more special consideration. Operative situations that require good maneuverability include assisting other ships, making ships free from ice, changing direction (astern), operating in narrow ports, and breaking out of channels. In addition to the raw power, some of these operations require precision and good controllability of both bow and stern of the vessel at various speeds. If the icebreaker is specified to have dynamic positioning capabilities, requirements for propulsion system are even more difficult to be satisfied with traditional shaftline and rudder arrangements.

Redundancy requirements of higher DP classes result in a need of multiple transversal or azimuthing thrusters at both ends of the ship. In such cases, azimuthing thruster propulsion is highly efficient as it can be used to reduce the total number of the thrusters. The axis of the Azipod propeller is vertical axis, and the propeller can rotate 360° around the axis, and it can obtain the maximum thrust in any direction. It can make the ship rotate in place, move laterally, retreat swiftly, and perform special operations such as steering in the low-speed range. Azipod propeller is divided into the Z-Azipod propeller and podded propulsion system. The structures of the two

propellers are different, but their hydrodynamic performance is similar. For the current widely favored bi-directional dynamic icebreaking, selecting the Azipod propulsion would bring too many benefits. The azimuth thruster eliminates the structure of rudder, tail, and stern tube. It also would simplify the ship stern shape and reduce hull resistance. Besides, it would not be neglected that the maneuverability of polar ship would be outstanding by using Azipod propeller. However, it would be not neglected that adjustable-pitch propellers would be another good choice for polar propulsion. Almost all of the propulsion of icebreakers built and rebuilt in the last two decades apply adjustable-pitch propellers or azimuth thrusters (including Rolls-Royce propellers, Azipod propellers, Schottel propellers, and Steerprop azimuth propellers). Adjustable-pitch propellers can use the operating mechanism in the hub to rotate the blades and change the pitch distribution of the blades, which will help to make the main engine power being fully absorbed under various operating conditions. It can not only improve the propeller performance but also reduce fuel consumption and improve economic efficiency. One of the advantages of adjustable-pitch propellers in the ice area is that they could produce more great thrust. This point is of great significance in the current situation of high energy costs. In addition, the effect of free ice in the propeller wake can be eliminated by adjusting the pitch. However, as the design of the propulsion system for the vessels in ice areas, the adjustable-pitch propeller and the Azipod propulsion device both have their own advantages and disadvantages. A new idea that the propulsion device is a Z-type transmission propulsion system has additional conduit with adjustable blades. The advantage is that the conduit has a protective effect on both the blade and the hub and improves the reliability of the propeller. This is particularly important for special propellers that are difficult to maintain after being damaged. The adjustable pitch and the overall rotation of the propeller around the shaft can greatly improve maneuverability of the vessels in ice areas and the efficiency of icebreaking. Adjustable-pitch blades can not



Polar Propulsion, Fig. 1 Azipod propeller and ducted propeller. (Referenced website: <http://www.engineeringsociety.co.uk/pod.html>)

only increase the propulsion efficiency in open water areas and the ability to break thick ice but also increase the thrust of the blades during icebreaking operations. The adjustable-pitch propellers and the Azipod propulsion system have been the research direction of icebreakers' propulsion devices in recent years. From the side, this data statistic also verifies the applicability and rationality of the application of adjustable-pitch propellers and Azipod propulsion system to the icebreaker. Figure 1 shows the Azipod propeller and ducted propeller.

P

Propeller Load

The polar propeller would be subjected to the following three types of loads: inseparable hydrodynamic load, separable hydrodynamic load, and ice load. Separable hydrodynamic load refers to the hydrodynamic load in open water conditions. Inseparable hydrodynamic load refers to the hydrodynamic load considering the influence of ice on the disturbance of the flow field, generally including the blocking effect, proximity effect, and cavitation effect. The ice load is caused by direct physical contact between ice and propeller blade and can generally be divided into two kinds: ice milling load and ice impact load. The division is based on the contact model and the contacting ice block size.

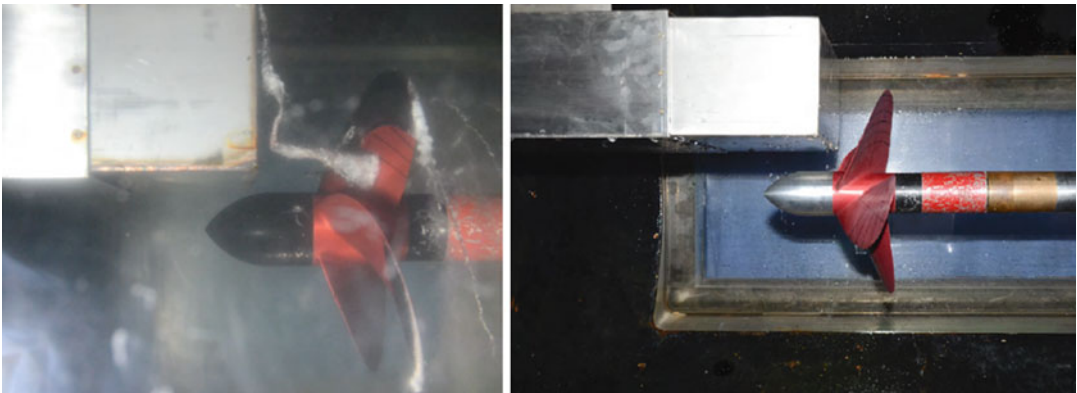
Hydrodynamic Load

At present, researchers study hydrodynamic loads mainly through the model tests and numerical predictions. People choose the model test as the main test style, and this is because compared with the full-scale test, model test has the advantages of less finical and labor cost as well as less demand for experimental conditions. And the shortcomings of model test are the uncertainty of similarity law between model test and real working condition. Besides, the selection of sea ice material in the model test is also the key to the success of the test. At present, there are frozen ice models as well as nonfrozen sea ice models made from other materials such as nonfrozen breakable synthetic materials.

Many scholars around the world have carried out research work on the prediction of propeller hydrodynamic load under ice loading. Among them, Yamaguchi (1993) proposed a lifting surface method to predict the propeller performance considering the interference of flow from ice. Bose (1996) used a three-dimensional unsteady surface element method to predict the performance of a propeller under the conditions of blocking flow. Doucet et al. (1998) predicted and analyzed the fluctuation of axial force of several large skew ice class propellers with the help of panel method PROPELLA. Liu Pengfei et al. (2002) simulated blocking flow conditions with three different ice shapes using the panel method program PROPELLER. Numerical results show that the blade load during the blockage is close to

the experimental results. On this basis, they have made a series of further studies on the hydrodynamic load prediction of propeller surface under the interaction between ice and flow in the following years. They improved the Kutta conditions of the unsteady program and carried out the design of heavy load ice class propeller based on improved panel method (2010). Previous researches on the inseparable hydrodynamic load were mostly based on potential flow theory. Therefore, the fluid viscous effects and boundary layer problems on ice and propeller could not be analyzed. In fact, these factors would show great influence on the performance of propeller. Wang (2013–2018) used the viscous flow theory and CFD overlap grid technology to numerically predict the propeller hydrodynamic performance and induced excitation force during the ice-propeller interaction process and developed the numerical prediction method of the induced excitation force and cavitation excitation force under ice blocking conditions and ice progressive conditions (Fig. 2).

Polar propellers are generally under heavy load conditions during icebreaking. It is difficult to avoid cavitation on the surface of the propeller, and the phenomenon of cavitation tends to aggravate the vibration and noise of the propeller, which in turn affects the hydrodynamic performance and strength of the propeller. The propeller-ice-flow interaction process is very complex. In order to simulate the ice-propeller milling and impact conditions, a large number of model tests have been carried out in the ice tank.



Polar Propulsion, Fig. 2 Propeller-ice milling process in cavitation tunnel (Wu et al. 2018)

However, due to the fact that atmospheric pressure cannot be scaled in the ice pool, the determination of the hydrodynamic load in the ice tank is often performed under the assumption that there is no cavitation on the propeller surface. In order to study the problem of cavitation in ice-propeller interactions, Lindroos and Björkestam (1986) simulated the propeller cavitation under the ice-propeller interaction in a cavitation tank for the first time. They placed a flat plate in front of the propeller to induce cavitation. The results showed that the blocking flow will make the propeller vibration and cavitation more serious, which in turn will increase the thrust and torque of the propeller. His study pointed out the importance of the study of the cavitation phenomenon. Sampson (2007) conducted propeller cavitation model tests in Emerson's cavitation tunnel during the period. The results of the study showed that the cavitation during propeller-ice interaction has a great influence on the strength and performance of the propeller. Cavitation will influence the thrust and torque of the propeller, causing severe vibration and noise, and cause fatigue damage to the propeller blade.

Ice Load

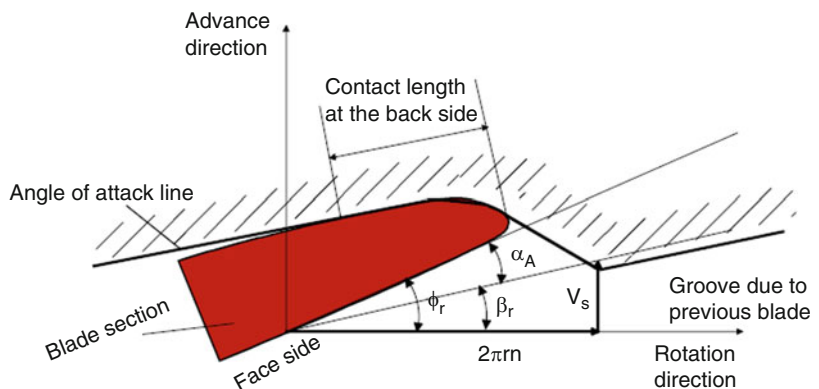
The ice load is caused by the direct contact between ice and propeller. It accounts for the most important part of the total load on the propeller. The study shows that the ice load is more than one order of magnitude larger than the hydrodynamic load. The contact loads are usually separated into milling-type or impact-type loads.

The difference between the two is not very clear. The load has often been classified as a milling load if the contact lasts more than one revolution, which means the same blade hits the ice block more than twice. In this work, the contact is always considered to be of the milling type when the leading edge of the blade comes into contact with ice, even if the ice block is small and the duration is not enough for a blade to pass. If the blade and ice block velocity geometry is such that the block hits the blade at the back or face side, without leading edge contact, the contact load is classified as an impact load. According to the propeller-ice contact model proposed by Kinnunen (2015), the angle of attack is accordingly one of the main parameters in contact load calculations (Fig. 3). The contact geometry between an ice block and the blade back side depends on the angle of attack as shown in Fig. 5. It is, however, not well-known in practice, since the actual wake factor at the disk level in ice conditions is not known. The propeller can accelerate the ice block before contact, or the ice block can be supported by other ice blocks. Therefore, an apparent angle of attack is used to indicate the contact geometry in this work, assuming the ice block enters into contact at the ship speed, V_s , as shown in Fig. 4. α_A is the apparent angle of attack for the blade section, V_s is the ship speed, n is the propeller speed, r is the radius of the section, β_r is the angle of advance of the section, ϕ_r is the pitch angle of the section.

At present, the research on the ice load is mainly obtained by numerical prediction and

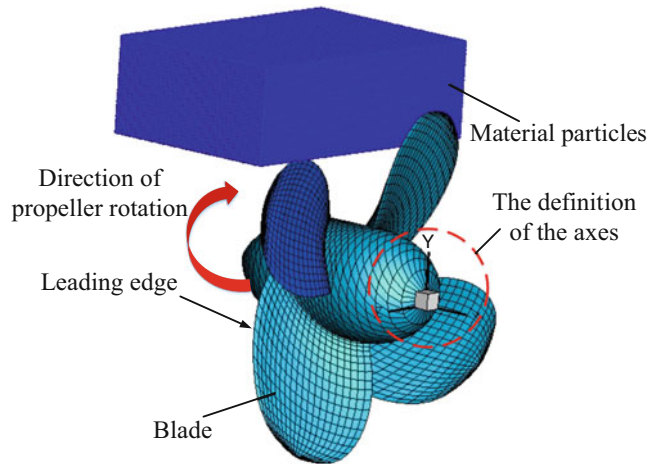
Polar Propulsion,

Fig. 3 Contact length between ice and back side of a blade section (Soininen 1998)



Polar Propulsion,

Fig. 4 Propeller-ice contact model established by PD theory



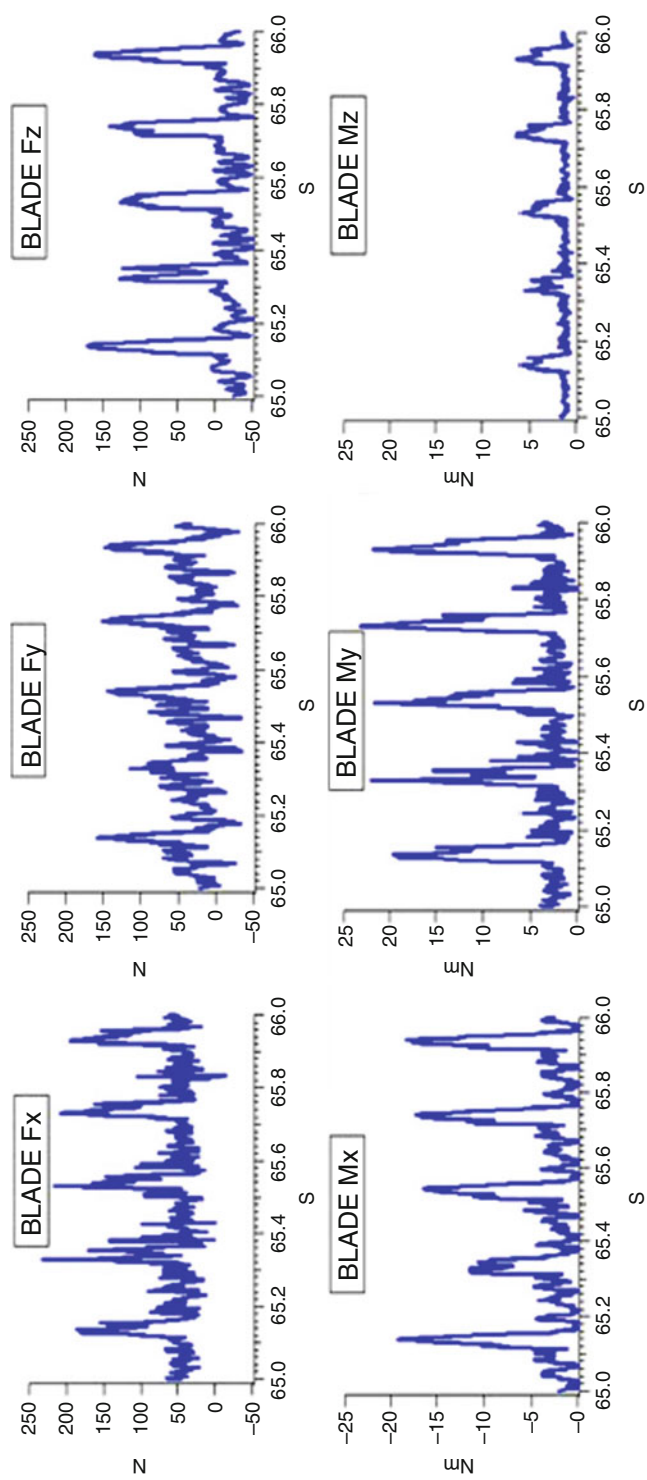
model test. Sampson carried out a series of ice cutting tests in air test platform and measured the ice load of propeller under ice cutting condition separately. With the development of computing ability, the use of commercial software or self-programming to predict blade ice load has become a mainstream. Compared with model tests, numerical prediction has the advantages of simple calculation, free working conditions, and low investment. However, how to simulate ice material and the fracture process of ice materials during milling process has become the main constraint to the application of numerical prediction. Among them, Hu and Xia (2013) built the propeller-ice contact model based on the ANSYS/LS-DYNA platform. The sea ice was simulated by SPH method. And the finite element method is used to solve the structural deformation of the blade under ice load. In the same year, the structural dynamic response of propeller blade under different milling areas, advance speeds, and collision angles were studied and analyzed through this numerical model. To handle a large number of discontinuous problems happening in the ice-propeller milling process, Ye et al. (2017) introduced meshless method peridynamics and established ice-propeller numerical model. He carried out many prediction calculations to ice impact conditions and ice milling conditions and obtained the variation regularity of the blade ice load with external conditions, as shown in the following picture.

Mixing Load

The ice-water mixing load of propeller in ice areas has always been the focus and difficulty in the research. At present, the most effective research methods are model test measurements. Among them, Wang et al. (2005) carried out model test of Azipod propeller milling ice in IOT's tank and obtained valuable data of the mixing loads as Fig. 6 shows. These graphs except for the carriage speed are plotted over a 1-s interval, the 65th to 66th seconds. In the time series records for the forces and moment, five peaks can be seen because the propeller rotational speed was 5 rps. The carriage speed for the duration of the 65th and 66th seconds was 0.5 m/s. Wang et al. (2007) installed a 6-degree-of-freedom mechanical sensing device at the root of the main blade and successfully measured the ice-fluid mixed load of the blade. There are no well-developed numerical prediction due to the complexity mechanism of fluid-solid coupling. Wang et al. (2017) try to establish the numerical model with PD-FEM-SPH coupling method, but the meshless method makes the calculations work too large being performed. So establishing a numerical method to this kind of problem would be very innovative and of engineering practicality.

Polar Propulsion Strength

The strength of polar propulsion is very essential. In order to check the propeller strength more



Polar Propulsion, Fig. 5 Six degrees of loads measured in model test conducted in IOT ice pool (Wang et al. 2005)



Polar Propulsion, Fig. 6 Propeller blade structural damaged by ice loads. (Referenced source: DNV Classification Notes No. 51.1)

effectively, many classification societies have formulated special standards for the polar propeller strength and add them into the ice class rules separately. The International Association of Classification Societies (IACS) had collated and arranged the existing regulations to ensure the safe navigation of ships in the ice areas and formulated the IACS ice grade specification in 2008. At present, most of the ice class specifications around the world are very similar to the specifications put forward by IACS. The specification does not only contain the demand for the strength checking under ice load but also provide the requirements for the material properties of propeller. The load given in the IACS ice class rules is only applicable to the propeller, also defined as the maximum load that the propeller is expected to sustain during the life cycle.

The load given should be recommended as the total load including hydrodynamic load and ice load. For conventional propeller blades, the design load calculation method introduced by the IACS specification is given by Eq. 1 through Eq. 6.

F_b is defined as the maximum backward bending load on the blade during the ship life cycle, D is the propeller diameter, n is the propeller speed, EAR is the disc area ratio, Z is the number of blades, H_{ice} is the ice thickness, D_{limit} is the ice thickness index, and S_{ice} is the ice intensity index:

$$D < D_{limit}$$

$$F_b = -27S_{ice}(nD)^{0.7} \left(\frac{EAR}{Z} \right)^{0.3} D^2 \text{ kN} \quad (1)$$

$$D \geq D_{limit},$$

$$F_b = -23S_{ice}(nD)^{0.7} \left(\frac{EAR}{Z} \right)^{0.3} H_{ice}^{1.4} \cdot D \text{ kN} \quad (2)$$

$$D_{limit} = 0.85 \cdot H_{ice}^{1.4} \quad (3)$$

F_f is defined as the maximum forward bending load on the blade during the ship life cycle:

$$D < D_{limit},$$

$$F_f = 250 \left[\frac{EAR}{Z} \right] [D]^2 \text{ kN} \quad (4)$$

$$D \geq D_{limit},$$

$$F_f = 500 \left(\frac{1}{1 - \frac{d}{D}} \right) H_{ice} \left[\frac{EAR}{Z} \right] [D] \text{ kN} \quad (5)$$

$$D_{limit} = \left(\frac{2D}{D - d} \right) \cdot H_{ice} \quad (6)$$

Polar Propulsion Design

Due to the special working environment of the polar ship, it is necessary to consider the influence of the environment, low temperature, and sea ice on the propulsion system. Therefore, the design of the polar propulsion system is different from the conventional propulsion system. The propulsion performance in ice conditions and open water conditions should be considered for the design work. Therefore, it is necessary to perform proper performance analysis under different navigation conditions during the propeller design process. In general, it is essential to determine whether the design of the blades meets the strength requirements according to the ice class rules. Due to the special requirements on strength, the blade section thickness of the ice class

propeller will be slightly thicker than that of the conventional propellers. In addition, the trailing edge of the propeller will become blunt in the rules for the radius of the guide circle. For such kind of propeller with a blunt trailing edge, the special blade section shape can cause a large pressure drag, resulting in a decrease in the propulsion efficiency. Usually, the propulsion efficiency of polar ship would be reduced by about 6% compared to a conventional propeller. The too low ambient temperature of the polar propulsion system determines the special design requirements of the main engine, ventilation system, and cooling water system. Thermal insulation measures for these systems need to be taken into account in the design to prevent the low-temperature environment from affecting its normal operation. In order to ensure that the polar ship can navigate in some areas where the noise level is limited, the designed propeller must also be optimized for radiated noise. Of course the propeller-ice contacting behavior could not be ignored; the shape of the blade must be specially designed so that the blade can effectively mill sea ice. The design of the ice class propeller must also be coordinated with the rational design of the hull lines to reduce the amount of sea ice flowing to the propeller. This would be helpful to avoid the blocking effect, noise, and cavitation effect.

Cross-References

- [Ice Breaking Vessel](#)
- [Ice Tank Test](#)

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Polar Research and Supply Vessel

- [Polar Research Vessel](#)

Polar Research Vessel

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Synonyms

[Polar research and supply vessel](#)

Definition

Polar research vessel is a special kind of oceanographic research ships, which can also be seen as a special icebreaker.

Like most research vessels, polar research vessel has good control performance, improved the navigation and positioning system, and also is equipped with special equipment and instruments necessary for the execution of task. However, based on the geographical environment of the polar regions, polar research vessel should be able to work in harsh ice area for a long time.

Therefore, polar research vessels are constructed around an icebreaker hull, allowing them to engage in ice navigation and operate in polar waters. Due to most of ships cannot enter into the polar region, polar research vessels often play different roles.

Polar Regulations and Polar Research Vessel

At present, involving the polar shipping unified international rules mainly has two aspects. On the one hand, IACS has issued unified requirements for polar Marine, “Requirements Concerning Polar Class,” which is formulated by the nation classification societies jointly (IACS 2011). On the other hand, International maritime organization (IMO) issued a guide “International Code for Ships Operating in Polar Waters” in December 2002 (IMO 2013). On the basis of the guide, ships sailing in Antarctic waters are also

supplemented. The “Polar Code” was adopted in 2014 as a mandatory rule for polar sailing ships, which took effect on January 1, 2017.

The polar regulations regulate not only the ice grade, structure, stability, and seaworthiness of polar ships, but also the requirements for environmental protection. The regulations place strict limits on the polar regions, especially the Antarctic, and express trust in the polar research vessel. In other words, polar research ships are polar pioneers. She should fulfill her responsibilities and lead the trend of polar environmental protection and launch the polar green voyage.

Typical Polar Research Vessel

There are many famous polar research vessels in the world. Since the 1970s, the United States and the Soviet Union have begun to build. Nowadays, Russia have the largest number of polar ships. Japanese polar research vessel owns the comprehensive functions. The American polar star is one of the most famous polar scientific research vessels in history.

Polar Star

USCGC Polar Star (WAGB-10) is a United States Coast Guard heavy icebreaker. In 1976, the ship was built by Lockheed Shipbuilding and Construction Company of Seattle, Washington, along with her sister ship, USCGC Polar Sea ([USCGC Polar Star – History](#)). Now the polar star is the only heavy icebreaker in the United States, after the USCGC Healy was classified as a medium icebreaker.

Unique Ship Design

Polar Star (Baker and Jones 1998), which is shown in Fig. 1, uses four different methods of electronic navigation to overcome the difficulties of high-latitude operations, and a computerized propulsion control system to effectively manage six diesel-powered propulsion generators, three diesel-powered ship’s service generators, three propulsion gas turbines, and other equipment vital to the smooth operation of the ship. The extensive use of automation and low maintenance

Polar Research Vessel,
Fig. 1 The US coast guard
Polar Star



materials has greatly reduced staffing requirements.

Polar Star’s three shafts are turned by either a diesel-electric or gas turbine prime mover. This ensures her strong power. The combination of the hull and the reasonable internal structure endows the polar star very good structural strength. It is worth mentioning that the steel used in these structures has good low-temperature strength. In addition, Polar Star’s hull form is designed to maximize icebreaking by efficiently combining the forces of the ship’s forward motion, the downward pull of gravity on the bow, and the upward push of the inherent buoyancy of the stern. The curved bow allows Polar Star to ride up on the ice, using the ship’s weight to break the ice. Under the guarantee of these engineering techniques, polar star is able to break through ice up to 6.4 m thick by backing and ramming and can steam continuously through 1.8 m of ice at 3 knots. Polar voyages are difficult, so the Polar Star provides the crew with comfort living environment as much as possible.

General Characteristics

Class and type: Polar-class icebreaker
Displacement: 10,863 long tons (11,037 t) (standard)

(continued)

13,623 long tons (13,842 t) (full)		
Length: 399 ft (122 m)	Beam: 83 ft 6 in (25.45 m)	Draft: 31 ft (9.4 m)
Installed power: Six Alco 16 V-251F diesel engines (6 × 3,000 hp (2,200 kW))		
Three Pratt & Whitney FT-4A12 gas turbines (3 × 25,000 hp (19,000 kW))		
Propulsion: Combined diesel-electric or gas (CODLOG)		
Three shafts; controllable-pitch propellers		
Speed: 18 knots (33 km/h; 21 mph)		
3 knots (5.6 km/h; 3.5 mph) in 6-foot (1.8 m) ice		
Range: 16,000 nautical miles (30,000 km; 18,000 mi) at 18 knots (33 km/h; 21 mph)		
28,275 nautical miles (52,365 km; 32,538 mi) at 13 knots (24 km/h; 15 mph)		
Complement: 15 officers 127 enlisted 33 scientists		
12-person helicopter detachment		
Armament: 2 × 0.50 caliber machine guns Various small arms		
Aircraft carried: 2 HH-65A Dolphin helicopters		

Multiple Mission

Polar Star has a variety of missions while operating in polar regions. First of all, Polar Star serves as a scientific research platform with five laboratories and accommodations for up to 20 scientists. Researchers can do some work about fields of geology, volcanology, oceanography, sea-ice physics, and other disciplines which only be operated on a polar research vessel. Besides, the polar

research vessel is tasked with transporting supplies to polar research stations. For example, during Antarctic deployments, the primary missions include breaking a channel through the sea ice to resupply the McMurdo Research Station in the Ross Sea. Resupply ships use the channel to bring food, fuel, and other goods to make it through another winter. Last but not least, the polar research vessel sails and studies alone in the polar regions. Therefore, it is necessary for researchers on the polar research vessel to own strong self-survival and crisis management ability. It is also an important task to protect themselves and rescue other research vessels in distress.

Monumental Achievement

Polar Star was back in operation in late 2013, and assigned to Antarctic operations as part of Operation Deep Freeze in early 2014 (Boyle 2013). She was dispatched from Sydney on January 4, 2014, to attempt a rescue of the Russian research vessel Akademik Shokalskiy and Chinese icebreaking research vessel Xue Long trapped at that time in Antarctic ice, the former since 24 December 2013 (U.S. Breaker to Help Russian 2014; U.S. Coast

Guard Cutter Polar Star 2014). However, on January 8, 2014, the Australian Maritime Safety Authority confirmed that Polar Star had been released to scheduled duties as both vessels had broken free and were proceeding to open water (Antarctic Rescue Operations Complete 2014). In February 2015 Polar Star was involved in the rescue of the Australian fishing vessel Antarctic Chieftain, towing her and her 27 crew to safety, through ocean ice and snow nearly 20 feet deep in the Southern Ocean (<http://www.ktvz.com/news/us-coast-guard-icebreaker-frees-stuck-ship/31284578>). In February 2017, fire crews from the USCGC Polar Star were to made available to help the New Zealand Fire Service and NZDF Fire Crews in fighting against the Christchurch Port Hills Fires in Christchurch, New Zealand.

Shirase

New Shirase, as shown in Fig. 2 (Hull number: AGB-5003), was a new Japanese icebreaker operated by the Japan Maritime Self-Defense Force (JMSDF) and Japan's fourth icebreaker for Antarctic expeditions. She is the successor of the old Shirase (AGB-5002). They have the same name.



Polar Research Vessel, Fig. 2 The Japanese icebreaker Shirase

New Shirase was launched in April 2008 and was commissioned in May 2009. Since November 2009, She served in 51 Antarctic expeditions. She was able to move at three knots breaking 1.5 m of ice.

General Characteristics

Type: Icebreaker	
Displacement: 12650 t (standard) 20373 t (full)	
Length: 138 m (452 ft 9 in)	
Beam: 28 m (91 ft 10 in)	Draft: 9.2 m (30 ft 2 in)
Propulsion: Diesel-electric Four propulsion motors, 22,050 kW (30,000 hp) (combined)	
Two shafts; fixed-pitch propellers	
Speed: 19.5 knots (22.4 mph; 36.1 km/h) (max)	
Complement: Approx. 175 sailors + 80 researchers	
Aircraft carried: 3 helicopters	

The Development of Polar Research Vessel in Recent Years

Polar research vessels in new age mean more than ice-breaking capabilities, longer endurance, and better research facilities. At the same time, the new polar research vessel represents a greener

and more efficient energy source and more flexible ways to break ice.

RRS Sir David Attenborough (Planet Ice and the Dual-Functional 2017)

Design Characteristics

Tonnage: 15,000 GT (4,475 DWT)	
Length: 128.9 m (423 ft)	Beam: 24 m (79 ft)
Draught: 7 m (23 ft)	Depth: 11 m (36 ft)
Ice class: PC 4 (propulsion system PC 5)	
Installed power: 2 × Bergen B33:45L6A (2 × 3,600 kW) 2 × Bergen B33:45L9A (2 × 5,400 kW)	
Propulsion: Diesel-electric; two shafts 2 × 2,750 kW (Two 5-bladed controllable pitch propellers) + four Tees White Gill thrusters (2 bow + 2 stern ,4 × 1,580 kW)	
Two 5-bladed controllable pitch propellers	
Speed: 17.5 knots (32.4 km/h; 20.1 mph) (maximum) 13 knots (24 km/h; 15 mph) (cruising) 3 knots (5.6 km/h; 3.5 mph) in 1 m (3 ft) ice	
Range: 19,000 nautical miles (35,000 km; 22,000 mi) at 13 knots	
Endurance: 60 days Crew: 30 crew 60 scientists	
Aircraft carried: 1 helicopter	

The British Antarctic survey asks currently the UK Cammell Laird shipyard to build a new polar research vessel. “RRS Sir David Attenborough”



Polar Research Vessel, Fig. 3 RRS Sir David Attenborough

as shown in Fig. 3, which is used to replace “RRS Ernest Shackleton” and “RRS James Clark Ross” working in the south and the north Pole, expected to be delivered in 2019. The new polar research vessel costs about 200 million pounds. The ship is 128 m long and 24 m wide, owning a self-sustaining capacity of 60 days (polar region), and can carry more than 30 crew members and 60 scientists. Sir David Attenborough will have a twin-shaft hybrid diesel-electric propulsion system. The power plant, which can run with different configurations depending on the mission and operating conditions, produces electricity to two 2,750 kW (3,690 hp) asynchronous electric motors driving two 5-bladed controllable pitch propellers. At an economical cruising speed of 13 knots (24 km/h; 15 mph), she will have an operating range of 19,000 nautical 3D model diagram of RRS Sir David Attenborough miles (35,000 km; 22,000 mi). For maneuvering and dynamic positioning, the vessel will have four 1,580 kW (2,120 hp) Tees White Gill thrusters, two in the bow and two in the stern (Polar Research Vessel).

At 3kn speed, it can break 1 m ice, and the ice-class is PC4 (the propulsion system is PC5). The

ship is equipped with a large number of professional scientific research equipment and instruments and has advanced laboratories for Marine, submarine, and atmospheric research.

Researchers can collect the marine environment and biological data through the underwater robot and underwater glider, and at the same time, aerial robots and shipborne environmental monitoring systems can provide detailed polar environmental information. In order to meeting the increasing demands of scientific research, the laboratory space is designed to be reconfigurable and to be extended for building laboratory modules.

Xue Long 2

Xue Long 2, which is shown in Fig. 4, is a Chinese new generation of icebreaking research vessel that had entered service in 2019. She is named after Xue Long (China builds first polar research vessel 2018).

Design

Xue Long 2 measures 122.5 m (402 ft) long, with a beam of 22.3 m (73 ft) and a draft of 8.3 m (27 ft) at full load. She has a designed displacement of 13,996 tones (Xinhua 2017). She has a diesel-



Polar Research Vessel, Fig. 4 First icegoing picture of Xue Long 2

electric propulsion system, with two 16-cylinder and two 12-cylinder engines, both Wärtsilä 32-series designs, powering two 7.5 MW Azipods that give her a speed of up to 15 knots (17 mph) in open water and 2–3 knots (2.3–3.5 mph) when breaking ice. Xue long2 combines the technology, functional requirements, and environmental protection concept of the international new-generation research vessel. She adopts the internationally advanced hull form design with double directions icebreaking and owns a 360° Azimuth pod electric propulsion system, DP-2 dynamic positioning system, and capability of impacting ice to fragmentation. She can also realize the 360° free rotation in the polar region and break through the 20-m ice ridge in the polar region. In addition, the ship's maneuverability will be greatly improved, which can meet the navigation needs of the global unlimited navigation area. The structural strength meets the requirements of PC3, and the ice can be continuously broken at a speed of 2–3 knots on a 1.5 m thick ice layer covered with 0.2 m of snow. At the same time, the entire ship generator set is equipped with an exhaust gas cleaning system, focusing on the protection of the polar environment.

Furthermore, Xue Long 2 will be equipped with a smart ship system. It is the first time that China has equipped an intelligent Hull (China Classification Society (CCS) notation: i-ship H) system on a scientific research vessel. And it is also the first time that China has installed a smart Machinery (CCS notation: i-ship M) in an electric propulsion ship. This will effectively reduce the labor intensity of the crew and improve the operational efficiency of the ship.

Summarize the evolution of polar research vessel technology over the decades. The main development directions are as follows: hull design, auxiliary ice-breaking capacity, power plant, and hull form design with double directions icebreaking.

Polar Research Vessel Outlook and Conjecture

In this section, research and development with respect to the hull design, assist in ice breaking ability, power plant and doubling ice-breaking design are presented.

Hull Design

During the development of the polar research vessel for several decades, the complicated ice conditions brought her a lot of trouble. In order to adapting to the harsh ice conditions, polar research vessel need to have a larger displacement, a more reliable hull structure, and a longer range.

Assist in Ice Breaking Ability

In the last years, a variety of auxiliary ice-breaking systems have been developed to improve the ice-breaking capacity of polar research vessel. But these systems have only been applied to a few advanced ships and have not yet been popularized. Improvement and popularization of new technologies will make it possible to improve the working environment of polar research vessels.

Power Plant

Based on polar regulations and people's growing environmental awareness, we should search for more green energy. Similarly, we should also look for more powerful energy to support us to explore the polar regions. Both nuclear and electric power may be a direction for us, especially electrical propulsion.

Doubling Ice-Breaking Design

Doubling icebreaking is a very practical ice-breaker design for the ship with 360° azimuth propulsion units. The icebreaker with this design can not only use the bow to break ice, but also use the stern to break ice. The new technology of double icebreaking greatly enhances the maneuverability of the polar ship. When a polar ship possessing this design sails in the ice zone, she may take less time to turning around in ice area and she is less likely to get stuck in the ice layers.

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Polar Trading Vessel

- [Polar Merchant Vessel](#)

Polarization Imaging

- [Photoelectric Detection Technology in Underwater Vehicles](#)

Polyamide (PA) Net

- [Net Structures: Design](#)

Polyester Rope

- [Cable](#)

Polyethylene (PE) Net

- [Net Structures: Design](#)

Pontoon Bridge

- [Floating Bridge](#)

Position Mooring (PM)

- [Thruster-Assisted Mooring](#)

Position-Holding

- [Station-Keeping System for VLFS](#)

Positioning

- [Station-Keeping System for VLFS](#)

Positioning System

- [External Turret Single Point Mooring System](#)
- [Internal Turret Single-Point Mooring \(SPM\) System](#)
- [Soft YOKE Single Point Mooring System](#)

Position-Keeping

- [Station-Keeping System for VLFS](#)

POSMOOR, Position-Mooring

► Dynamic Positioning in Ice

Power Cable

► Cable

Power Take-Off System

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Synonyms

LIMPET – Land Installed Marine Power Energy Transmitter; OWC – Oscillating water column; PTO – Power take-off; WEC – Wave energy converter

Definition

A power take-off of a wave energy converter is a mechanism with which the absorbed energy in the form of mechanical/pneumatic/potential energy from the primary energy conversion stage is transformed into useful mechanical energy for further energy conversion, mostly into electricity if a generator is connected to the PTO.

Scientific Fundamentals

Power Conversion Principle

For power conversion, it can be made using either translational or rotational motions and their corresponding PTO force or torque. When a translational motion is used for power conversion, the capture power by the PTO is calculated as

$$P = F_{pto} \cdot V \quad (1)$$

where F_{pto} is the force acting on PTO and V the velocity of the PTO for power generation.

When a rotary motion is used for power conversion, the power is given by

$$P = T_{pto} \cdot \omega \quad (2)$$

where T_{pto} is the torque acting on PTO and ω the angular velocity of the PTO for power generation.

From the formulae for power conversion, it can be easily seen that to generate a large power, two significantly different ways can be adopted:

- 1) Either with a very high (translational/rotational) speed but a small force/torque. Therefore, such a system tends to have a very high system reliability, and a high energy conversion efficiency (generally higher than 90% in conventional energy conversion) can be obtained. These can be seen in conventional power generation using steam turbines, which could directly generate electricity to grid (with frequency of 50 Hz or 60 Hz).
- 2) Or with a very large force when the velocity in the power convention system is low. These happened in most renewable energy converters in wind, tidal, and wave energy systems. Such systems tend to have a low reliability and low energy conversion efficiency. For wave energy conversion, a specific disadvantage is the reciprocating wave force and velocity (with a dominant period about 6–12 s for wave energy conversion (Falcao 2008; Sheng and Lewis 2016a)). The nature of wave energy conversion with a large force and a low speed (both are reciprocating in most cases) presents a serious engineering challenge in practical application in the following aspects:
 - System and component reliability: fatigue problems with stress concentration
 - Low energy conversion efficiency: directly linking to the initial capital of cost of energy production

Power Conversion

To convert the trapped energy by the WEC, different PTOs may be used, including linear PTOs

(direct drive or Wells turbine), nonlinear PTOs (impulse turbine), or coulomb PTO (hydraulic cylinder). For a simple linear PTO, the PTO force is proportional to the PTO displacement, velocity, and acceleration, given in an expression following (Babarit et al. 2012):

$$F_{pto} = M_{pto1}\ddot{x} + B_{pto1}\dot{x} + C_{pto1}x \quad (3)$$

where M_{pto1} , B_{pto1} , and C_{pto1} are constants, independent of the PTO displacement, velocity, and acceleration.

It is noted that only the damping term could extract a net power conversion, while other two terms could absorb energy in a part of the wave cycle, but they also feed energy back to wave in the other part of the wave cycle. As such, the overall net power is zero if the energy conversion efficiency is 100% or negative if the efficiency is less than 100%.

Wave Energy Conversion Maximization

PTO optimization for maximizing wave energy conversion for a given wave energy converter is a very important aspect when designing a wave energy converter and setting the relevant PTO parameters.

When examining the maximized wave energy absorption for a wave energy converter, it is generally accepted that the analysis on a linear system will give the correct information on how much wave energy can be converted by the device, since the maximized wave energy conversion for a given wave energy converter is same for both linear and nonlinear PTOs if the PTOs are optimized (see Sheng and Lewis 2016b). For this reason, the maximization of wave energy conversion by a wave energy converter can be simply carried out in frequency domain

$$\left[i\omega(\mathbf{M} + \mathbf{A} + \mathbf{M}_{pto}) + (\mathbf{B} + \mathbf{B}_{pto}) + \frac{1}{i\omega}(\mathbf{C} + \mathbf{C}_{pto}) \right] \mathbf{U} = \mathbf{F}_{ex} \quad (4)$$

where the bold letters in square bracket mean matrices and the letters with subscript *pto* correspond to the matrices for the power take-off system and $\mathbf{U}$ and $\mathbf{F}_{ex}$ are the complex vectors.

Following (Zheng et al. 2016a; Evans 1981; Falnes 1999), the average absorbed power from wave can be given as

$$\bar{P} = \frac{1}{4} (\mathbf{F}_{ex}^* \mathbf{U} + \mathbf{U}^* \mathbf{F}_{ex}) - \frac{1}{2} \mathbf{U}^* \mathbf{B} \mathbf{U} \quad (5)$$

where the asterisk $*$ means the conjugate of the complex vector. This formula indicates that the converted average power $\bar{P}$ equals the gross power due to the equivalent excitation force, $\mathbf{F}_{ex}$, less the power due to the wave radiations.

The equation can be further written as

$$\begin{aligned} \bar{P} = & \frac{1}{8} \mathbf{F}_{ex}^* \mathbf{B}^{-1} \mathbf{F}_{ex} \\ & - \frac{1}{2} \left(\mathbf{U} - \frac{1}{2} \mathbf{B}^{-1} \mathbf{F}_{ex} \right)^* \mathbf{B} \left(\mathbf{U} - \frac{1}{2} \mathbf{B}^{-1} \mathbf{F}_{ex} \right) \end{aligned} \quad (6)$$

This could lead to an overall maximized wave energy conversion:

$$\bar{P}_{MAX} = \frac{1}{8} \mathbf{F}_{ex}^* \mathbf{B}^{-1} \mathbf{F}_{ex} \quad (7)$$

under the following constraint for an optimal velocity:

$$\mathbf{U}_{opt} = \frac{1}{2} \mathbf{B}^{-1} \mathbf{F}_{ex} \quad (8)$$

The condition implies that the overall maximal wave energy conversion happens at the resonance of the dynamic system with the PTO damping equalling to the radiation damping. The explanation is given as follows.

The solution of Eq. (4) can be simply expressed as

$$\mathbf{U} = \left[i\omega(\mathbf{M} + \mathbf{A} + \mathbf{M}_{pto}) + (\mathbf{B} + \mathbf{B}_{pto}) + \frac{1}{i\omega}(\mathbf{C} + \mathbf{C}_{pto}) \right]^{-1} \mathbf{F}_{ex} \quad (9)$$

When in resonance, the following condition must be satisfied:

$$\omega^2(\mathbf{M} + \mathbf{A} + \mathbf{M}_{pto}) - (\mathbf{C} + \mathbf{C}_{pto}) = 0 \quad (10)$$

When the system is in resonance, the velocity is given as

$$\mathbf{U} = (\mathbf{B} + \mathbf{B}_{pto})^{-1} \mathbf{F}_{ex} \quad (11)$$

Although the matrices $\mathbf{B}$ and $\mathbf{B}_{pto}$ are both real matrices, Eq. (11) cannot be simply interpreted as the optimal velocity $\mathbf{U}$ is in phase with excitation $\mathbf{F}_{ex}$. However, if the single motion mode is for wave energy conversion, the velocity is indeed in phase with the excitation. Hence the resonance condition is called as the phase optimal condition.

Taking $\mathbf{B}_{pto} = \mathbf{B}$ as the amplitude control, Eq. (11) is same as Eq. (8), a condition corresponding to the maximal wave energy absorption, given by Eq. (7). This optimal PTO damping level is the amplitude optimal condition.

The maximization of energy absorption by a wave energy converter above actually provides the basic principle for control technologies on how to achieve the maximized wave energy absorption: in regular wave, it can be done simply by adjusting PTO's $\mathbf{M}_{pto}$ and/or $\mathbf{C}_{pto}$ to make the dynamic system in resonance with the wave excitation to achieve the optimal phase control, while by setting the PTO's damping level, it is possible to attain the optimal amplitude control (Falnes 1995).

When the phase and amplitude are both optimally controlled, the control is optimal. When only the phase or the amplitude is optimized or partially optimized, the control is suboptimal. In reality, suboptimal controls may be more realizable due to less strict constraints.

PTO Optimization

Fix-referenced WECs (One Motion Mode)

The simplest case would be the fix-referenced wave energy converters where a single motion mode is accounted for wave energy conversion, such as the fix-referenced point absorber using

heave motion (Seabased 2015; CETO Rafiee and Fievez 2015, etc.), the bottom-fixed oscillating surge wave energy converter using the pitch of the flap (Oyster Whittaker and Folley 2012), or the bottom-fixed OWC device, using the heave motion of the fluid in the water column (LIMPET Folley et al. 2006). For such simple devices, the analytical PTO optimization can be easily derived (see Falnes 2002; Sheng and Lewis 2012).

Taking a fix-referenced point absorber as an example, its heave motion is used for wave energy conversion, and the solution of the dynamic system is simply given as

$$U_3 = \frac{F_{ex3}}{i\omega(M + A_{33} + M_{pto}) + (B_{33} + B_{pto}) + \frac{1}{i\omega}(C_{33} + C_{pto})} \quad (12)$$

where subscript "3" means the heave motion following the convention in the boundary element method.

The average wave energy absorption by the PTO is

$$\bar{P} = \frac{1}{2} B_{pto} \frac{|F_{ex3}|^2}{(B_{33} + B_{pto})^2 + [\omega^2(M + A_{33} + M_{pto}) - (C_{33} + C_{pto})]^2} \quad (13)$$

The maximal power absorption happens when

$$B_{pto} = \sqrt{B_{33}^2 + \omega^2 \left[(M + A_{33} + M_{pto}) - \frac{1}{\omega^2} (C_{33} + C_{pto}) \right]^2} \quad (14)$$

The maximal average power is calculated as

$$\bar{P}_{\max} = \frac{1}{4} \frac{|F_{ex3}|^2}{B_{33} + \sqrt{B_{33}^2 + [\omega^2(M + A_{33} + M_{pto}) - (C_{33} + C_{pto})]^2}} \quad (15)$$

When the point absorber is in resonance with incident wave, a condition must be satisfied:

$$\omega^2(M + A_{33} + M_{pto}) - (C_{33} + C_{pto}) = 0 \quad (16)$$

we have from Eqs. (14) and (15) as

$$B_{pto} = B_{33} \quad (17)$$

$$\bar{P}_{MAX} = \frac{1}{8} \frac{|F_{ex3}|^2}{B_{33}} \quad (18)$$

Eq. (18) is a special case of Eq. (7) when a single motion mode is used for wave energy conversion.

Self-Referenced WECs (Multiple Motion Modes)

For massive wave energy production, the wave energy converters may be more likely deployed in the relative large water depth (larger than 50 m) where the wave energy resources are better, and it is more economic for floating wave energy devices. As such, they may use multiple motion modes or relative motions for wave energy conversion.

When the relative motion from two motion modes is used for wave energy conversion, the analytical PTO optimization can be still obtained (Falnes 1999) and Sheng et al. (Sheng and Lewis 2016b). Taking the relative heave motion of two bodies as the motion for wave energy conversion as seen in OPT and RM3 floating point absorbers, the dynamic equation is

$$\begin{cases} \left[i\omega(m_{33} + a_{33}) + b_{33} + \frac{c_{33}}{i\omega} \right] v_3 + \left[i\omega a_{39} + b_{39} + \frac{c_{39}}{i\omega} \right] v_9 = f_3 - B_{pto}(v_3 - v_9) \\ \left[i\omega a_{93} + b_{93} + \frac{c_{93}}{i\omega} \right] v_3 + \left[i\omega(m_{99} + a_{99}) + b_{99} + \frac{c_{99}}{i\omega} \right] v_9 = f_9 + B_{pto}(v_3 - v_9) \end{cases} \quad (19)$$

where m_{33} and m_{99} are the mass of the bodies; a_{33} , a_{39} , a_{93} , and a_{99} the added mass coefficients; b_{33} , b_{39} , b_{93} , and b_{99} the potential damping coefficients; c_{33} , c_{39} , c_{93} , and c_{99} the restoring force

coefficients; f_3 and f_9 the complex heave excitation force amplitudes of two bodies; and v_3 and v_9 the complex heave motion velocity amplitudes of the two bodies.

$$\begin{pmatrix} i\omega(m_{33} + a_{33}) + (b_{33} + B_{pto}) + \frac{c_{33}}{i\omega} & i\omega a_{39} + (b_{39} - B_{pto}) + \frac{c_{39}}{i\omega} \\ i\omega a_{93} + (b_{93} - B_{pto}) + \frac{c_{93}}{i\omega} & i\omega(m_{99} + a_{99}) + (b_{99} + B_{pto}) + \frac{c_{99}}{i\omega} \end{pmatrix} \begin{pmatrix} v_3 \\ v_9 \end{pmatrix} = \begin{pmatrix} f_3 \\ f_9 \end{pmatrix} \quad (20)$$

Once the dynamic equation is solved, the average power absorption is given as

$$\bar{P} = \frac{1}{2} B_{pto}(v_3 - v_9)(v_3 - v_9)^* \quad (21)$$

Both Falnes (1999) and Sheng et al. (Sheng and Lewis 2016b) have given the expressions for the optimized B_{pto} and the corresponding average power $\bar{P}$. Obviously, the detailed expressions are

much more complicated than those in a single motion mode for wave energy conversion (the complicated mathematical expression and derivation can be found in (Sheng and Lewis 2016b; Falnes 1999)).

For more general cases with multiple motion modes, the analytical PTO optimizations are more complicated. In fact, they can be made only in some special cases (more details can be found in Pizer (1993) and Zheng et al. (2016b)).

PTO Optimization for Irregular Waves

The PTO optimizations mentioned above are valid only for regular waves. In irregular waves, they can be regarded as a combination of many different regular components of different amplitudes, frequencies, and phases, and for each wave cycle, its period and height are different. To achieve an optimized PTO damping for a given sea state, the trial-and-error method is usually used. For instance, Bull (2014) suggested 200 different damping levels are used to try out the optimized PTO damping for the RM3 BBDB OWC wave energy converter, while Sheng et al. (Sheng and Lewis 2016b) suggested that using T_e as a reference wave period for the irregular waves to calculate optimized PTO damping (same method as for regular waves) and the search of the optimized PTO damping levels could be close to the one based on the reference period of T_e . As such, the optimized PTO damping can be more easily found.

Key Applications

Types of PTO

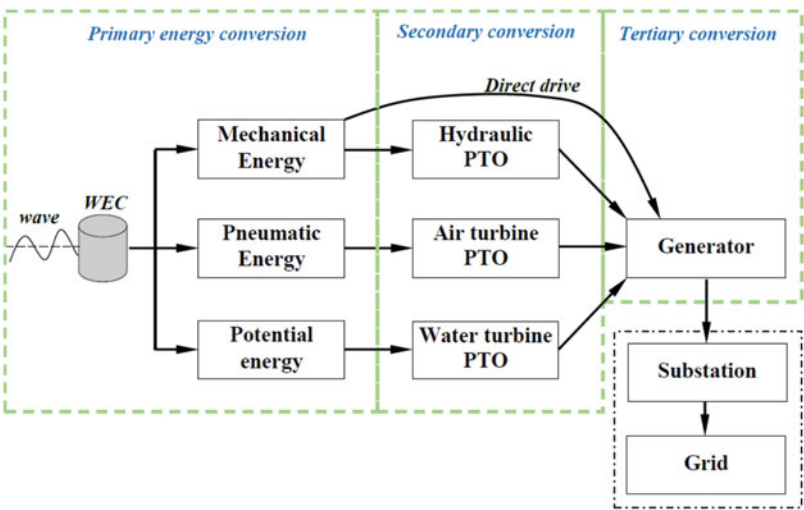
For different types of wave energy converters, the primary converted energy can be in different forms: in the oscillating water columns, it is pneumatic power trapped in the air chamber in the

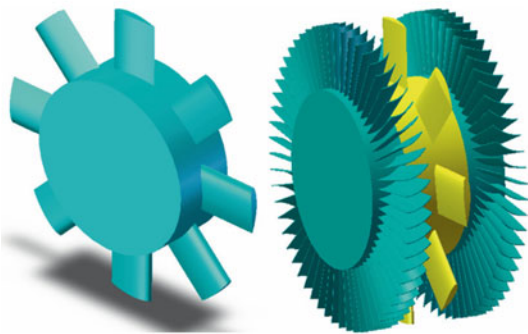
OWC devices, the potential power in the overtopping devices or the kinetic power of the body (or bodies) in the oscillating body (bodies) wave energy converters (see Fig. 1). To convert the different forms of the primary converted energy, different types of PTOs are employed: for pneumatic power, it is an air turbine, such as the Wells turbine and impulse turbine (Fig. 2a (Falcao and Gato 2012)); for potential energy, low head water turbines are used (Fig. 2b, the Kaplan low head water turbine in Wave Dragon (Kofoed et al. 2005)); while for the body kinetic power, hydraulic PTO (Fig. 2c, Falcao 2010) or the direct drive PTO (Fig. 2d, Drew et al. 2009) can be used. These PTOs may be the most used PTO technologies, but we should not exclude other types of PTOs, for instance, the mechanical PTO and the recent PTO developments supported by WES (Wave Energy Scotland 2015).

Control Technologies

Control technologies for wave energy converters have been a hot and interesting topic of research work since the principle of the optimal control for improving wave energy absorption was identified in the 1970s (Budal and Falnes 1977), and since then researchers have been seeking different control technologies to solve the problems of low wave energy conversion efficiency, especially for those small devices (small devices have been

Power Take-Off System,
Fig. 1 Basic process of wave energy conversion

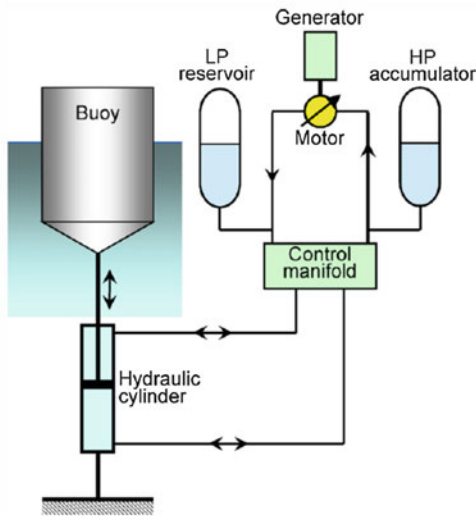




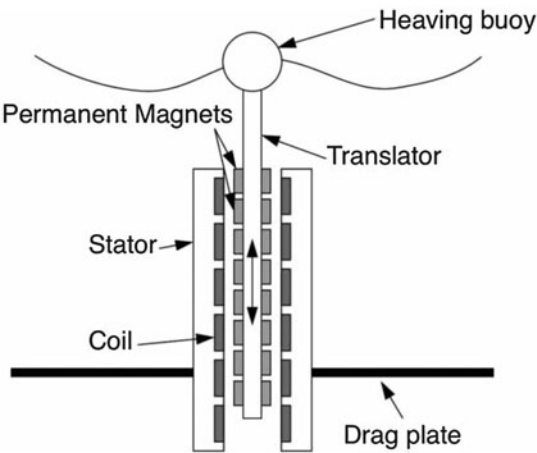
(a) Wells and impulse turbines [18]



(b) Kaplan turbines for WaveDragon [19]



(c) sketch for hydraulic PTO [20]



(d) Direct drive [21]

Power Take-Off System, Fig. 2 Conventional power take-offs for wave energy conversion.

preferred due to their robustness in structure, and also the smaller the device, the higher the ratio of the capture power over the mass according to (Falnes 2002)). A good review on controls for improving wave energy conversion can be found in (Ringwood et al. 2014).

In the previous sections, it has been shown that the maximum energy absorption by a wave energy converter occurs when the device is in resonance. To maximize wave energy conversion, the PTO control system could provide an additional mass or a negative spring to change the converter resonance period, i.e., the wave energy system has a

cancelled intrinsic impedance. More specifically, with setting or changing of the PTO parameters (i.e., the PTO control, see Eq. (3)), the maximal wave energy conversion can be achieved through the device resonance with the incident wave and the optimal damping level of PTO (Evans 1981; Falnes 1980). While in regular waves such an optimal control may be very straightforward, it becomes much more complicated in irregular waves since every wave cycle in irregular waves has different frequencies and amplitudes, and it is practically unknown what would be in the next wave cycle. This applies a challenge for the

control systems since they need the future information to decide the control parameters and to allow the time for the system to make the control. For instance, a latching control through peak-matching requires the information when is the peak in the future. As shown in (Fusco and Ringwood 2012), most of the active control system needs the future information of a time horizon more than 20s, and an accurate prediction of such a long time horizon is really challenging. This essentially becomes one of two barriers in practical control applications for many wave energy converters controls (the other barrier is the physical implementation of the control system, for instance, how the control system provides the required inputs of large forces to make the control working. In (Hals et al. 2011), these two challenges are specially attributed to latching control, but in fact, they are the common challenges for most control technologies: to make the control optimal, the required future information must be known to the control algorithms, and an ideal control system may be required to provide a large force in a rather swift manner). To remove the barriers partially or fully and make the control more practical, different control technologies have been proposed and attempted. For instance, Falcao (2007) and Sheng et al. (2015) developed the different latching control technologies. In these two novel latching controls, the controllers lock the devices when the device motion velocity vanishes, which are same as in conventional latching control technologies, but both latching controls remove the requirement of the future event prediction. In the former method, it is proposed to release the device when the excitation force is larger than a preset threshold, while in the latter method, a simple latching duration is calculated based on the wave statistics (the wave statistics can be reliably forecast hours ahead of real time or calculated from the wave measurement of the last 30 or 60 min. It is generally accepted that wave statistics may not change in 3 h or a longer period). As such, the barrier for requiring the prediction of the future information is removed!

To date, many different types of control technologies have been proposed and studied, and

more are being proposed and/or developed. Ozkop and Altas (2017) have reviewed the literature and summarized that more than 30 different type control technologies. The most popular controls in the summary include (short names in the brackets) following the popularity order in the reference: phase control (PhC), latching control (LC), optimal control (OC), PI control (PIC), predictive control (PC), on-off control (OOC), power control (PoC), valve control (VC), reactive control (RC), etc. It should be seen that although different names are used in the control technologies, some of them may be same in principle. Strictly speaking, a full optimal control (OC) is the ultimate goal for all control technologies, by achieving both the phase and amplitude control conditions, such as the predictive control (PC), which is essentially seeking the optimal conditions in the control method. Alternatively, sub-optimal conditions are made. For instance, many control technologies are implemented the full or partial phase optimal conditions, including the phase control (PhC), reactive control (RC), latching control (LC), on-off control (OOC), etc.

In a rather detailed comparison, Hals et al. (2011) present a conventional optimized resistive loading (RL) and five popular wave energy control technologies. The former is essentially an optimized PTO damping for the given waves, which provides a base for comparison. The compared control technologies include the classic phase control by (i) latching (Falcao 2008; Babarit and Clement 2006) and (ii) clutching (Babarit et al. 2009) and the advanced active controls, (iii) the approximate complex-conjugate control (Nebel 1992), (iv) tracking of approximate optimal velocity (Falnes 2002), and (v) model predictive control (MPC) (Cretel et al. 2011; Brekken 2011). From the comparisons, it can be seen that all the controls are superior to the pure optimized PTO resistive loading (RL), with the MPC control being the best control since in all the compared cases, MPC gives highest energy conversion, generally 200% more capture energy when compared to RL, and also for longer waves, the controls could produce more wave energy. It should be noted that the comparisons are made for wave energy period $T_e = 6$ s, 9 s, and 12 s, all longer

than the natural period of the semisubmersed sphere, $T_0 = 4.4$ s.

Cross-References

► Wave Energy Converters

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Power Transmission and Distribution

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Synonyms

PDU, Power distribution unit; PTD, Power transmission and distribution

Definition

Deep-sea mining system is a system that can excavate, collect, and transport minerals from the seabed to the mothership (Liu et al. 2014).

System Brief Introduction

The system that combines deep-sea mining vehicles with pipeline transportation system is widely regarded as the most promising commercial mining system at the moment. This mining system is mainly composed of deep-sea mining vehicle,

mineral transportation system, surface support system, power transmission, and control system. The power transmission system mainly provides the power for deep-sea mining vehicles, mineral transportation system, and surface support system. It is an important part of the whole mining system and is the basic condition of whether the whole mining system can work properly. It can be regarded as the “nervous system” of the mining system, so it requires very high reliability.

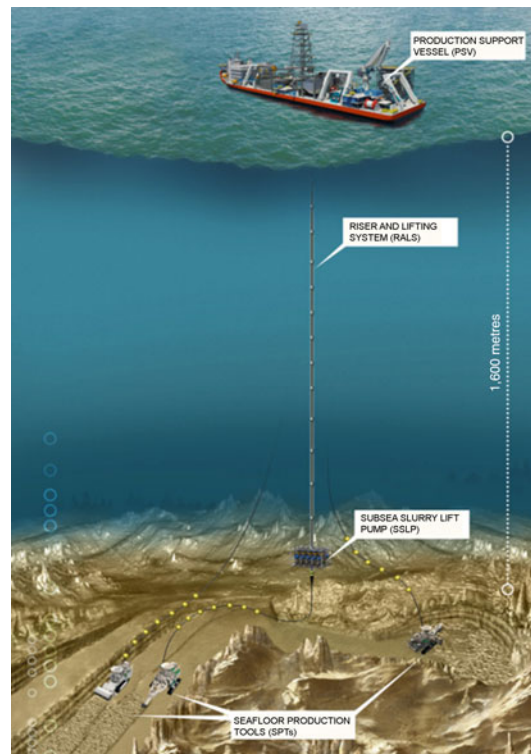
Figure 1 shows a typical mining system.

Surface Support System PTD

Surface Support System Composition

The surface support system is mainly composed of the following equipment:

- Mining vehicle launch and recovery system.
- Umbilical launch and recovery system.



Power Transmission and Distribution, Fig. 1 Typical mining system

- Flexible pipe launch and recovery system.
- Heave compensation system.
- Pipe lift system.
- Pipe storage and transfer system.
- Pipe docking and disassembly system.

Surface Support System Power Requirements

There are several kinds of electrical equipment voltage of surface support system: three-phase AC690 V, three-phase AC380 V (50 Hz) or AC440 V (60 Hz), and single-phase AC220 V. These types of voltage can be output directly by the ship generator, and the distance from the ship's distribution cabinet to the electrical equipment is not very far so low voltage can be used directly for power transmission.

Underwater PTD System

The power transmission and distribution equipment of the underwater mineral transportation system and mining vehicle covers a large area and is usually installed on deck, but it can't meet the waterproof requirements if directly installed on the deck. Also it is dangerous to expose the high-voltage equipment on the deck, so it is required to be installed in a container with a protection grade of at least IP65.

The output of ship generator is usually AC low voltage, while the input voltage of underwater motor is generally AC high voltage, so the ship electrical output needs to pass through a step-up transformer to obtain the high voltage required by the motor. The voltage of underwater instruments and control systems are usually AC220 V, AC110 V, DC24 V, and DC5 V. These equipment power can't be transported directly through the low voltage so it also needs to be transported

through high voltage. The high voltage is converted to the required AC220 V and AC110 V under the water. The DC24 V and DC5 V are available from AC110 V via PSU.

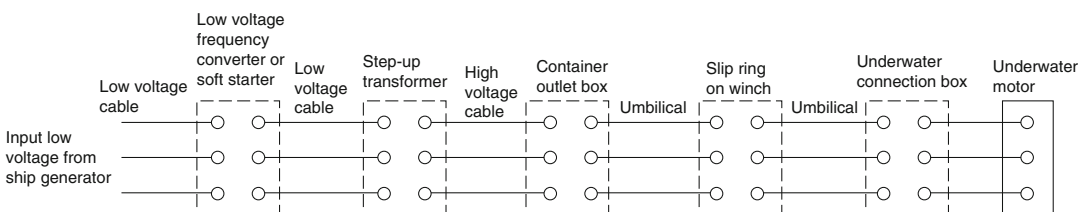
For lift pump, hose pump, and mining vehicle, the starting current of the motor is large, so soft start equipment is needed. The lift pump needs to adjust speed so frequency converter soft start is needed. The high-voltage frequency converter covers a large area so it is not suitable to install in the container. We can choose the low-voltage frequency converter. The motor of hose pump and mining vehicle has no speed regulation requirement, so we can choose low-voltage thyristor soft starter for economic reason. Buffer station hydraulic station motor power is small and can start directly. The input power from ship is connected to the step-up transformer through the frequency converter or soft starter, and the high-voltage output terminal of the transformer is connected to the outlet box in the container through the high-voltage cable. The high-voltage power and communication fiber is combined together in the outlet box and then connected to the slip ring in the umbilical winch through umbilical. The power is transported to the underwater motor by umbilical via slip ring.

Figure 2 is a road map of underwater equipment power supply, and Fig. 3 is a schematic diagram of underwater equipment power supply.

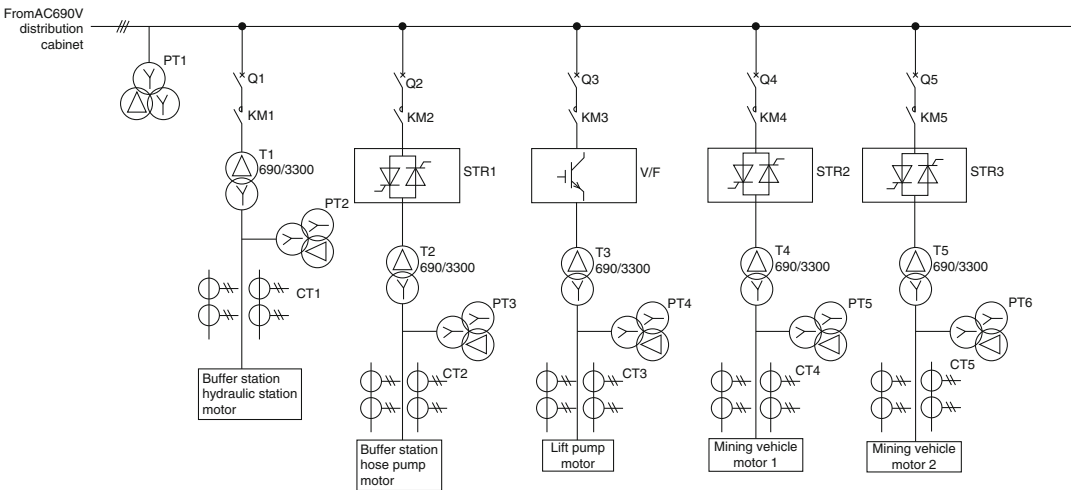
Key Technology of Underwater PTD

Deep-Sea Long-Distance High- Efficiency PTD Technology

Ship power supply is different from land power supply; the electric power is supplied by ship generators. The power network capacity is generally small and has poor impact resistance. Heavy



Power Transmission and Distribution, Fig. 2 Road map of underwater equipment power supply



Power Transmission and Distribution, Fig. 3 Schematic diagram of underwater equipment power supply

use of nonlinear loads on ships results in a large number of harmonics. It not only seriously affects the power quality of the ship power grid but also interferes the normal operation of other electrical equipment. The current generated at the start of high-power load is as high as four or seven times the rated current; it will have a huge impact on the ship power grid when it starts. On the other hand, because the underwater equipment can work at a depth of 6,000 m (Wang and Wang 2007), such a long-distance transmission of electric power will cause a serious voltage drop of the line, and this will make the motor not stop because the start voltage is too low. Therefore, how to ensure the reliable transmission of underwater electric power over a long distance needs serious consideration. In addition, the underwater load is far away from the power supply, the transmission loss will be very high if the current is too large. But if the voltage is too high, there will be some problems in the manufacture of umbilical. So there are many factors that need to be considered in selecting the transmission voltage level.

Online Monitoring and Diagnosis Technology for Insulation and Grounding Circuit of Deep-Sea Circuit

Whether the key components such as power supply, umbilical, and transformer can work normally will directly affect the safe and reliable operation

of transmission and distribution system. Most of the insulating materials are organic materials, which are prone to deterioration due to various factors such as electricity, heat, machinery, environment, and so on, resulting in insulation aging, especially in the harsh environment of the ocean, which is very corrosive and complex. Therefore, it is necessary to have a set of reliable scheme to monitor the equipment insulation and line grounding condition in real time so as to detect power supply failure in time, avoid endangering personal safety, and ensure the safe and reliable operation of the whole system.

Development of Deep-Sea Optoelectronic Composite Umbilical

Deep-sea optoelectronic composite umbilical is the “nerve and lifeline” of underwater system; it connects electrical equipments on ship and underwater production facilities to transmit the ship power and all communication signals to deep-sea mining equipment. In the deep sea, the umbilical is affected by the chemical corrosion of sea water, ocean current movement, sea bottom biological impact and bite, and gravity of relay stations and mining vehicles, which can easily cause functional units and tensile failure. So it's important to develop high reliability and long life cycle composite umbilical.

Binding Technology of Umbilical

The umbilical that transmits power to the lift pump and buffer station must be bound and fixed on the pipe; otherwise the umbilical will swing violently under the influence of the current, which will not only produce a huge pulling force on the winch and the underwater connection box on the ship. It will also collide violently with the delivery pipe, which will easily lead to the breakage of the umbilical. The binding scheme should meet the requirements of fast operation on ship, and the binding material should be corrosion-resistant in sea water, have enough friction to fix itself on the pipe, and have enough load bearing capacity to hold the umbilical.

Underwater PTD Mode

The main load of underwater system is lift pump motor, buffer station hose pump motor and hydraulic station motor, and mining vehicle motor. Motor power of lifting pump and mining vehicle is very high, for example, the power of lifting pump motor is 800 kW in the Korean mining experiment project, and the power of mining vehicle motor is 550 kW. The power of underwater mining vehicle motor is even more than several MW in the mining system of Nautilus Mining Company of Canada. The water depth of deep-sea mining is generally more than 1000 m. For example, the “Blue Nodule” project in Europe is

Power Transmission and Distribution, Table 1 Comparison of different transmission modes

Sequence number	Index name	AC low voltage transmission	AC high voltage transmission	DC high voltage transmission
1.	Line current	Very big	Small	Small
2.	No load voltage drop	Big	Small	Very small
3.	Full load voltage drop	Very big	Small	Small
4.	Terminal voltage control	Complicated	No	Complicated
5.	Line reactive capacity	Common	Common	No
6.	Transmission distance	Near	Far	Far
7.	Transmission cable number	3 for 3 phase or 2 for single phase	3 for 3 phase or 2 for single phase	2
8.	Transmission cable diameter	Very big	Small	Small
9.	Transmission voltage level	Low	High	High
10.	Transmission starting equipment	No	Step-up transformer	Step-up transformer and rectifier
11.	Transmission terminal equipment	No	No or step-down transformer	Underwater inverter or inverter and rectifier
12.	Line loss	Very big	Small	Small
13.	Equipment area	Small	Small	Big
14.	Technology reliability	High	High	Common
15.	Technology maturity	High	High	Common
16.	Transmission cable cost	High	Low	Very low
17.	Transmission equipment cost	Low	Common	High
18.	Maintenance	Simple	Relatively simple	Complicated

designed with a maximum depth of 6000 m, and four lifting pumps are used. If we use the low voltage from the ship generator to the transmission of such a long distance of high power, the current will be very large and the line voltage drop will be severe, and the underwater motor will not be able to start. Table 1 is the comparison of three common modes of high electric power transmission in long distance.

As can be seen from the above table, when the three-phase or single-phase underwater equipment is in long-distance, high-capacity transmission, the AC high-voltage mode is the best scheme. The transmission mode is simple, the efficiency is high, the cost of transmission equipment is relatively low, and the technology is mature. So the input voltage of deep-sea motor is generally three-phase AC300 0 V at present, while the higher power motor can use three-phase AC600 0 V.

Underwater Motor Start Mode

The capacity of the ship power supply network is very small compared with the land power supply network. The starting current of the high-power motor is very large when it starts directly which will have a serious impact on the whole ship power supply, so the underwater motor should choose the proper starting mode. There are several general starting methods of AC motor as follows: frequency converter start, thyristor soft start, and direct start. Table 2 is the comparison of three different motor start methods.

As can be seen from above table:

- (a) Direct start has a great influence on input power supply so it is only suitable for low power motor.
- (b) It is necessary to use thyristor or frequency converter to start the heavy-duty motor so as to reduce the impact on the ship power grid.

Power Transmission and Distribution, Table 2 Comparison of different motor start methods

Sequence number	Index name	Frequency converter start	Thyristor soft start	Direct start
1.	Start current	Small	Small	Can be as big as 4 to 7 times rated current
2.	Start torque	Big	Small	Big
3.	Applicable load	All loads	Light load or no load	Big system
4.	Output voltage	Continuous adjustable	Continuous adjustable	Can't adjust
5.	Output voltage harmonic	Low	High	No
6.	Input power factor	High	Common	Low
7.	Influence on ship power	Low	Common	Big
8.	Speed regulation function	Yes	Yes	No
9.	Speed regulation performance	Excellent	Common	No
10.	Control difficulty	Complicated	Simple	Simple
11.	Area	Big	Small	Small
12.	Reliability	High	High	High
13.	Maturity	High	High	High
14.	Cost	High	Common	Low
15.	Maintenance	Complicated and high cost	Relatively simple and common cost	Simple and low cost

- (c) We can use frequency converter to start and control speed when the heavy-duty motor has speed regulation requirement.

The motor of the lifting pump has a large power and needs to adjust the speed according to the working conditions so it is suitable to start with the frequency converter. Relay station hose pump motor and mining vehicle motor have a large power but do not need to adjust the speed so it is suitable to start with thyristor soft start. Relay station hydraulic station motor power is small so it can start directly.

PTD Equipment

PDU

PDU is used to switch on and switch off the circuit; it also can show the circuit parameters such as voltage, current, power, and so on. A PDU mainly contains breaker, contactor, meters, protection devices, and other auxiliaries. Breaker is a kind of switchgear which can not only turn on and off normal load current and overload current but also switch on and off short circuit current. In PTD system, there are two types of breakers widely used, molded case circuit breaker and air circuit breaker. Molded case circuit breaker rated voltage from AC38 0 V to AC69 0 V and rated current from 16A to 1600A. Air circuit breaker rated voltage from AC38 0 V to AC115 0 V and rated current from 630A to 6300A. Figure 4 shows a PDU. Figures 5 and 6 are typical molded case circuit breaker and air circuit breaker.

Contactor is a kind of electrical appliance used to switch on and break AC/DC main circuits and large capacity control circuits frequently, rated current from 9A to 2600A and rated voltage from AC22 0 V to AC100 0 V. Figure 7 shows a typical contactor.

Transformer

Transformer is a device that uses the principle of electromagnetic induction to change the AC voltage. The main components are primary coil, secondary coil, and iron core. Main functions are



Power Transmission and Distribution, Fig. 4 PDU

voltage conversion, current conversion, impedance conversion, isolation, voltage stabilization, etc. There are generally three types of transformers used in deep-sea mining system. One is three-phase or single-phase step-up transformer to convert the low voltage from ship generator into high voltage required for underwater equipment. One is single-phase transformer to convert single-phase high voltage that transported to underwater into single-phase low voltage. One is low-voltage transformer which converts the low voltage from ship generator into other required low voltage. Figures 8, 9, and 10 show typical transformers of these three types.

Current Transformer

Current transformer is an instrument for measuring primary side high current via secondary side small current according to the electromagnetic induction principle. It is composed of closed iron core and winding. Low-voltage current transformer and high-voltage current transformer are



Power Transmission and Distribution, Fig. 5 Molded case circuit breaker



Power Transmission and Distribution, Fig. 6 Air circuit breaker



Power Transmission and Distribution, Fig. 7 Typical contactor



Power Transmission and Distribution, Fig. 8 Three-phase step-up transformer

commonly used in power transmission and distribution system. Figures 11 and 12 are typical low-voltage current transformer and high-voltage current transformer.



Power Transmission and Distribution, Fig. 9 Single-phase step-down transformer



Power Transmission and Distribution, Fig. 11 Low-voltage current transformer



Power Transmission and Distribution, Fig. 10 Low-voltage transformer



Power Transmission and Distribution, Fig. 12 High-voltage current transformer

Potential Transformer

Potential transformer is an instrument used to convert the primary side high voltage of the line into the standard low voltage of the secondary side. It is used to measure the line high voltage via the secondary side low voltage. Figure 13 is a typical potential transformer.

Frequency Converter

Frequency converter is a kind of electric power control equipment which uses frequency



Power Transmission and Distribution, Fig. 13 Potential transformer

conversion and microelectronic technology to control AC motor by changing the frequency of motor working power supply. Frequency converter is mainly composed of rectifier, filter, inverter, brake unit, drive unit, detection unit, micro-processing unit, and so on. Frequency converter adjusts the voltage and frequency of output power supply by switching on and off internal IGBT. It provides the required power supply voltage according to the actual needs of motor, thus achieving the purpose of saving energy and adjusting speed. Figure 14 shows the typical low-voltage frequency converter.

Soft Starter

Soft starter is a kind of motor control equipment which integrates soft start, soft stop, light-load energy-saving, and multifunction protection. The soft starter uses a three-phase opposite parallel thyristor as a voltage regulator and connected between the power supply and the motor stator. When starting the motor with soft starter, the output voltage of the thyristor increases gradually, the motor accelerates gradually until the thyristor



Power Transmission and Distribution, Fig. 14 Low-voltage frequency converter

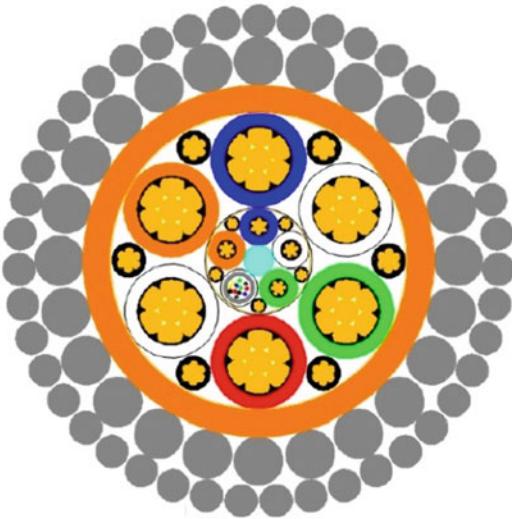
is fully switched on, and the motor works on the mechanical characteristics under rated voltage to achieve smooth start. The starting process ends when the motor reaches the rated speed, and the soft starter automatically replaces the thyristor with a bypass contactor to provide the rated voltage for the normal operation of the motor. Figure 15 shows the typical low-voltage soft starter.

Umbilical

Umbilical is a combination of conductor (power conductor or signal conductor or both), optical fiber (single mode or multimode), insulation material, and outer sheath (steel or Kakra). The main parameters include rated voltage, rated current, outer diameter, minimum bending radius, air weight, sea water weight, safe working load, minimum breaking strength, etc. Figure 16 is a typical section of umbilical.



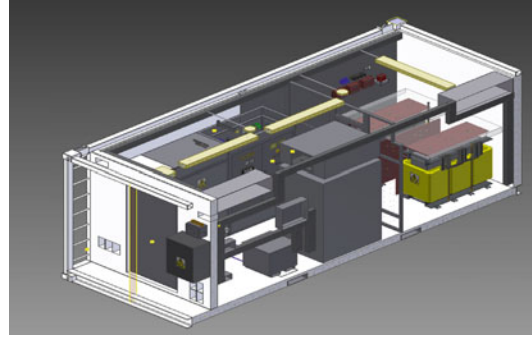
Power Transmission and Distribution, Fig. 15 Low-voltage soft starter



Power Transmission and Distribution, Fig. 16 Typical umbilical sectional view

Offshore Container

Offshore container is used to install the PDU, transformer, inverter, and other electrical devices or control system. It usually has IP56 protection



Power Transmission and Distribution, Fig. 17 Typical layout of container

level and is installed on the deck. The transformer and inverter generate mass of heat when in operation so it's very important for the container to cool down its internal temperature. Figure 17 shows typical internal layout of the container.

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Preliminary Design

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Synonyms

Concept design; Design of submersibles; Design spiral; Detailed design; Empirical design; Hydrodynamic design; Multidisciplinary design optimization (MDO); Optimal design; Reliability based design (RBD); Structural design; Technical design

Definition

Preliminary design of a submersible is a stage of the design of a submersible. The process of the design of a submersible may be referred to in chronological order as the (1) pre-design, (2) design, and (3) post-design, and then the design process is composed of three design phases: (1) basic design, (2) contract design, and (3) detail design. The basic design consists of concept design and preliminary design stages, and preliminary design is the second stage of basic design. Starting with baseline data provided by the selected concept design(s), preliminary design refines and firms up the significant characteristics of systems, constraint, and relative cost estimates. Consequently, it provides a precise definition of the mission system and assurance that the design goal can be attained.

Scientific Fundamentals

Introduction to the Design Stage

As noted, the design process is composed of three phases called as basic design, contract design, and detailed design shown in Fig. 1. In this stage, the levels of labor, preparation time, and costs associated with each phase increase exponentially as the process progresses. Design creativity and flexibility are essentially limited to the basic design phase – the mission system, including its submersible system, being well defined at the beginning of contract design.

The design process, at this point, is discussed from the perspective of the mission system. As has been seen, this system is composed of (I) systems which may be labeled as new construction or existing systems. Design efforts focus on the total or “from scratch” design of new construction systems and on the selection of suitable existing systems and their alteration, if required, to convert them to (I) systems. The submersible, herein, is considered to be a new-construction (I) system (Allmendinger 1990).

The basic design is the primary design phase concerned with how best to accomplish what the

potential user/owner of the mission system wants to do underwater as set forth by his mission requirements. Basic design, as considered herein, is composed of the concept design and preliminary design. Concept design develops two or more design alternatives from which design is selected in accordance with specified criteria. Preliminary design advances the selected alternative’s development to the point of entry into the contract design stage. Basic design proceeds, in general, as discussed above, its process is discussed further in the next section.

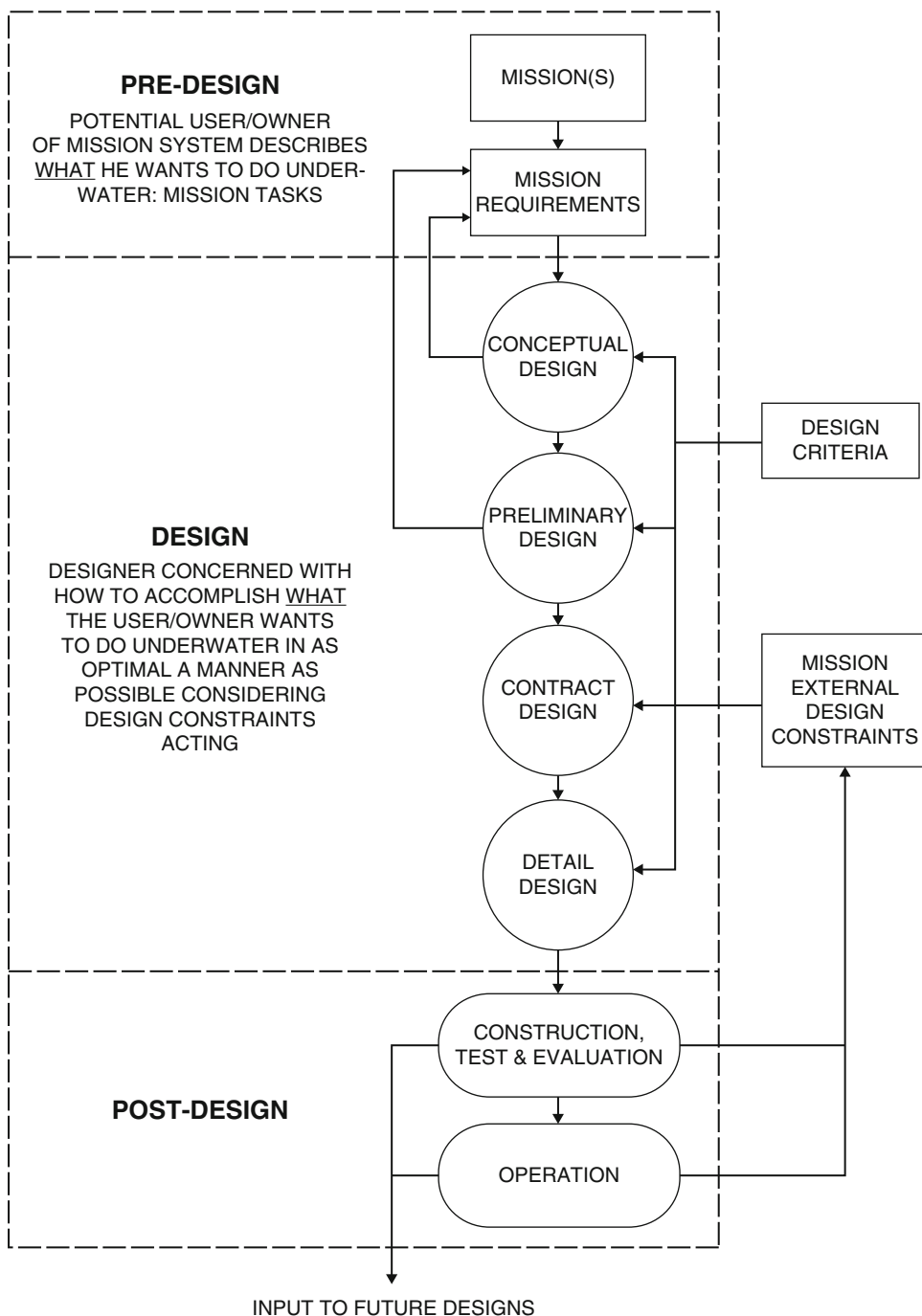
Contract design requires yet further refinement of design and additional detail. It yields contract plans and specifications necessary for interested parties to bid on the construction of new (I) systems or the alteration of existing (I) systems. It also provides contractual documents for the construction and alteration work. Specifications delineate quality standards of material and workmanship as well as setting forth performance expectations for the (I) systems and their subsystem. They also describe tests and trails which must be performed successfully before acceptance of the systems.

Detailed design is the final phase of the design process and entails the development of detailed working plans from which the (I) systems are constructed or altered. In one sense, it is not a design phase since all the creative design effort is made in preceding phase, the design being unequivocally defined prior to entering this phase. It does, however, require the greatest amount of work of all the phases and is often undertaken by entities building or altering the system (Allmendinger 1990).

Missions in Basic Design Phase

As discussed above, the basic design phase leads the process of design, and the missions of this stage, in general, involves (Allmendinger 1990):

1. Development of performance requirements or (I) system capabilities, from the mission requirements.
2. Determination of the principal characteristics of the (I) systems required to achieve these



Preliminary Design, Fig. 1 Design and associated processes (Allmendinger 1990)

- capabilities, which enable new construction systems to be designed and existing systems to be selected and altered if necessary.
3. Estimation of capital and operating costs of (I) systems.
 4. Identification of one or a few mission systems from among a series of alternatives which optimally meet performance requirements to the extent possible considering design constraints acting.
 5. Refining and firming-up characteristics and cost estimates of the (I) systems composing the optimum concept design(s).

The concept design stage of basic design is concerned with the first four of these functions – function four, herein, is based on cost-effectiveness as the optimization criterion. The preliminary design stage of this phase is concerned with the fifth function.

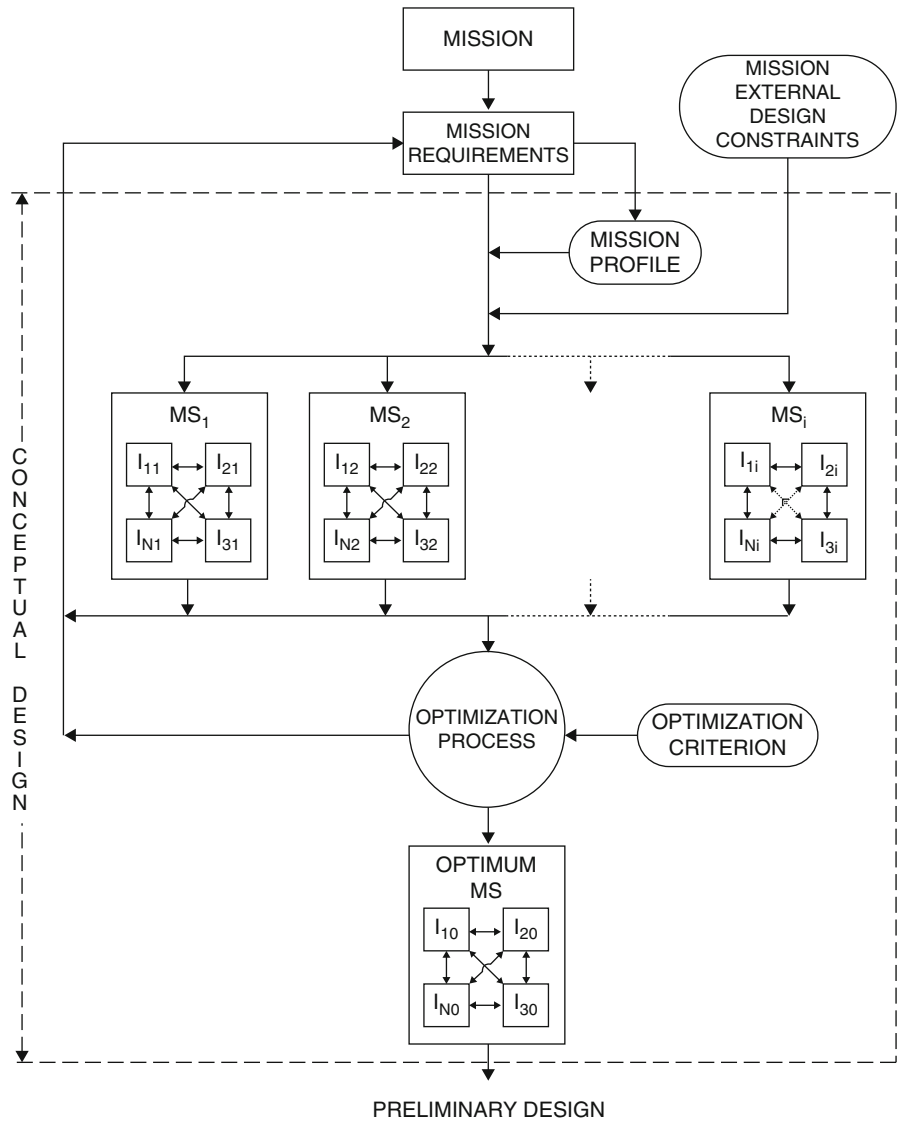
Concept design, diagrammed in Fig. 2, is the first attempt to translate all the user/owner's mission requirements into performance requirements and characteristics of the (I) systems composing the mission system. It is often viewed as consisting of feasibility studies and completion of the optimum conceptual design (s) once they are identified by these studies. Feasibility studies are concerned with the development of those performance requirements and associated system characteristics which have significant impact on the cost-effectiveness optimization process. Completion of the optimum design(s) involves developing other major performance requirements and characteristics those which are essential in defining (I) systems but which do not have a significant impact on the optimization process (Allmendinger 1990).

Feasibility studies create an orderly series of mission system alternatives, all of which meet the mission requirements and are technically feasible, and select one or a few of these alternatives as the conceptual design(s) to be carried on into preliminary design. These alternatives are shown in Fig. 2 as $MS_1, MS_2, \dots, MS_i$, where "i" can be any small to large number depending on the simplicity/complexity of the mission and

optimization techniques used. As indicated in the figure, each alternative is composed of a unique combination of (I) systems. Each (I) system, in turn, has unique capabilities, characteristics required to achieve these capabilities, and costs associated with providing these characteristics. Certain unique (I) systems may appear in more than one combination, but individual combinations are not duplicated in other alternatives. Mission system alternatives may be generated by (1) varying performance requirements with the cascading effect of varying (I) systems' capabilities, characteristics, and costs and (2) varying (I) systems' capabilities, characteristics, and costs in ways each of which meets a specific set of performance requirements. Alternatives generated in this manner contain information necessary to conduct the search for the optimum, cost-effective mission system(s).

Combinations of (I) systems forming mission system alternatives may vary in number, type/capabilities, and status. These features are related to functions required to accomplish mission tasks, the time frame for completing these tasks, if critical, and economic considerations. A mission profile, derived from mission requirements, is useful in providing functions and time-frame information – its usefulness increasing as the complexity of the mission increases. This profile is a chronological listing of all events occurring from the mission's beginning to end and including time-frame data where necessary.

The other "exit line" from the optimization process, as indicated in Fig. 2, leads to the optimum mission system(s) identified by this process in a manner suggested by the cost-effectiveness equation. The concept design(s) of the selected mission system(s) can now be completed by developing the primary characteristics of the (I) systems not considered in the feasibility studies. The identification of more than one optimum concept design means that the "optimization curve" is reasonably flat over a limited range of alternatives – that there is little to choose between them. In this instance, the user/owner may select one of them based on subtleties not heretofore considered or more than one may be carried



Preliminary Design, Fig. 2 Basic design of mission system (Allmendinger 1990)

forward into preliminary design. The mission system(s) carried forward should have associated performance requirements firming up and provide baseline (I) system characteristics which are further developed and during the preliminary design stage.

Preliminary design, as noted, is the second stage of basic design and is concerned with this phase's fifth function. Starting with baseline data provided by the selected concept design(s), preliminary design refines and firms up the major

characteristics of systems, constraint, and relative cost estimates. Consequently, it provides a precise definition of the mission system and assurance that the design goal can be attained.

Figure 1 shows a "feedback line" from preliminary design to the mission requirements, indicating that refinements in the design may reveal technical feasibility reasons for altering performance and, in summary, mission requirements that were overlooked in concept design. These discoveries should be few in number and minor

in their effect on the requirements. A significant discovery of this nature would, most likely, raise doubts about the validity of selected concept design(s) and be the cause for repeating the concept design stage or terminating the design altogether (Allmendinger 1990).

Regarding basic design, the major systems comprising submersible are summarized in the following outline (Allmendinger 1990):

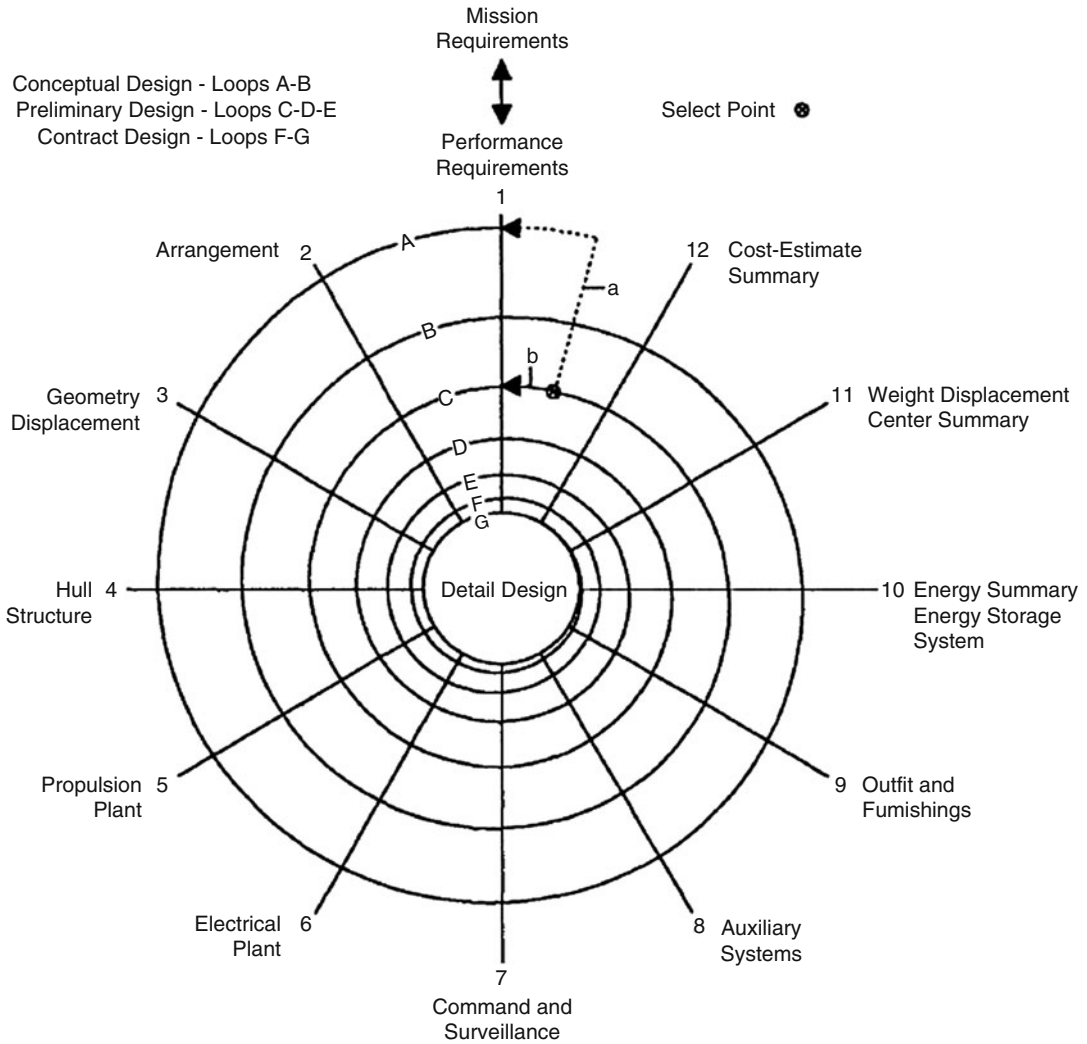
1. Hull Structure System: (a) Structure System, (b) Special System.
2. Propulsion Plant System: (a) Energy Generating System, (b) Propulsion System, (c) Special Purpose System.
3. Electrical Plant System: (a) Electrical Power Generation System, (b) Power Distribution System, (c) Lighting System, (d) Special Purpose System.
4. Command and Surveillance System: (a) Command and Control System, (b) Navigation System, (c) Communication System, (d) Surveillance System, (e) Special Purpose System.
5. Auxiliary System: (a) Human System, (b) Life Support System, (c) Air, Gas and Miscellaneous Fluids System, (d) Submersible Control System, (e) Mechanical Handling System, (f) Special Purpose System.
6. Outfit and Furnishings System: (a) Fittings System, (b) Hull Compartmentation System, (c) Preservatives and Covering System, (d) Furnishings, (e) Special Purpose System.
7. Armament System (This system applies only to submersibles having offensive/defensive military missions requiring the launching of weapons). The above introduction mostly came from Allmendinger (1990) which was based heavily on Busby (1976). More detailed introduction from Busby (1990) and Funnel (1999).

The Process of the Preliminary Design

This section discusses the process of the preliminary with the well-known design spiral which serves as a basis for a brief discussion of the entire design process. The design spiral is shown in Fig. 3.

The design spiral – The iterative nature of the overall design process may be modeled by the design spiral shown in Fig. 3. The spiral consists of spokes and loops. The spokes represent design considerations including performance requirements, submersible characteristics, and cost estimates; submersible characteristics are embodied in its arrangement, geometry, six systems groups, and summaries of energy requirements and weight/displacement centers. The loops indicate design iterations with the design's refinement increasing as the loops spiral inward. Outer loop procedures use empirical formulations and estimates based on similar design data, whereas middle and inner loop procedures are based on the progressively more detailed development of the design considerations utilizing increasingly precise formulations. The intersection of spokes and loops form "points" which are identified alphanumerically. Thus, for example, 3B is a point in the spiral at which the geometric characteristics of the submersible are being considered in the second iteration in a more detailed manner than they were initially at point 3A. At 3B, adjustments are made as the result of both the more detailed consideration of the geometric characteristics required and the impact on geometric of interrelated design considerations for which data were generated at points in spiral preceding 3B. Adjustments are made at other points in the spiral for the same reasons. For the design to converge to a satisfactory solution of the design problem, these adjustments must become progressively similar with each succeeding loop through concept, preliminary, and contract design. This fact is demonstrated in the spiral by the decreased spacing between loops as they spiral inward. Convergence is illustrated by the innermost loop becoming a circle, indicating that all design considerations are fully developed and required no further adjustments. At this point in the spiral, detailed design can begin (Allmendinger 1990).

Preliminary design is arbitrarily limited to loops C, D, and E. Note in the Fig. 3 that loops D and E are relatively close together, indicating that only very small adjustments in design



Preliminary Design, Fig. 3 Design spiral (Allmendinger 1990)

considerations are being made at this point. Should preliminary design's more detailed procedures and exact formulations reveal the need for relatively large adjustments to be made in one or more of the considerations, the validity of the selected concept design alternative may well be questioned. The end result of preliminary design is a well-defined submersible system which meets performance requirements and for which a firm cost estimate exists. As indicated on the spiral, completion of this phase of design marks the beginning of contract design.

The C, D, and E loops – The preliminary design procedures of these loops progressively refine all aspects of the submersible's design, including those aspects not considered in the concept design phase. Only small adjustments, if any, are required at the spiral's point at the end of the E loop. Some general comments on these points follow.

- (1) C, D, E Performance Requirements: Review these requirements in light of the design's more detailed development. C and D loop

procedures may reveal that it is still necessary or desirable to make small changes in some of these requirements. There should be no changes in them as the result of E loop procedures or, in words, they should be fixed at entry into the contract design phase.

- (2) and (3) C, D, E Arrangement and Geometry Displacement: Obtain a detailed view of the submersible with the location of systems being essentially fixed so that the systems can function properly, routine inspection and maintenance can be made with relative ease, there are no conflicts in space requirements, and the design condition criteria of adequate stability and zero trim are met. These procedures require detailed inboard/outboard profiles and cross-section views to be drawn. A line drawing may also be drawn depicting the envelope's geometry; this drawing is necessary to obtain accurate data required for hydrostatic and hydrodynamic calculations.
- (4–9) C, D, E Systems Groups: Develop these systems in sufficient detail to provide the assurance that, both individually and collectively, they will satisfy all performance requirements and very close estimates of sizing and cost data. Direct calculations and preliminary drawings, as contrasted with working drawings, are used extensively. Procedures, for example, may involve the construction and testing of structural and appended envelope models to verify data derived from theoretical strength and hydrodynamic formulations.
- (10) C, D, E Energy Summary-Energy Storage Systems: Make progressive refinements of electrical power-energy profile and pneumatic storage system based on increasingly precise C, D, E loop SWBS data. Electrical and pneumatic storage systems should be very accurately defined at point 10E.
- (11) C, D, E Weight-Displacement-Center Summary: Complete this summary in more detail and investigate stability and trim in design and other operating condition as well as in the emergency ascent condition. These procedures will require a breakdown of SWBS systems into their components.

Procedures at 11C and, to a lesser extent, at Point 11D usually indicate that some adjustments in arrangement or system details or both are necessary to attain design condition criteria of neutral buoyancy, adequate stability, and zero trim. They may also indicate that adjustments in envelop geometry and systems influenced by this characteristics are necessary to obtain adequate surface stability. These adjustments, if any, are to be made, should be relatively small on entry into the contract design phase.

- (12) C, D, E Cost Estimate Summary: Summarize the progressively refined first and annual operating cost estimates obtained in these loops. A close estimate of the submersible's operating cost per year should be in hand at the beginning of contract design (Allmendinger 1990).

Key Applications

Preliminary design is a significant step of the design of submersibles. Here we introduce the autonomous underwater vehicle (AUV) and remotely operated vehicle (ROV).

Autonomous Underwater Vehicles (AUV)

The applications of the underwater vehicle have shown a dramatic increase in recent years, such as, mines clearing operation, feature tracking, cable or pipeline tracking, and deep ocean exploration. To satisfy different applications, different kinds of development of autonomous underwater vehicles or their systems appear. Martz (2008) applied multidisciplinary design optimization method to the preliminary design of an AUV. Allotta et al. (2011) developed an innovative AUV called Tifone and finished the preliminary design, as is shown in Fig. 4. Salleh et al. (2013) performed preliminary design of AUV with higher resolution underwater camera for marine exploration.

Remotely Operated Vehicles (ROV)

In the design stage of ROV, in order to make it a free-flying vehicle, floatation is added to

Preliminary Design,

Fig. 4 Preliminary layout of Propeller and Thrusters on Tifone Vehicles (Allotta et al. 2011)

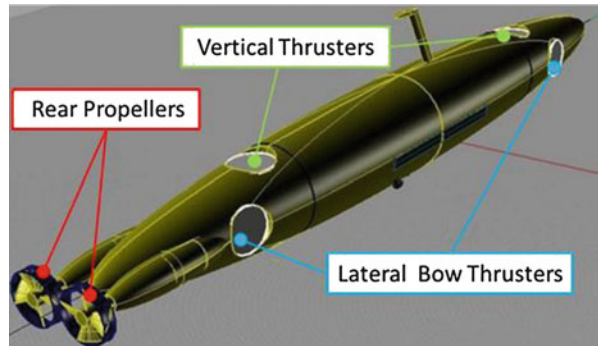
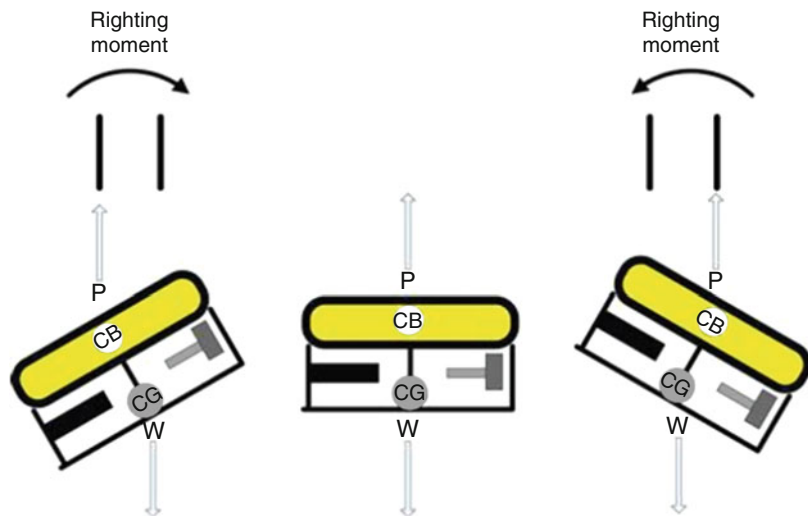
**Preliminary Design,**

Fig. 5 Vehicle's righting moment (Christ and Wernli 2014)



neutralize vehicle buoyancy and achieve mobility in Fig. 5 (Christ and Wernli 2014). Thus, one of the key points in preliminary design of an ROV is to realize the performance requirements and, at the same time, to keep its stability. Bowen et al. (2004) finished the preliminary design of a novel light-tethered hybrid ROV for global science in extreme environments. Govinda et al. (2014) studied modelling, design and robust Control of a ROV in preliminary design stage.

- Detailed Design
- Empirical Design
- Hydrodynamic Design
- Multidisciplinary Design Optimization (MDO)
- Optimal Design
- Reliability Based Design (RBD)
- Structural Design
- Technical Design

Cross-References

- Concept Design
- Design of Submersibles
- Design Spiral

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Preserve Heat

- [Thermal Insulation](#)

Pressure-Area Relationship

- [Ship-Iceberg Interactions](#)

Prism Based SPR (P-SPR)

- [Fiber Optic Hydrophone](#)

PRO - Pressure-Retarded Osmosis

- [Salinity Gradient Power Conversion](#)

Probabilistic Aspects for Ice Loads on Ships

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Introduction

For ships in the Arctic, ice loads caused by ship and ice interaction represent the dominant load. Ice loads on ship hull is random by nature as other environmental loads. The knowledge of ice loads can promote design and operation of ice-capable vessels in Arctic regions. However, due to the complexity of ship and ice interaction process, studies with respect to the ice loads are limited and still under development. Ice loads on ship hull can be calculated by empirical formula and numerical simulation and collected by laboratory experiments and full-scale field measurements. Among these methods, full-scale field measurements are the most effective way to study the properties of ice loads (Ehlers et al. 2015). In this entry, the randomness of ice loads and the reasons that cause the randomness of ice-induced loads are introduced. Previous studies for probabilistic descriptions and analysis of the full-scale measured ice loads on ship hull are summarized in this work. The challenges and future works for studying the stochastic properties of ice loads on ship hull are also described in this entry.

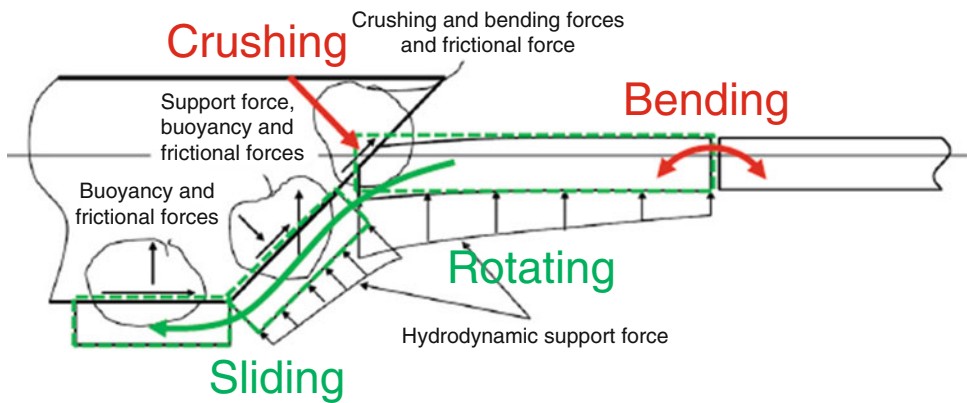
Randomness of Ice Loads to Ships

Ice loads are caused by the ship and ice interaction process, which depends on the ice conditions, the geometry of ship hull, and the relative velocity between the vessel and the ice features. Generally, for first-year ice, the ship and ice interaction process is initialized by a localized crushing of

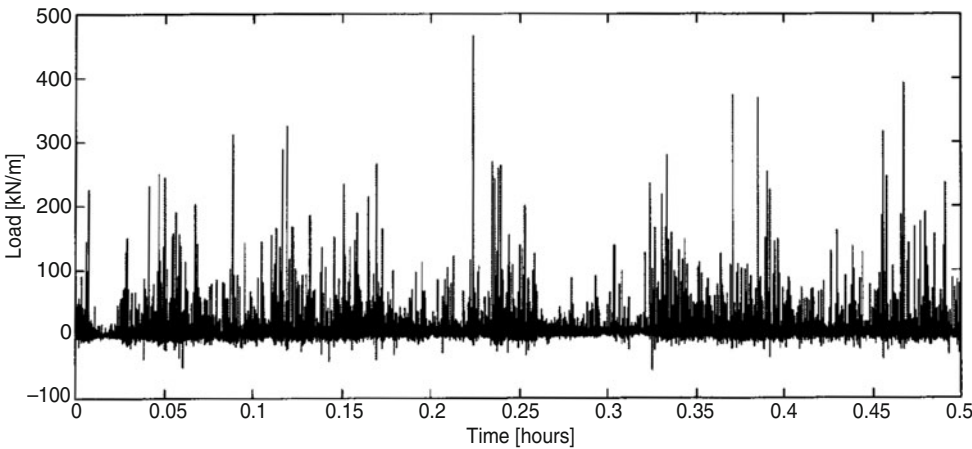
the free ice edge, and then the contact area and crushing forces increase as the ship advances and penetrates the ice features. The ice eventually deflects, and the bending stress promotes a flexural failure at a certain breaking distance from the crushing region (Jordaan 2001). The interaction process is illustrated in Fig. 1, and some other kinds of failure mode (e.g., creep, cracking, bucking) may occur separately or in combination during the interaction process. Even though the interaction between ice features and ship hull is governed by deterministic laws of mechanics, ice loads on ship hull are random by nature (e.g.,

Fig. 2). The stochastic nature ice loads have has also been validated by various full-scale field measurement campaigns, which are listed in Ehlers et al. (2015) and Suominen (2018).

During the full-scale measurement voyages in ice regions, strain sensors are instrumented to measure the (local) ice loads on ship hull. It is seen in Fig. 2 that the time series of measured ice loads on a frame looks like a sequence of impulses with sharp peaks, which indicate that during the ship and ice interaction process, the accumulated crushing force is high enough to initiate bending failure of the ice feature. Such impulses are



Probabilistic Aspects for Ice Loads on Ships, Fig. 1 An illustration of ship and (first-year) ice interaction process (Riska 2010)



Probabilistic Aspects for Ice Loads on Ships, Fig. 2 Stochastic nature of the ice loads measured on a frame (Lensu 2002)

repeated as the ice-breaking process continues and the peak values of the ice loads are referred to as the ice load peaks.

There are mainly two categories of sources causing the randomness of ice-induced loads. One is the variation of the ice conditions in the Arctic regions, which includes physical ice properties (such as ice types, thickness, density, porosity, floe size, etc.) and ice mechanical properties (e.g., flexural, tensile, shear, uni- and multiaxial compression strength, and so on). The other is related to the complex ship and ice interaction process in association with different ice failure mechanisms and randomness of flaw in ice features. In addition, ship operations and movements in different ice conditions also affect the ice-breaking process and the values of ice loads.

Probabilistic Analysis of Ice Loads to Ships

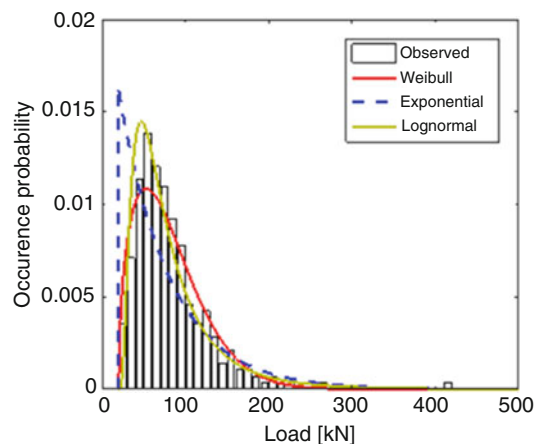
For stochastic processes, probabilistic models and methods should be applied to describe their properties. Within the scope of probability theory, standard methods, such as the statistical parameters (e.g., the mean value, standard deviation, skewness, kurtosis), correlation functions, power spectrum, statistical distributions, extreme value statistics, etc., can be applied to analyze the time series of the random load process (Ochi 1990). In this entry, previous studies on the statistical distributions of the measured ice loads, extreme value statistics, and fatigue damage evaluation based on the collected ice load peaks are summarized.

Probabilistic Models of Ice Loads

In general, the full-scale measurements can be classified as long term and short term. Long-term measurements are usually taken over several winter seasons, and they have been performed only in the Baltic Sea and the Antarctic sea (Kujala 1994). Basically, most of the full-scale measurements are based on the short-term measurements in which the time periods are hours and minutes. Ice load peaks collected by short-term measurements are widely used for probabilistic analysis.

Kheisin and Popov (1973) were the pioneers in introducing the probability theory to study the ice loads on ship hull. In their study, it was found that the number of ice load events in the ship bow region is distributed according to the Poisson law and the measured ice load peaks were found to follow an exponential distribution. Subsequently, there have been plenty of studies on probabilistic models of the measured ice loads on ship hull. Kujala and his colleagues suggested that the Weibull probability distribution or the exponential distribution or lognormal distribution would give the best fit to the ice loads data collected in the Baltic Sea (Kujala 1994; Suominen 2018). Figure 3 presents an example of measured ice load peaks fitted by different probabilistic models. In addition, Jordaan et al. (1993) applied the event-maximum method and found that the tails of ice loads collected in the North Chukchi sea can be fitted as an exponential distribution by using probability paper method.

However, from the full-scale measurements of ice loads collected onboard KV Svalbard in the sea area of Spitsbergen in the year 2007, Suyuthi et al. (2013b) mentioned that some sets of (short-term) ice load peaks cannot be well modeled by the traditional statistical models, such as the lognormal and the exponential distribution. Later, they proposed a generalized probabilistic model, i.e., a three-parameter exponential distribution to provide a better description of these collected



Probabilistic Aspects for Ice Loads on Ships, Fig. 3 Different probabilistic models for fitting the measured ice load peaks (Suominen 2018)

data sets (Suyuthi et al. 2014). As a result, the three-parameter exponential distribution can serve as an effective supplement to the traditional statistical models for describing the measured ice load peaks.

Extreme Value Statistics

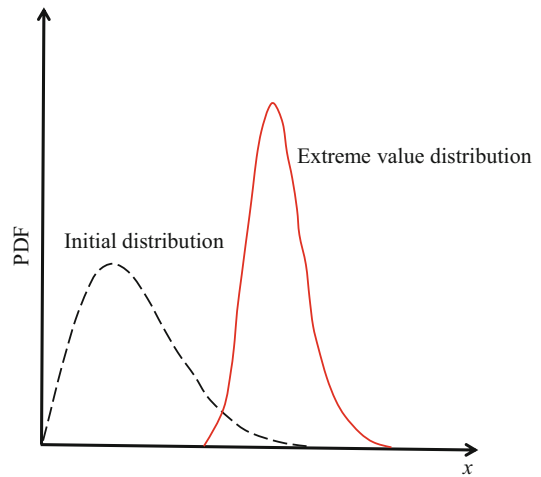
The extreme value statistics of ice loads are directly related to the reliability of the ice-capable vessels since the ultimate limit states (ULS) are generally based on extreme load effects. Assume that the stochastic process $X(t)$ represents the ice load peaks over a time interval $[0, T]$. The values $X_1, \dots, X_N$, which were derived from the observed process, are allocated to the discrete times $t_1, \dots, t_N$ in the time interval $[0, T]$. The extreme value among the N outcomes of the stochastic process $X(t)$ is defined as $M_N = \max \{X_1, \dots, X_N\}$, and the extreme value distribution for large values of η is expressed as:

$$\begin{aligned} P(\eta) &= \text{Prob}(M_N \leq \eta) \\ &= \text{Prob}(X_1 \leq \eta, \dots, X_N \leq \eta) \end{aligned} \quad (1)$$

Former studies of extreme loads prediction are mainly based on the classic extreme value theory, which includes the peak amplitude approach (Ralph and Jordaan 2013) and the asymptotic method (Lensu 2002). For the peak amplitude approach, it is assumed that all the N collected ice load peaks are independent and identically distributed with common distribution, $F_X(x)$, such as the Weibull, exponential, and lognormal distributions, described in section “Probabilistic Models of Ice Loads.” Then, the extreme value distribution is given in the form of the power of encountered number of ice loads events as Eq. 2. The relationship between the distribution of ice load peak $F_X(x)$ and the extreme value distribution $P(\eta)$ is illustrated in Fig. 4.

$$\begin{aligned} P(\eta) &= \text{Prob}(X_1 \leq \eta, \dots, X_N \leq \eta) \\ &= \prod_{i=1}^N \text{Prob}(X_i \leq \eta) = [F_X(\eta)]^N \end{aligned} \quad (2)$$

In the asymptotic method, a proper number of time windows with equal duration are



Probabilistic Aspects for Ice Loads on Ships, Fig. 4 Illustration of the relationship between the initial distribution (i.e., distribution of ice load peaks) and the extreme value distribution

selected to divide the measured time series into a corresponding number, K , of intervals. The maxima value in each interval is identified. Then, based on the collected maxima data, the type I asymptotic extreme value distribution, i.e., the Gumbel distribution, is applied to estimate the extreme distribution:

$$P(\eta) = G_Y(\eta) \quad (3)$$

where Y represents the process of maxima values in the K intervals and $G_Y(y)$ is the Gumbel distribution with the following expression:

$$G_Y(y) = \exp \left\{ -\exp \left[-\left(\frac{y - \beta}{\alpha} \right) \right] \right\} \quad (4)$$

where α and β are the parameters for the Gumbel distribution and they can be estimated by ordinary fitting of the empirical cumulative distribution, such as the least-square fitting in a probability paper and the method of moment, etc.

This method is also known as the Gumbel method. In this method, for small level of exceedance probability λ , the corresponding extreme value is given as:

$$\eta = G_Y^{-1}(1 - \lambda/K) \quad (5)$$

In addition to the abovementioned two methods based on the parametric distribution functions, there is another method, named as the ACER (average conditional exceedance rate) method, which can be applied to estimate the extreme distribution of the collected ice load peaks. The ACER method is based on a sequence of nonparametric distribution function and estimates the extreme value distribution by constructing different orders of the ACER functions, which are available for both the stationary and non-stationary data sets. The principle and development of the ACER functions are described in Næss and Gaidai (2009).

In this method, with the time series of the measured ice load peaks, the extreme value can be estimated in the following manner:

$$P(\eta) \approx P_k(\eta) \approx \exp(-(N - k + 1) \cdot \hat{\varepsilon}_k(\eta)) \quad (6)$$

where k is the order of the ACER function and P_k represents the approximation of the extreme value distribution based on the k th order ACER function. $\hat{\varepsilon}_k(\cdot)$ is the empirical ACER function of order k , which can be obtained by applying the existed time series.

In order to predict the extreme value distribution in the tail region, an extrapolation scheme is applied. This extrapolation technique is based on the observations that for ships and marine structures being considered, the mean upcrossing rate and the ACER functions are in general highly regular in the tail region. The empirical ACER function is assumed to be in the form of:

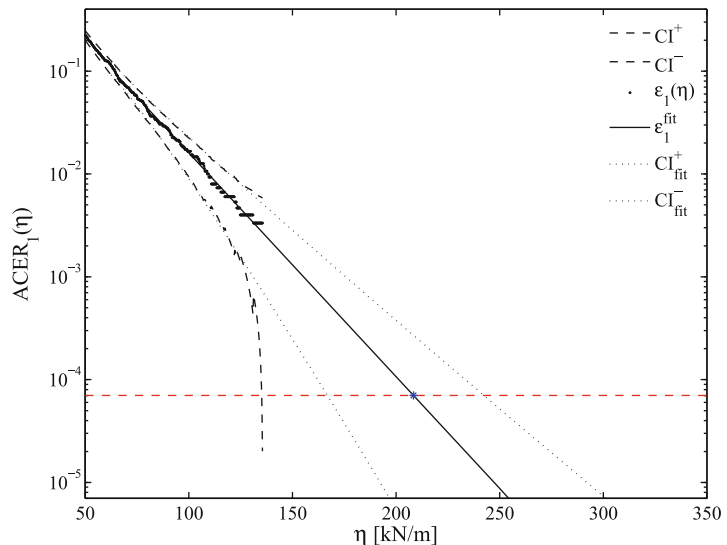
$$\hat{\varepsilon}_k(\eta) \approx q_k \exp\{-a_k(\eta - b_k)^{c_k}\}, \quad \eta \geq \eta_0 \quad (7)$$

where a_k , b_k , c_k , and q_k are suitable constants, which are dependent on the order k .

Generally, under the aforementioned assumptions for the tail region of the ACER functions, plotting $\log|\log(\hat{\varepsilon}_k(\eta)/q_k)|$ versus $\log(\eta - b_k)$ would give an almost perfect linear tail behavior. In addition to this method, a more robust Levenberg-Marquardt least-squares optimization method can be used to determine the optimal values for a_k , b_k , c_k , and q_k . Details of this method can refer to Naess and Moan (2012) and Chai et al. (2016). An example of extreme value estimation for the collected ice load peaks is presented in Fig. 5. Moreover, the performance of the abovementioned three methods applied for estimation of the extreme values of ice load peaks has been systematically reported in Chai et al. (2018a).

Probabilistic Aspects for Ice Loads on Ships,

Fig. 5 An example of extreme estimation for the collected ice load peaks based on the ACER₁ function



Fatigue Damage Evaluation

In addition to the extreme ice loads statistics, fatigue damage caused by cycle ice loading is also an important issue for Arctic ships navigating in severe ice regions. Generally, fatigue damage evaluation is based on the S-N data in association with the Palmgren-Miner law of damage accumulation. For the ice load-induced fatigue damage, when the time series of ice-induced stress ranges are available, fatigue damage of ship hull can be directly estimated by summing the fatigue damages of all stress cycles according to the linear Palmgren-Miner cumulative rule or estimated by probabilistic manners when the distribution of stress ranges is known. Current studies for fatigue damage of ship hull due to ice loads actions are limited and only for short-term collected ice load peaks and the associated ice-induced stress ranges (Chai et al. 2018b; Suyuthi et al. 2013a). Assessment of fatigue damage for long-term measurements remains undeveloped.

Future Work

The abovementioned former studies are focus on the short-term measured ice load peaks. The methods mentioned in this entry can also be extended for long-term studies when relevant long-term measured data is available. Such studies for extreme value analysis of the ice loads and fatigue damage evaluation are directly related to the ultimate limit states and fatigue limit states of the Arctic ship during its service life.

In addition, ice thickness is assumed to be the most representative parameter for the severity of the ice conditions. The ice thickness during the full-scale measurement voyage can be regarded as a random process with respect to time space. Due to the stochastic nature of ice loads and ice thickness, it is a challenge to establish a link between the prevailing ice conditions and ice-induced loads. This study would be achieved when the relevant long-term measured data is available.

Furthermore, for current rule-based design method for Arctic ships, the reliability-based design method with considerations of the

randomness and uncertainties of the ice condition and ice loads could be an effective supplement.

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Probability of Reliability

► [Reliability and Safety in Offshore Engineering](#)

Probability Theory

► [Reliability-Based Design \(RBD\)](#)

Profiling Float

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Synonyms

[Autonomous Lagrangian circulation explorer \(ALACE\)](#); [Deep-ocean profiling float \(DOPF\)](#); [Neutral float \(NF\)](#); [Regular profiling float \(RPF\)](#); [Self-sustaining profiling automation circulation detector \(SPACD\)](#)

Definition

Profiling float is also called “Autonomous Lagrangian circulation explorer (ALCE)” or “Self-sustaining profiling automation circulation detector (SPACD).” It is a measurement instrument which can freely drift in the ocean, and it uses the Lagrangian circulation method to automatically measure seawater temperature, electrical conductivity (salinity), and pressure from the sea surface to a certain depth and tracks the drift trajectory to obtain the velocity and direction of the current (Xu 2002).

Scientific Fundamentals

Working Principle

There are usually three ways in which objects can ascent and descent in water. One is to change the volume of the object without changing the weight; if the volume changed, the buoyancy will be changed; the other is to change the weight while keeping the volume constant, and then the net buoyancy is changed; the third is to increase or decrease the applied external force (Xu 2002; Yu et al. 2001).

Early neutral floats changed the net buoyancy by changing their weight to achieve the motion ups and downs, so the volume were relatively large and cumbersome, and repeated measurements could not be performed. After a period of development, a method of changing the volume was used in the design of neutrally buoyant float. The neutrally buoyant floats periodically change their buoyancy by pumping the hydraulic fluid from an internal reservoir to an external bladder, thereby increasing float volume and buoyancy, so repeated measurements can be achieved, and the volume of the float was reduced a lot at the same time.

History of Profiling Float

The prototype of profiling float is a measurement instrument made by using Euler and Lagrange method to detect the sea surface or deep currents. Profiling floats were developed on the basis of neutral buoyant floats and integrated sensor technology, satellite positioning technology, and communication technology.

In the summer of 1955, John Swallow made his first visit to the National Institute of Oceanography to discuss with oceanographers the possibility of making direct measurements of the vertical profile of currents in the deep ocean (Gould 2005). Swallow is the pioneer to develop the neutrally buoyant float.

The first floats were constructed at the beginning of 1955 in the National Institute of Oceanography (NIO). At that time, materials and money were in short supply; the standard tubing had too great a wall thickness, so they put the tubes in a bath of caustic soda to thin the wall. Only

6 months after construction started, the first floats were deployed in June 1955.

The floats weighed around 10 kg in air but had to be weighed in water in order that they could be ballasted to stabilize at their target depth. This was done by suspending each float from a simple chemical balance mounted above a tube of salt solution (close to a salinity of 35 and mixed by repeatedly lowering a bucket down the tube and hauling it up again to prevent stratification) in the stairwell of the NIO. The density of the saline solution was measured using high school physics techniques (specific gravity bottles). The floats needed only 38 g of negative buoyancy to stabilize at 1000 m, so great care was needed with the weighing and density calculations and to eliminate trapped air bubbles. Floats were used by NIO scientists in Labrador (See, Fig. 1).

Though only two worked satisfactorily in the six floats, the method had been demonstrated, and the results were detailed enough to show evidence of tidal variations (Swallow 1955).

However, the concept of neutrally buoyant float had also been developed simultaneously and independently on the other side of the Atlantic. Stommel had also put forward to directly measure the deep currents by using the subsurface neutrally buoyant floats (Stommel 1955). He supposed that deep currents should be tracked through the Sound Fixing and Ranging (SOFAR) channel by the floats creating regular explosions.

However, the discovery of an energetic ocean mesoscale exposed the limitations of ship-tracked floats. If the mean ocean circulation were to be revealed, so continuous observations over months would be needed.

The problem was addressed by Rossby and Webb (1970); the floats could be tracked by sound transmissions through the SOFAR channel. Then hydrophones could be used to monitor the floats. The first two SOFAR floats (Fig. 2) were deployed in the Sargasso Sea in 1968 and showed that reception of signals was possible at ranges



Profiling Float, Fig. 1 John Swallow (left) and Gordom Volkmann on R. V. Erika Dan in 1962. The float shown here is essentially identical to floats deployed between 1957 and 1970 and using 10 kHz magnetostrictive nickel scroll transducers (Gould 2005)



Profiling Float, Fig. 2 Prototype SOFAR float (Gould 2005)

of up to 1000 km and that float positions could be determined with an accuracy of the order of 3–5 km and with a life time of 9–12 months. In 1969, a float operating at 380 Hz was tracked for 4 months and confirmed the robustness of the technique. Each float weighed around 430 kg and was over 5 m long. The floats also had a 10 kHz, short-range navigation system to allow their location and subsequent recovery by a ship (Rossby and Webb 1971).

The restriction of SOFAR floats at that time to depths near the sound channel axis (due to depth restrictions on the pressure cases) meant that it was not possible to use these floats to explore the vertical structure of currents over most of the water column (Swallow 1977). Thereby, a ship-based system using transponding floats made it possible by improved transducers, and microelectronics was developed by John Swallow and his coworkers (Swallow et al. 1974). The system allowed up to 18 floats to be tracked simultaneously. This ability to interrogate from a wide range of depths allowed tracking of floats at all depths and the achievement of ranges of up to 70 km. The SOFAR floats enabled day-to-day objective mapping of the ocean meso-scale over an area 400 km² and revealed the long-term propagation of these features (Freeland et al. 1975; Freeland and Gould 1976).

From the mid-1970s, acoustically tracked floats were used extensively to further explore the ocean's mesoscale structure and variability.

In the mid-1980s, there was a considerable interest in the potential for the disposal of radioactive waste either below or on the sea bed. Acoustically tracked floats were ideal for investigating the ocean circulation around potential disposal sites (Gould 2005).

With the increasing of the investigation depth, the pressure cases of the aluminum tube could not meet the needs. Then floats using glass spheres as pressure cases were developed both in France and in the USA. The floats designed by Webb Research Corporation (WRC) using four glass spheres and transmitting at 260 Hz could reach the depths close to 3000 m with a lifetime of about 4 years (Rees and Gmitrowicz 1989) (Fig. 3).



Profiling Float, Fig. 3 Deep SOFAR float using glass spheres. The uppermost sphere provides buoyancy, the middle two hold batteries, and bottom sphere contains electronics. The float is seen in a deployment cradle that was opened hydraulically when the float was below the sea surface (Gould 2005)

Due to the SOFAR tracking system needed to place the bulky and heavy sound sources on moorings, thus the usage of long-range floats was restricted. For this reason, the smaller and cheaper RAFOS floats were developed by Tom Rossby and his group (Rossby et al. 1986, 1993) to record the signal arrivals from an array of moored sources and transmit the data back to satellites when they surface at the end of their mission.

The applications of various types of floats during the 1970s, 1980s, and 1990s on the scientific investigation were extensive and significantly improved our understanding of the oceanic eddy fields (Webb et al. 1970; Gascard 1973; Rossby et al. 1994; D'Asaro et al. 1996).

Due to the measurement could not be done repeatedly by means of dropping a ballast weight,

the Autonomous Lagrangian circulation explorer (ALACE) was developed (Davis et al. 1992). It is a novel system which can be tracked globally. ALACE was subsurface float that cycles vertically from a depth where it was neutrally buoyant to the surface where it was located by, and relays data to, System Argos satellites (Davis et al. 1992). This type of float periodically changes their buoyancy by pumping hydraulic oil from an internal reservoir to an external bladder, thereby increasing float volume and buoyancy.

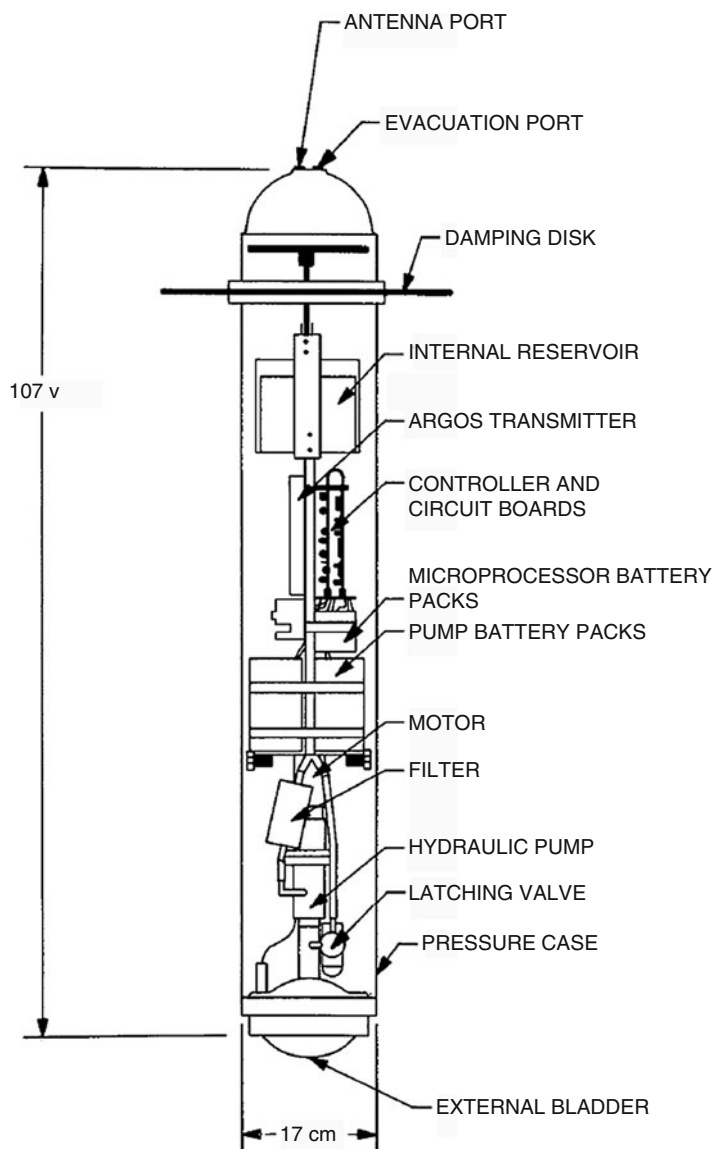
Because of positioning and data relay are accomplished by satellites, ALACEs are autonomous of acoustic tracking networks and are suitable for global deployment. Schematic of ALACE is shown in Fig. 4.

After that, the oil bladder regulation device was widely used in the design of profiling float.

As the small, low-power, autonomous CTD sensors with good long-term stability were developed by Falmouth Scientific Instruments and Sea-Bird Electronics, the full temperature and

Profiling Float,

Fig. 4 Schematic of an ALACE. To ascend, the hydraulic pump moves oil from internal reservoir to external bladder. To descend, the latching valve is opened allowing oil to flow back into internal reservoir. Antenna shown to right is mounted on the top hemispherical endcap (Davis et al. 1992)



conductivity profiles were developed, and ALACE floats evolved to P-ALACE floats (Davis et al. 2001).

The research of profiling floats began at the end of the Ninth Five-Year Plan (1996–2000) in China. COPEX profiling float was successfully developed by the National Ocean Technology Center of China in 2003. Another type of profiling float HM2000 was developed by 710 Research Institute of China Shipbuilding Industry Corporation (CSIC) in 2015.

Although the development of profiling floats started relatively late in China, the Beidou satellite navigation and positioning system developed in China has the ability to determine the geographical location of users. In and around China, two-way brief digital message communication can be realized between the users, the user and the central control system in the Beidou user machine system. The user terminal has two-way digital message communication capabilities and has been successfully applied in profiling floats (Zhang et al. 2009; Wu et al. 2013).

Due to the observation-based studies that demonstrated the warming of the deep ocean below 2000 m has significantly contributed to the mean sea level rise (Purkey and Johnson 2010; von Schuckmann et al. 2014), a strong requirement was proposed from the ocean and climate research community to expand the Argo coverage into the deep ocean below 2000 m. This was an essential element of the global ocean observing system envisioned by the OceanObs'09 conference (Garzoli et al. 2010; Church et al. 2011). Then four types of deep-ocean profiling float were developed, Deep Arvor, Deep NINJA, Deep APEX, and Deep SOLO. Deep Arvor and Deep NINJA can reach the depth of 4000 m; Deep APEX and Deep SOLO can reach a greater depth of 6000 m (Zilberman and Maze 2015; Reste et al. 2016). The deep-ocean profiling floats were shown in Fig. 5.

Technical Challenge

From the development process of profiling float, we can see that the industrial foundation largely determined the performance. The buoyancy

regulation devices evolved from changing weight to changing volume; pressure hull evolved from aluminum housings to alloy hulls to glass spheres, etc., which can stand greater water pressure and explore deeper. However, to design a profiling float with superior overall performance is a systematic engineering.

1. Energy Efficiency

The buoyancy regulation system is the basis for achieving ups and downs of the profiling floats, and it determines the performance of the float largely. The power consumption of the motor accounts for about 60% of the total power consumption of the measurement process (Reste et al. 2016). The energy efficiency is not only related to the motor, hydraulic pump, but also related to the control algorithm, the transmission speed of the communication and positioning system, and other factors.

2. Diving Depth

The increased diving depth will inevitably lead to an increased difficulty of design. An increased design depth results in lower pump efficiency, higher motor power consumption, and higher requirements to the pressure material. Weight is also an important indicator for evaluating the performance of profiling floats. Lightweight profiling floats facilitate handling and deployment. The weight control is also a systematic project that involves every design element and component of the floats. Therefore, the pursuit of better performance will inevitably increase the cost.

3. System Reliability

There are many causes of failure of the profiling floats, such as failure of the communication and positioning function, sensor failure, battery leakage, hydraulic oil leakage, failure of the pressure hull under special conditions, and so on.

Reliable satellite transmission is critical to achieving the scientific goal of measuring subsurface currents. Surface drift must be accurately measured so that measurements of the much slower subsurface motions are not contaminated. Positions are not obtained exactly at



Deep SOLO float courtesy of Scripps
Institution of Oceanography (6000m)



Deep APEX float courtesy of
Teledyne Webb (6000m)



Deep NINJA float courtesy
of JAMSTEC (4000m)

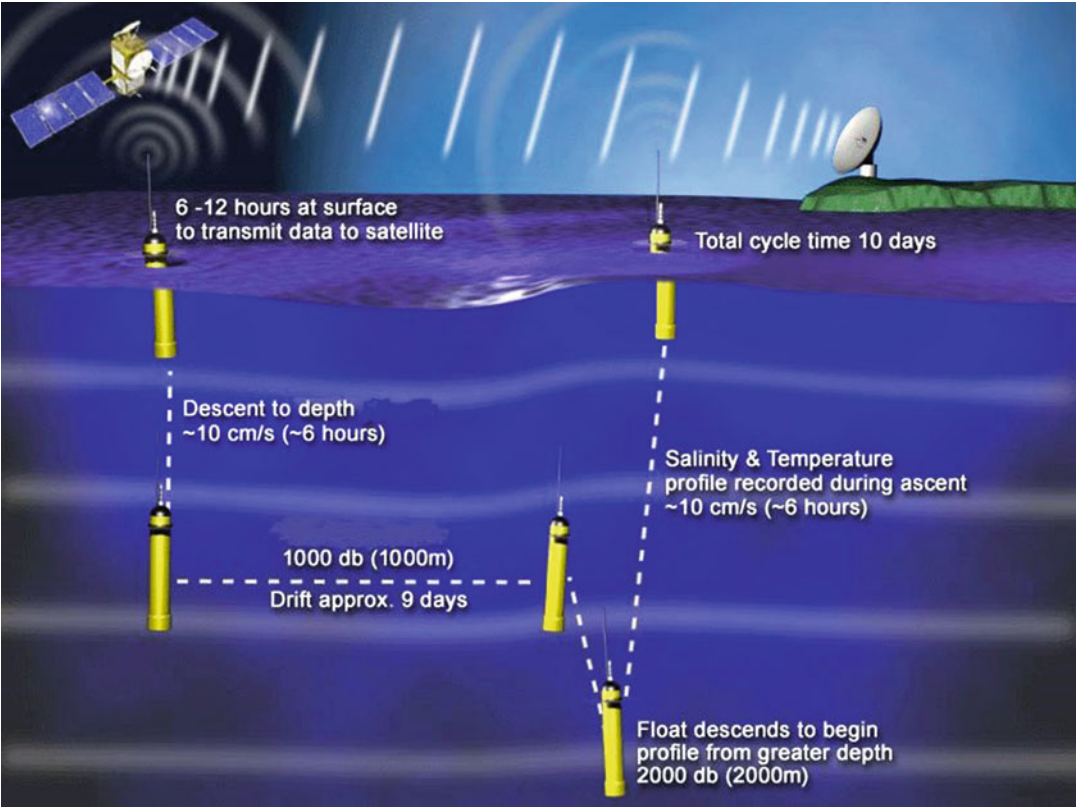


Deep Arvor float courtesy
of NKE (4000m)

Profiling Float, Fig. 5 Deep-ocean profiling floats (<http://www.argo.ucsd.edu/pictures.html>)

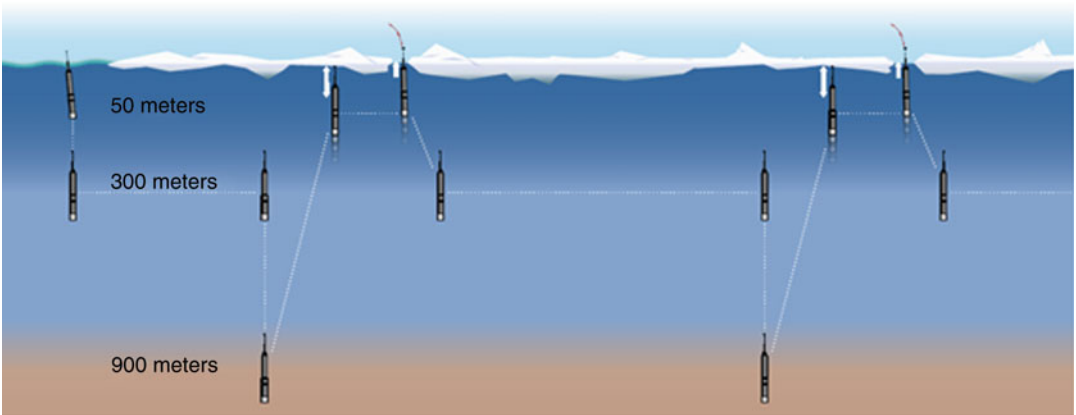
the surfacing and descent times; therefore, it is necessary to extrapolate the observed positions to these times. Because this requires obtaining as many positions as possible, an important design objective for profiling float is to obtain

good surface following so that the antenna is rarely submerged.
Another difficult problem is to avoid the loss of prime in the high-pressure pump caused by air bubbles in the hydraulic fluid.



Profiling Float, Fig. 6 Park & Profile Mission Operation (http://www.argo.ucsd.edu/How_Argo_floats.html)

P



Profiling Float, Fig. 7 Measurement process of float equipped with an ice probe sensor (<http://www.argo.org.cn/index.php?c=index&catid=17&contentid=453&f=show&m=content>)

The system reliability is closely related to the design and development process. Therefore, the reliability runs through the entire process of the floats.

Key Applications in the Argo Plan

Argo is a global array of 3800 free-drifting profiling floats that measures the temperature and salinity of the upper 2000 m of the ocean. This allows, for the first time, continuous monitoring of the temperature, salinity, and velocity of the upper ocean, with all data being relayed and made publicly available within hours after collection. The measurement process of regular Argo float and high-latitude ice exploration float was shown in Figs. 6 and 7, respectively.

The origins of Argo can be found in the 1990–1997 World Ocean Circulation Experiment (WOCE). WOCE is part of the World Climate Research Programme (WCRP) and set out to collect an unprecedented set of observations. WOCE needed to collect data on ocean currents at about 1000 m throughout the oceans.

Cross-References

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Glider](#)

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Project Execution Plan

► [Field Development](#)

Propeller Open Water Tests

► [Towing Tank Test](#)

Protocol Design

► [Underwater Acoustic Sensor Network](#)

PSF – Partial Safety Factors

► [Design of Renewable Energy Devices](#)

PTD, Power Transmission and Distribution

► [Power Transmission and Distribution](#)

PTM – Passive Towed Mining

► [Underwater Mining System](#)

PTO – Power Take-Off

► [Power Take-Off System](#)

Pure Loss of Stability

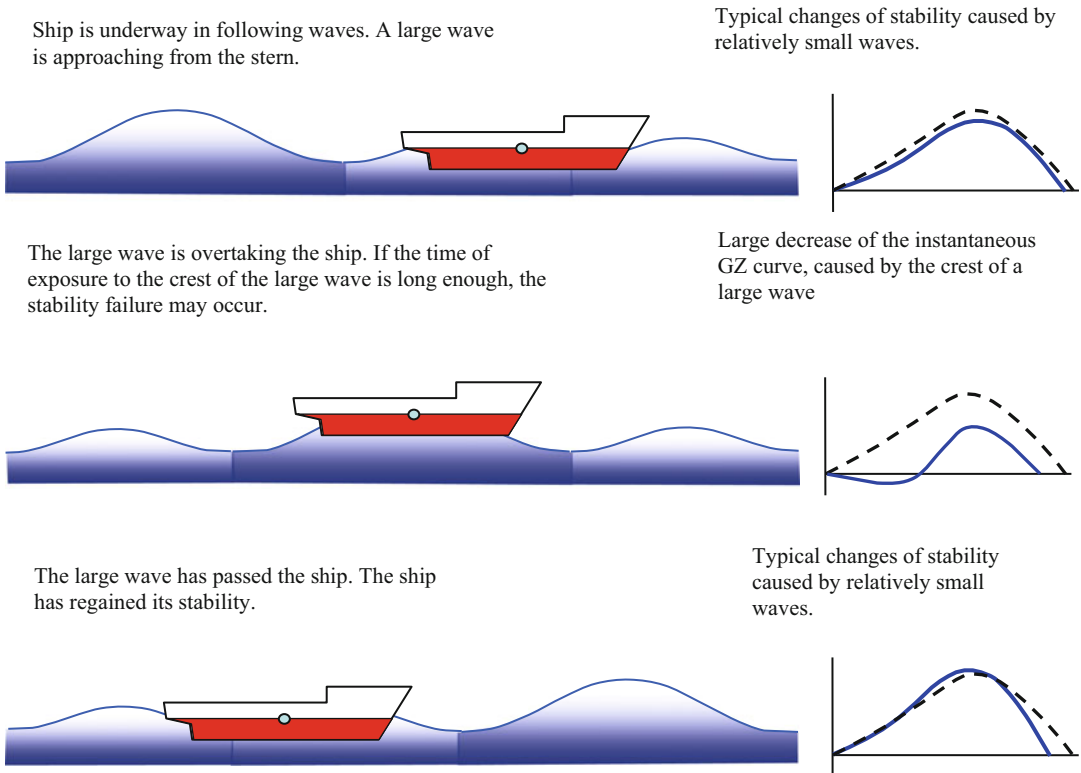
Min Gu, Jiang Lu, Shuxia Bu, Jilong Chu,
Ke Zeng and Tianhua Wang
China Ship Scientific Research Center (CSSRC),
Wuxi, China

Synonyms

[Loss of stability](#); [Ship loses restoring in waves](#)

Definition

Pure loss of stability was firstly named by Paulling as a kind of capsizing mode (1961) and afterward was under research and realization. Germany proposed pure loss of stability to be part of the stability criteria in the development of IMO in the second generation intact stability criteria. The case diagrammed in Fig. 1 shows a large wave approaching the ship from the stern, while the ship is underway with relatively high speed in following seas. If the celerity (speed) of the large wave is just slightly above the ship speed, the duration needed for the large wave to pass or overtake the ship may be long (“long” here means at least an order of magnitude greater than the natural roll period). Once the crest of the large wave is near the midship section of the ship, the righting lever may be significantly decreased. Further, because of the significant duration that this condition may exist, a large heel angle may develop, which could lead to capsize.



Pure Loss of Stability, Fig. 1 A possible scenario for the development of pure loss of stability (Belenky et al., 2011; SDC 3/WP.5/Annex 3, 2016)

Historical Development

The stability in waves often becomes larger at the trough and becomes smaller at the crest comparing with that in calm water (Paulling, 1961). Pure loss of stability was identified during the model experiments in San-Francisco Bay (Paulling et al., 1972, 1975), and considered as a static capsizing mode that ship loses static restoring in waves (Oakley et al., 1974). Many researchers focused on the method to calculate the GZ curve in regular and irregular waves (Kuo et al., 1986; Hamamoto & Nomoto, 1982). One method for calculating the restoring variation in waves was developed by using the strip method of heave and pitch motion instead of the static method (Umeda et al., 2005).

The restoring moment in irregular waves is considered a stochastic process (Dunwoody, 1989; Palmquist, 1994; Bulian & Francescutto, 2006). The Grim effective wave is used to

evaluate the probability of capsizing due to pure loss of stability in short-crested irregular stern-quartering waves (Umeda & Yamakoshi, 1993). The waves are assumed to be a narrow-banded stochastic process instead of the Grim effective wave (Vermeer, 1990). The concept of a “critical wave” is used to evaluate the probability of capsizing (Themelis & Spyrou, 2007). The effective wave approach combined with the up-crossing theory is used to produce a time-dependent probability of stability failure (Bulian et al., 2008). The application of the split-time method to pure loss of stability is discussed by Belenky et al. (2015).

Pure loss of stability is a nonlinear phenomenon involving a large amplitude roll motion, and Spyrou (Spyrou, 1997) suggested that the loss of stability could be related to periodic motion, accumulative broaching, and so on and it is still difficult to be predicted quantitatively. Hashimoto carried out experiments on pure loss of stability

in following seas with an initial heel moment induced by cargo shift (Hashimoto, 2009). Jiang Lu further confirmed capsizing due to pure loss of stability may not be reproduced in the simulation and experiment in following seas without initial heel moments (Lu et al., 2019).

Umeda firstly pointed out that pure loss of stability in stern quartering waves could not be really pure, and also the heeling moments induced by a centrifugal force due to ship maneuvering motions are the relevant external moments (Kubo et al., 2012). Umeda further argued the capsizing of the actual ship in stern quartering waves could be due to ship maneuvering motions of sway and yaw (Umeda et al., 2019). Jiang Lu further confirmed that the coupling force from the maneuvering sway and roll motions to roll motion is a key factor for pure loss of stability in stern quartering waves (Lu et al., 2020).

Key Numerical Method for Pure Loss of Stability

The 4-DOF mathematical model is expressed by the surge, heave, roll, and pitch for pure loss of stability in following waves in the reference (Lu et al., 2019). The 6-DOF mathematical model is expressed by the surge, sway, heave, roll, pitch, and yaw motions for pure loss of stability in stern quartering waves in the reference (Lu et al., 2020). The surge, sway, yaw, and roll motions refer to the well-established models (Yasukawa & Yoshimura, 2015; Umeda et al., 2016) as shown in Eqs. (1)–(4), respectively. The heave and pitch motions are expressed in Eq. (5) and (6), respectively. The heave and pitch motions are calculated at each constant forward speed applied to an upright hull by a strip theory method using an enhanced integrating method of direct line integral to solve the velocity potential (Kashiwagi et al., 2010). The time-domain heave and pitch motions are calculated according to the relative position of the ship to waves as shown in Eqs. (7) and (8), respectively. The control equation for course-keeping by steering is added in the 6-DOF mathematical model as shown in Eq. (9).

$$(m + m_x)\dot{u} - (m + m_y)vr = X_H + X_R + X_P + X_W \quad (1)$$

$$(m + m_y)\dot{v} + (m + m_x)ur = Y_H + Y_R + Y_W \quad (2)$$

$$(I_{zz} + J_{zz})\dot{r} = N_H + N_R + N_W \quad (3)$$

$$\begin{aligned} (I_{xx} + J_{xx})\dot{p} - m_x z_H ur - m_y z_H \dot{v} \\ = K_H + K_R + K_W - D_{lf} \\ - D(\dot{\varphi}) - W \cdot GZ_{W_FK}(\xi_G/\lambda, \zeta_G(t), \theta(t), \chi, \varphi) \end{aligned} \quad (4)$$

$$\begin{aligned} (m + A_{33}(u))\ddot{\zeta} + B_{33}(u)\dot{\zeta} + C_{33}\dot{\zeta} + A_{35}(u)\ddot{\theta} \\ + B_{35}(u)\dot{\theta} + C_{35}\theta = F_3^{FK}(u) + F_3^{DF}(u) \end{aligned} \quad (5)$$

$$\begin{aligned} (I_{yy} + A_{55}(u))\ddot{\theta} + B_{55}(u)\dot{\theta} + C_{55}\theta + A_{53}(u)\ddot{\zeta} \\ + B_{53}(u)\dot{\zeta} + B_{53}\zeta = F_5^{FK}(u) + F_5^{DF}(u) \end{aligned} \quad (6)$$

$$\zeta_G(t) = \zeta_{Ga}(u) \cos [2\pi \cdot (\xi_G/\lambda) - \delta_H(u)] \quad (7)$$

$$\theta(t) = \theta_a(u) \cos [2\pi \cdot (\xi_G/\lambda) - \delta_\theta(u)] \quad (8)$$

$$\dot{\delta} = \{-\delta - K_P(\chi - \chi_C) - K_P T_{Dr}\}/T_E \quad (9)$$

The subscript H, R, P, and W refer to hull, rudder, propeller, and wave, respectively. Eqs. (7) and (8) are seakeeping mathematical models, and their symbols are listed with the traditional seakeeping method as follows.

$\zeta_G(t)$: heave displacement; $\theta(t)$: pitch angle; F_3^{FK} , F_3^{DF} : wave exciting force on heave direction including Froude-Krylov component and diffraction component; F_5^{FK} , F_5^{DF} : wave exciting moment on pitch direction including Froude-Krylov component and diffraction component; $\zeta_{Ga}(u)$, $\delta_H(u)$: amplitude and initial phase of heaving when the ship's forward speed is u ; and $\theta_a(u)$, $\delta_\theta(u)$: amplitude and initial phase of pitching when the ship's forward speed is u . The dot denotes the differentiation with time. A_{ij} , B_{ij} , and C_{ij} are coupling coefficients for added mass, damping, and restoring coefficients, respectively.

Subscripts 3 and 5 denote heave and pitch directions, respectively.

Experimental Method for Pure Loss of Stability

The ship model was driven by propellers in stern quartering seas in the free-running experiment. The roll, pitch, and yaw angles were measured by an optical fiber gyroscope placed on the ship model and the roll, pitch, yaw, and rudder angles and the propeller rotation speed was recorded by an onboard system which is connected with an onshore control computer wirelessly. The wave elevation was measured at the middle position of the basin by a servo-needle wave height sensor attached to a steel bridge which is 78 m in length and spans over the basin.

Free roll decay tests in calm water were conducted to obtain roll damping coefficients. The speed is a key factor for pure loss of stability. Here the nominal Froude number (Fn) is used for the experiment of pure loss of stability in stern quartering waves by using the same specified propeller rate as in calm water. The specified propeller rate corresponding to one nominal speed in calm water is determined by measuring the instantaneous position of the model ship with a total station system.

First, the model is kept near the wave maker manually by two workmen sitting on the carriage. The initial heading of the model is kept referring to the steel bridge which can rotate about its center, up to 45 degrees. Next, the wave-making system starts to generate waves. Then, the propeller revolutions increase up to the specified value according to the order received from the onshore control computer. When the wave train propagates far enough, the model is released free near one wave crest with its initial heading, and then the model automatically runs in stern quartering waves with its specified propeller rate and autopilot course by a course keeping system. The course keeping system includes the 6-DOF optical fiber gyroscope installed on the ship model and a PD control system is used for course keeping. The latter reacts according to the bias between the yaw angle from

Pure Loss of Stability, Table 1 Principal particulars of the ONR tumblehome

Items	Ship	Model
Length:L	154.0 m	3.800 m
Breadth:B	18.8 m	0.463 m
Depth:D	14.5 m	0.358 m
Draft:d	5.494 m	0.136 m
Displ.:W	8507 ton	127.8 kg
C_B	0.535	0.535
GM	1.48 m	0.037 m
OG	-2.729 m	-0.067 m
L_{CB}	-2.569 m	-0.063 m
T_ϕ	14.0 s	2.199 s
κ_{yy}	0.25 L	0.25 L
K_{zz}	0.25 L	0.25 L
$2 \times A_R$	$2 \times 23.74 \text{ m}^2$	$2 \times 0.0145 \text{ m}^2$
D_P	5.22 m	0.129 m
$\delta_{\max}$	35deg	35deg

wave direction measured by the gyroscope and the yaw angle of autopilot course and the yaw velocity measured by the gyroscope. When the ship model is free running in calm water, a disturbance of yaw angle is made, and then the rudder gain is set by the experience according to the reaction of course keeping in calm water.

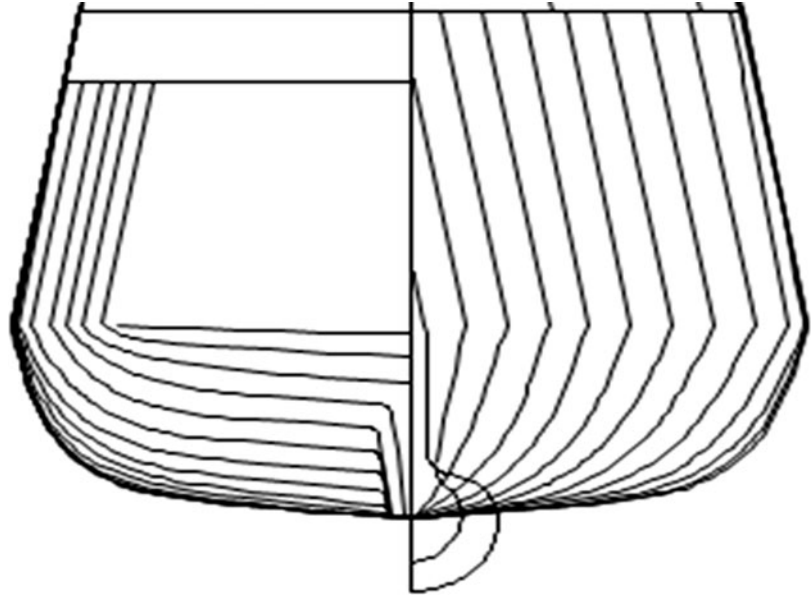
The principal particulars and body plan of the ONR tumblehome are shown in Table 1 and Fig. 2, respectively. The ship model in the free running experiment is shown Fig. 3.

The Effect of Wave on Roll Restoring Variation

When the midship section is located on the crest in following seas, the metacentric height is reduced and may be even negative.

The righting arm in calm water GZ , the restoring variations in waves with the static balance method GZ_W -static and with the strip method at different constant forward speeds GZ_W (for $Fn = 0.0, 0.1$ and 0.3) are shown in Fig. 4. The stability loss at the wave crest is significant, and if the state of stability loss at the crest exists long enough, the ship may capsize.

Pure Loss of Stability,
Fig. 2 The ONR
tumblehome lines



Pure Loss of Stability, Fig. 3 The ship model in the free-running experiment

The Effect of Constant Speed on Pure Loss of Stability

The righting arm in calm water GZ and the restoring variations in waves GZ_w at different constant forward speeds are shown in Fig. 5. The encounter period becomes larger as the ship increases its forward speed, and the state of stability loss at the wave crest becomes larger.

The roll angle due to pure loss of stability at different constant forward speeds is shown in Fig. 6, and the roll angle becomes larger as the ship forward speed increases because the state of

stability loss at the crest becomes larger as shown in Fig. 5. But the state of stability loss at the crest is not long enough to result in capsizing.

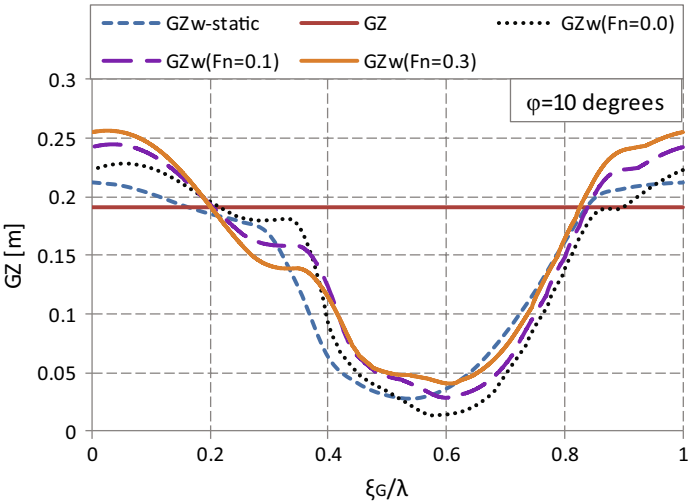
The Effect of Surge Motion on Pure Loss of Stability

The nominal velocity of the ship at $F_n = 0.3$, the actual velocity of the ship and the wave velocity are shown in Fig. 7. The nominal velocity of the ship is much smaller than the wave velocity, and the maximum actual velocity of the ship is also smaller than the wave velocity. The forward speed is varied in a large range around the nominal speed of the ship due to the surge motion, and the state at the crest exists longer than that at the trough.

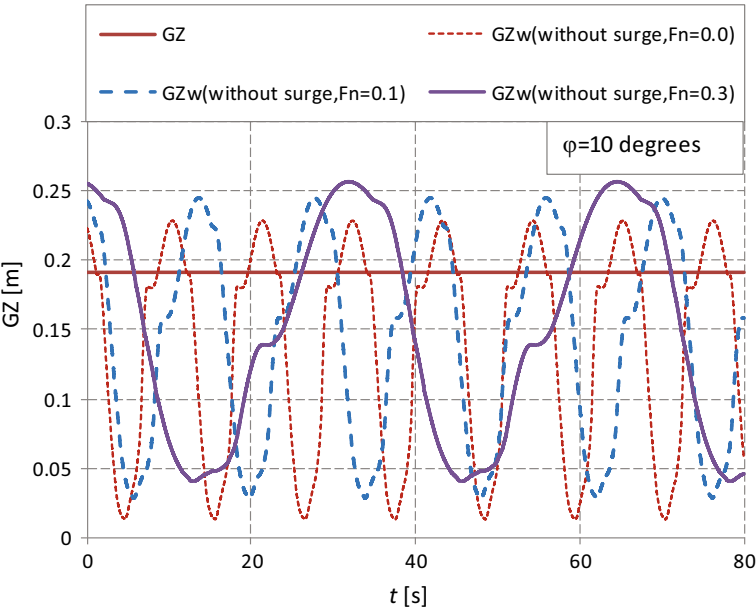
The righting arm in calm water GZ and the restoring variations in waves GZ_w with surge and without surge are shown in Fig. 8. The state of stability loss at the crest exists longer than that at the trough because the surge motion causes the state at the crest to exist longer than that at the trough as shown in Figs. 8 and 9.

As shown in Fig. 10, the mathematical model with 3-DOF of heave-roll-pitch coupled motions

Pure Loss of Stability,
Fig. 4 Restoring variation
in following seas with
 $\varphi = 10$ degrees, $\lambda/L_{pp} = 1.25$, $H/\lambda = 0.05$,
and $\chi = 0$ degree



Pure Loss of Stability,
Fig. 5 Time-domain
restoring variation in
following seas with
 $\varphi = 10$ degrees, $\lambda/L_{pp} = 1.25$, $H/\lambda = 0.05$,
and $\chi = 0$ degree



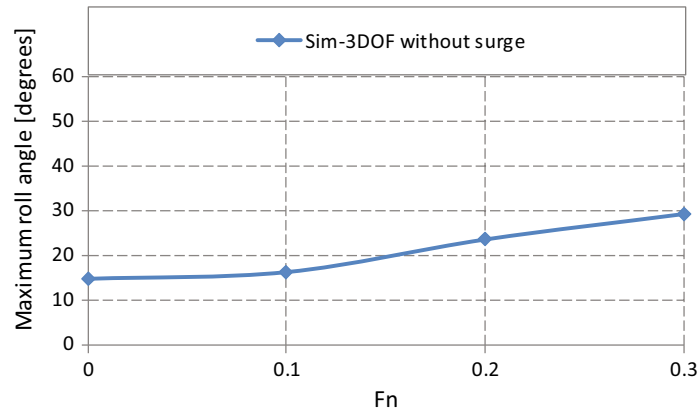
fails to predict capsizing because the state of stability loss at the crest is not long enough while that with 4-DOF of surge-heave-roll-pitch coupled motions could appropriately estimate the pure loss of stability in following seas. One key reason is that the state at the crest exists longer than that at the trough due to the surge motion and then the state of stability loss at the crest exists long enough. Therefore, the surge motion is important for predicting pure loss of stability in following seas.

The Effect of Initial Heel Angle on Pure Loss of Stability

Without an external heeling moment, once the wave crest passes the ship, the ship will finally return to the upright position with regained stability as shown in Fig. 10 with $\varphi = 0$ degree. The initial heeling angle is set as 8.6 degrees by cargo shift in the experiment and capsizing happens due to pure loss of stability as shown in Fig. 10. For investigating the effect of the initial heeling angle

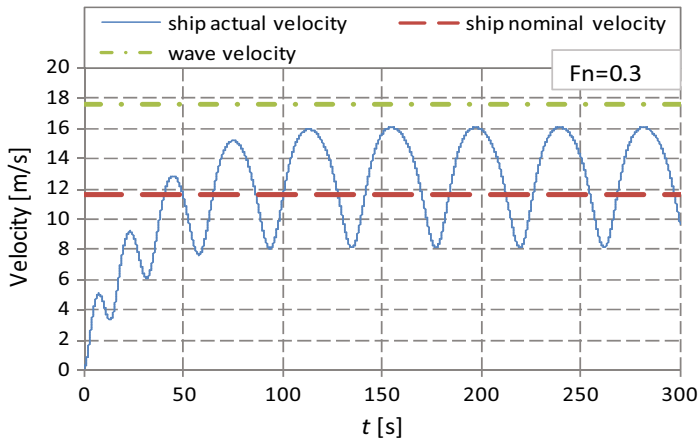
Pure Loss of Stability,

Fig. 6 The effect of the constant speed on pure loss of stability with an initial heeling $\varphi = 8.6$ degrees, $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 0$ degree



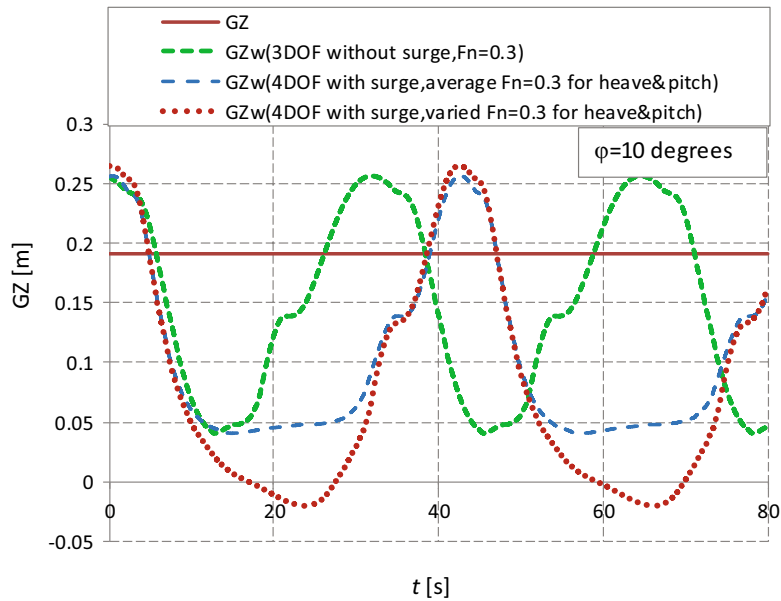
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Fig. 7 Comparison between ship velocity and wave velocity with nominal $F_n = 0.3$, $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 0$ degree



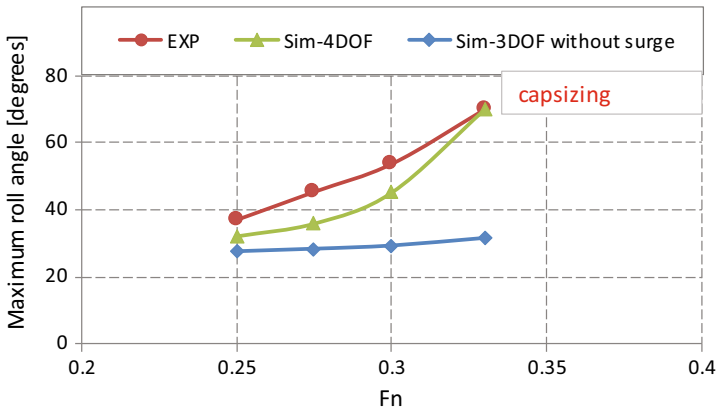
Pure Loss of Stability,

Fig. 8 The effect of surge motion on restoring variation with $\varphi = 10$ degree, $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 0$ degree



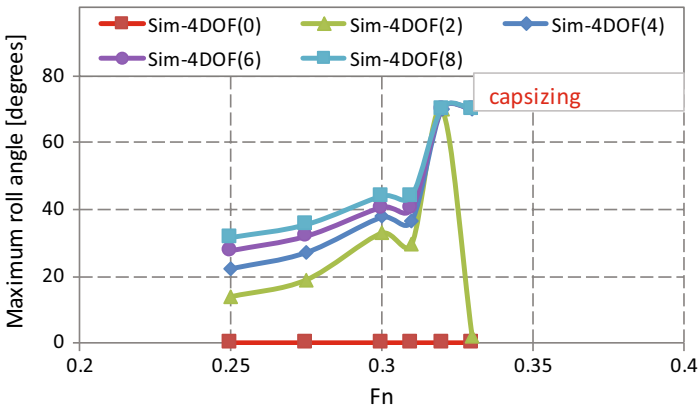
Pure Loss of Stability,

Fig. 9 The effect of the sure motion on pure loss of stability with an initial heeling $\varphi = 8.6$ degrees, $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 0$ degree



Pure Loss of Stability,

Fig. 10 The effect of initial heeling angles on pure loss of stability with $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 0$ degree ($\varphi = 0, 2, 4, 6, 8$ degrees)



on pure loss of stability, simulations with different initial heeling angles are carried out as shown in Fig. 10. The roll angles become larger as the initial heeling angles increases, and capsizing happens at the critical speeds due to pure loss of stability. However, the ship could be captured by a wave crest when the ship has a very small initial heeling angle in following seas at a high speed. That is to say, the ship reaches the speed of the wave in this case.

manoeuvring force in the sway direction underestimates roll angles and fails to correctly predict capsizing range of critical ship speeds as shown in Fig. 11. This is due to its negative contributions to the roll moment as shown in Fig. 12. The calculated results with the coupling forces in the roll direction due to the manoeuvring force in the yaw direction are smaller than without it, as shown in Fig. 11. This is due to its positive contribution to the roll moment as shown in Fig. 13.

The Effect of the Coupling Forces from the Maneuvering Forces in the Sway and Yaw Directions in Stern Quartering Waves

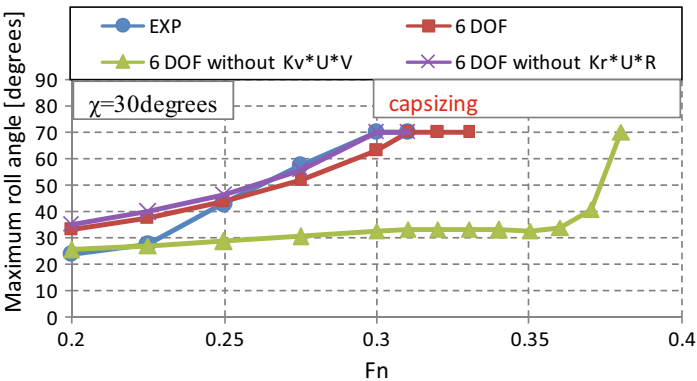
The 6-DOF mathematical model without the coupling forces in the roll direction due to the

The Type of Roll Motions During Pure Loss of Stability

The experimental results of yaw, roll, pitch motions, and rudder angle in stern quartering waves are shown in Fig. 14.

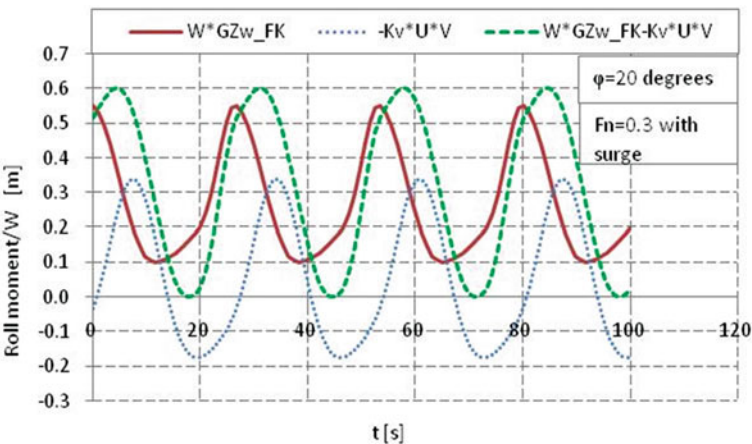
Pure Loss of Stability,

Fig. 11 Comparison of maximum roll angle as a function of the F_n between the experimental results and calculated results with the 6 DOF with and without centrifugal force in roll direction due to the manoeuvring forces in the sway and yaw directions with $\lambda/L_{pp} = 1.25$, $H/\lambda = 0.05$, and $\chi = 30$ degrees



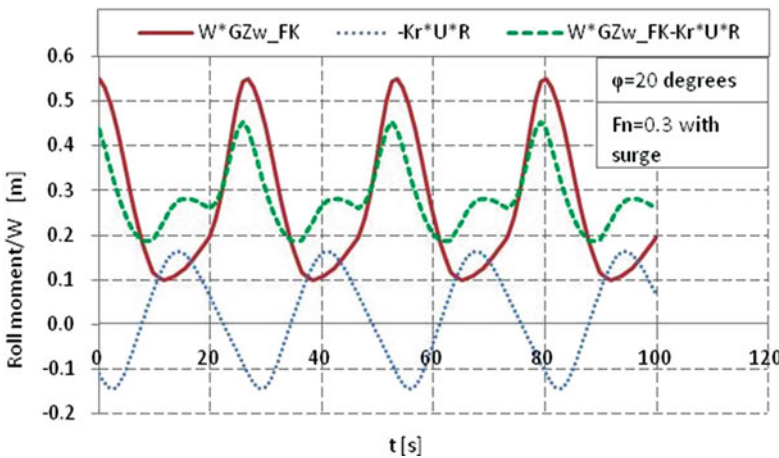
Pure Loss of Stability,

Fig. 12 The contribution to the rolling moments from the coupling force in the roll direction due to manoeuvring motions in the sway direction



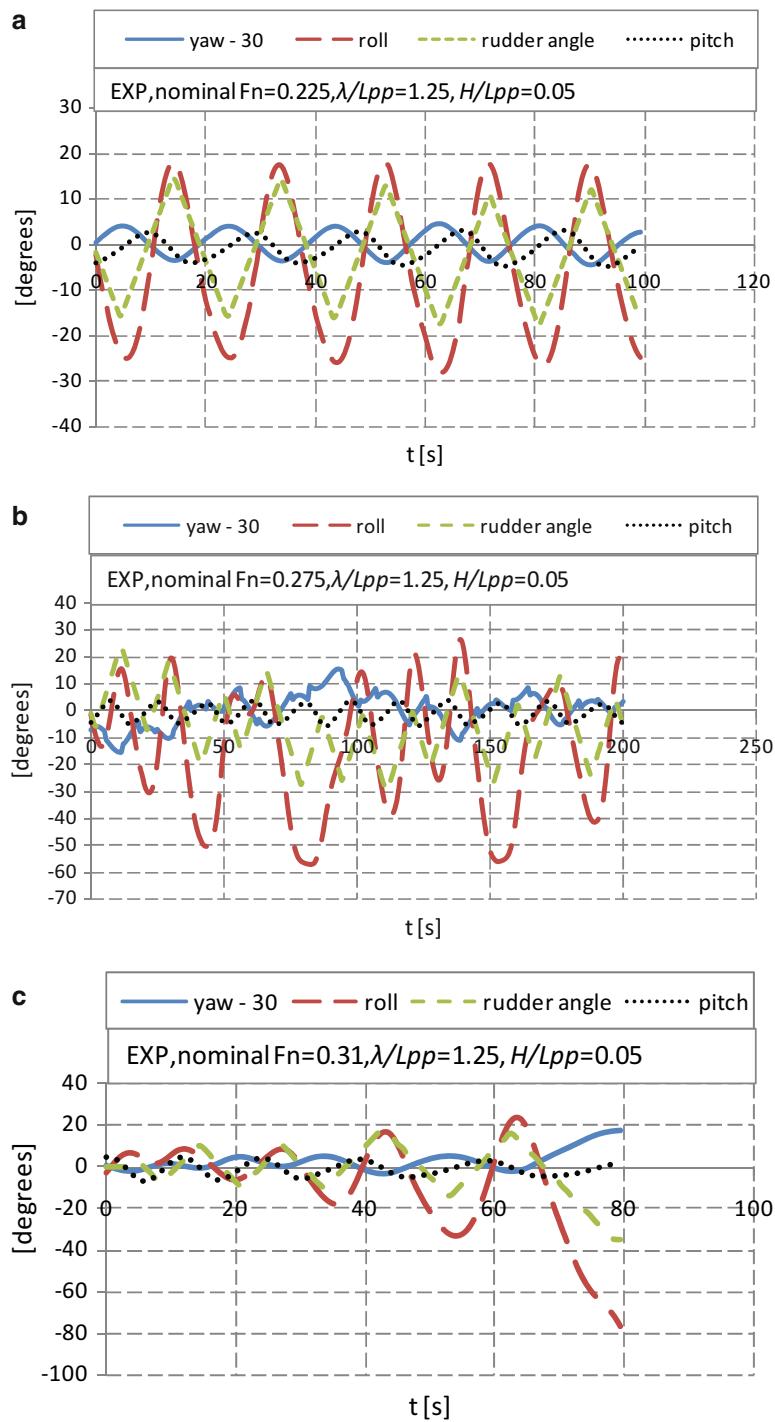
Pure Loss of Stability,

Fig. 13 The contribution to the rolling moments from the coupling force in the roll direction due to manoeuvring motions in the yaw direction



Pure Loss of Stability,

Fig. 14 Yaw, roll, pitch motions, and rudder angle in the free-running experiment with $\lambda/Lpp = 1.25$, $H/\lambda = 0.05$, and $\chi = 30$ degrees. (a) Stable roll motion far from the critical speed of pure loss of stability. (b) Unstable roll motion near the critical speed of pure loss of stability. (c) Capsizing due to coupled pure loss of stability and broaching



A stable periodic roll motion can be found when the ship speed is far from the critical speed of pure loss of stability as shown in Fig. 14a, while an unstable roll motion can be found when the ship speed is close to the critical speed, as shown in Fig. 14b. The capsizing due to pure loss of stability is shown in Fig. 14c, and the yaw angle subtracting the heading angle reaches 20 degrees and the rudder angle reaches the maximum 35 degrees when the ship capsizes. The capsizing occurs due to pure loss of stability before developing the larger yaw angle, i.e., the maximum rudder angle cannot correct the course and obviously broaching stops due to the capsizing of the model. This could be a new phenomenon of capsizing due to coupled pure loss of stability and broaching.

As parametric roll, resonant roll, unstable roll motions, and capsizing due to coupled pure loss of stability and broaching could exist during pure loss of stability, further research should be conducted in the future to shed some light on this complex phenomenon.

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Q

Quadrature Amplitude Modulation (QAM)

► [Underwater Acoustic Communication](#)

R

Radar Navigation System

► [Ship Navigation System](#)

Radio Navigation System

► [Ship Navigation System](#)

Range-Gated Imaging

► [Photoelectric Detection Technology in Underwater Vehicles](#)

Recycling Regulations

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Definition

Recycling regulations are the laws, rules, and conventions issued by authorities which are utilized to regulate ship recycling. They are introduced in the attempt to prevent, reduce, minimize, and, to the

extent practicable, eliminate accidents, injuries, and other adverse effects on human health and the environment caused by ship recycling.

Scientific Fundamentals

Introduction

According to the European Commission (EC), around 1000 large end-of-life ships all over the world are dismantled every year to recycle the steel and equipment. Most of this ship recycling takes place in South Asia, often on tidal beaches and under dangerous conditions which lead to health risks and extensive pollution of coastal areas. In fact, old ships contain many hazardous materials including asbestos, polychlorinated biphenyls (PCBs), tributyltin, and large quantities of oils and oil sludge. In order to reduce those negative effects of this industry, some laws, rules, and conventions have been made by national and international institutions. Nowadays, the most widely used and famous ones among them are the Basel convention, the Hong Kong Convention, and the EU Ship Recycling Regulation.

The Basel convention on the Control of Transboundary Movements of Hazardous Wastes and Their Disposal, usually known as the Basel Convention, was open for signature on 22 March 1989 and entered into force on 5 May 1992. It is an international treaty that was designed to reduce the movements of hazardous wastes between nations and specifically to prevent transfer of

hazardous wastes from developed to less developed countries (LDCs). It does not, however, address the movement of radioactive wastes. The Convention is also intended to minimize the amount and toxicity of wastes generated, to ensure their environmentally sound management as closely as possible to the source of generation, and to assist LDCs in environmentally sound management of the hazardous and other wastes they generate. As of February 2018, 185 states and the European Union (EU) are parties to the Convention. Among them, Haiti and the United States have signed the Convention but not ratified it.

The Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships (or the Hong Kong Convention, for short) was adopted on 15 May 2009 by an IMO Diplomatic Conference of the International Maritime Organization (IMO) held in Hong Kong, China. This Convention covers the design, construction, operation, and preparation of ships so as to facilitate a sustainable ship recycling without compromising the safety and operational efficiency of ships. It also regulates the establishment of an appropriate enforcement mechanism for ship recycling, incorporating certification, and reporting requirements. From 2011, the IMO has developed several guidelines to assist the States Parties in the early implementation of the Convention's technical standards.

The EU Ship Recycling Regulation (1257/2013) entered into force on 30 December 2013 to reduce the negative impacts related to the recycling of EU-flagged ships, especially in South Asia. The Regulation is based on the Hong Kong Convention and aims to implement the Convention quickly, without waiting for its ratification and entry into force. To speed up the formal entry into force of the Hong Kong Convention, the Commission proposal for a Ship Recycling Regulation was accompanied by a draft decision requiring member states to ratify the Hong Kong Convention.

History

Pressure demanding a safer and a more environmentally friendly ship recycling industry has been building up over past decades and has found outlets among politicians and administrations, who

have looked for ways to regulate ship recycling with international common standards. The first attempt at addressing the problem was to try to implement "The Basel Convention on the Control of Transboundary Movements of Hazardous Wastes and their Disposal," which was adopted in 1989 and entered in force in 1992. The Basel Convention protects the human health and the environment against adverse effects that result from the generation and management of hazardous and other wastes. In particular, the Basel Convention focuses on regulating the transboundary movement of hazardous wastes in its effort to protect developing countries from importing hazardous wastes that they are unable to manage in an environmentally sound manner.

However, Basel does not establish a dedicated system for ships. Its provisions, and particularly its system of Prior Informed Consent designating a State of export, did not envisage ships. This has created difficulties in enforcing the Convention to end-of-life ships, especially in the European Union where the Basel Convention is implemented along with an amendment forbidding the export of hazardous wastes to non-OECD countries. Examples of cases where serious difficulties were experienced include the Otapan, the Sea Beirut, the Sandrien, the Margaret Hill, the Tor Anglia, the Onyx, and others. As early as October 2004, the seventh Conference of the Parties to the Basel Convention, in its decision VII/26, invited IMO to consider the establishment in its regulations of mandatory requirements that ensure an equivalent level of control as established under the Basel Convention and also ensure the environmentally sound management of ship dismantling, and "which might include pre-decontamination within its scope."

In 2003, the IMO adopted the IMO Guidelines on Ship Recycling (International Maritime Organisation 2003) and later in 2004 published for the Guidelines for the Development of the Ship Recycling Plan (International Maritime Organisation 2004). The International Labour Organization (ILO) published guidelines regarding safety and health issues in shipbreaking in 2004 which were subsequently translated into the local languages of major ship recycling countries (International Labour Organisation 2008).

Cooperation between IMO, ILO, and the Basel Convention Secretariat resulted in the establishment of a joint working group in 2005 in order to coordinate each party's activities and reduce regulatory overlaps (International Maritime Organisation 2005). A ship recycling conference organized by Lloyd's List in April 2005 discussed inter alia the necessity of a mandatory system.

The EU adopted its "Waste Shipment Regulation" (WSR) which was fully transposed and implemented from both the Basel Convention and the Basel Ban Amendment in 2006. In 2007, the EC published the "Green Paper on Better Ship Dismantling," which was further elaborated in the report "An Integrated Maritime Policy for the European Union" (European Commission 2007) and finalized in the document "An EU Strategy for Better Ship Dismantling" in late 2008 (European Commission 2008).

Prior to this, as early as January 2008, IMO published its "Draft Convention on Safe and Environmentally Sound Recycling of Ships." About the same time, the International Organization for Standardization (ISO) launched a new series of standards with regard to the recycling of ships (Engels 2013). It took just over 3 years since IMO agreed to develop a "new legally binding instrument on ship recycling" through an assembly resolution in December 2005 to develop the "Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships," which is also known as the Hong Kong Convention (International Maritime Organisation 2009). And the Hong Kong Convention was finally adopted at a diplomatic conference held in Hong Kong in May 2009. On 30 December 2013, the EU Ship Recycling Regulation (1257/2013) (European Union 2013) which was based on the Hong Kong Convention entered into force.

Key Applications

Hong Kong Convention

Structure

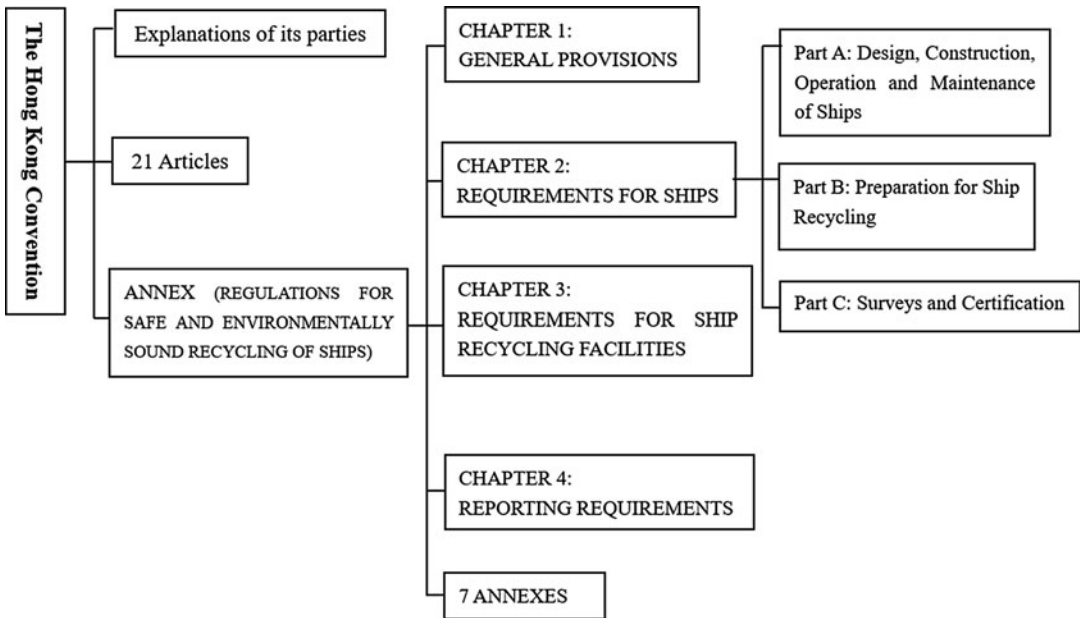
As its core, the Hong Kong Convention contains 21 articles and an annex comprising of another 26 regulations stipulating general provisions as

well as specific requirements for ships, ship recycling facilities, and reports. This annex is followed by additional seven appendices containing a particular range of forms and checklists which are supposed to facilitate compliance with the provisions of the Hong Kong Convention. Furthermore, seven guidelines have been developed under the auspices of the IMO which relate to the implementation of a specific set of major obligations (Engels 2013). Figure 1 clearly depicts the basic structure of it.

Applicability

The Hong Kong Convention applies to "ship" and to "ship recycling facilities" of parties. Here, "ship" means a vessel of any type whatsoever operating or having operated in the marine environment and includes submersibles, floating craft, floating platforms, self-elevating platforms, floating storage units (FSUs), and floating production storage and offloading units (FPSOs), including a vessel stripped of equipment or being towed. And "ship recycling facilities" means a defined area that is a site, yard, or facility used for the recycling of ships. However, ships less than 500 GT, warships, naval auxiliaries, or other ships "owned or operated by a Party and used, for the time being, only on government non-commercial service" and ships that are "operating throughout their life only in waters subject to the sovereignty or jurisdiction of the State whose flag the ship is entitled to fly," if recycled in the same state where they have operated, are exempted from its terms. Even so, each Party shall adopt appropriate measures to ensure such ships act in a manner consistent with this Convention, so far as is reasonable and practicable. With respect to non-Parties ships, Parties shall apply the requirements of this Convention in a manner so as not to afford any favorable treatment (Engels 2013).

"Ship recycling" is defined by the Hong Kong Convention as "the activity of complete or partial dismantling of a ship at a Ship Recycling Facility in order to recover components and materials for reprocessing and re-use, whilst taking care of hazardous and other materials, and includes associated operations such as storage and treatment of components and materials on site, but not their further processing or disposal in separate facilities" (Puthucherril 2010). Therefore, the scope of



Recycling Regulations, Fig. 1 Structure of the Hong Kong convention

the convention is confined only to activities like scrapping and storage in the recycling yard and does not extend to the remaining steps in the recycling chain.

Key Objectives

The Hong Kong Convention is applied to prevent, reduce, minimize, and, to the extent practicable, eliminate accidents, injuries, and other adverse effects on human health and the environment caused by Ship Recycling and enhance ship safety, protection of human health and the environment throughout a ship's operating life.

Major Obligations

Major obligations of the Convention stipulate requirements for ships, requirements for ship recycling facilities, and reporting requirements. Requirements for ships may generally be divided into three parts:

- (1) Provisions addressing the problem of hazardous materials such as Regulation 4 and Regulation 5
- (2) Regulation 9 on the establishment of a ship recycling plan

- (3) The far-ranging Regulation 10 on the establishment of a number of different surveys

Requirements for Ship Recycling Facilities Regulations 15–23 affect ship-recycling facilities by stating certain standards which must be complied with Regulations 17 and 18 addressing the general requirements and the ship recycling facility plan are central to the regulating chapter on ship recycling facilities.

Regulations 24 and 25 contain notification and reporting requirements, thus comprising the final category of obligations. They primarily address the ship recycling facility but require a certain quality of cooperation by the ship owner. By regulating the whole ship recycling process from the intention of the ship owner to recycle a ship until the completion of the recycling procedure and by constantly involving official and private stakeholders, these regulations essentially cover also the recycling process from “cradle to grave.”

Entry into Force

This Convention shall enter into force 24 months after the date on which the following conditions are met (Mikelis 2010):

- (1) Not less than 15 States have either signed it without reservation as to ratification, acceptance or approval, or have deposited the requisite instrument of ratification, acceptance, approval or accession in accordance with Article 16;
- (2) The combined merchant fleets of the States mentioned in paragraph 1.1 constitute not less than 40% of the gross tonnage of the world's merchant shipping; and
- (3) The combined maximum annual ship recycling volume of the States mentioned in paragraph 1.1 during the preceding 10 years constitutes not less than 3% of the gross tonnage of the combined merchant shipping of the same States.

1989 Basel Convention

Basic Scheme

The Basel Convention applies to “transboundary movement” of “hazardous wastes.” The transport is considered “transboundary movement” when it involves any movement of hazardous wastes or other wastes, from one national jurisdiction of one state to or through the national jurisdiction of another state. The waste is “hazardous waste” as is listed in Annex I of the Basel Convention, unless they do not contain the characteristics listed in Annex III.

The Basel Convention has proved to be a basis for actions against substandard shipping in developing countries, which has in some cases forced an improvement in practices. However, the Basel Convention regime has been criticized throughout its operation for being ineffective in solving the problems related to ships (University of Southampton 2013).

Applicability

Whether ships sent and/or sold for recycling fall within the regulatory scope of the Basel Convention, is contingent upon three elements:

- (1) The ships have to be classified as waste.
- (2) The ships have to be subject to transboundary movement.
- (3) Both the State of Export and the State of Import have to be parties to the Basel Convention.

Generally, the Basel Convention is open and broad in its scope of application in order to encompass as many different kinds of wastes as theoretically and practically feasible.

Objectives

The primary objectives of the Basel Convention include minimizing the generation and transboundary movement of hazardous and other wastes and ensuring environmentally sound management of these wastes.

The Parties to the Basel Convention are obliged to minimize the generation of hazardous wastes and to promote adequate disposal within the state where such wastes are generated. The goal is to prevent dumping of toxic wastes onto developing countries. This goal is sought accomplished by reducing the generation of wastes and increase the capacity of the generating state to dispose of its own wastes (University of Southampton 2013).

Main Obligations

It is of note that the Convention places a general prohibition on the exportation or importation of wastes between Parties and non-Parties. The exception to this rule is where the waste is subject to another treaty that does not take away from the Basel Convention. According to the definition as provided by the Basel Convention, ships sent for recycling operations fall, in principle, within the scope of the Basel Convention.

If a State of Export has been established, then it will need to comply with Article 6 and its “prior informed consent” (PIC) regime.

Article 4 of the Basel Convention calls for an overall reduction of waste generation. By encouraging countries to keep wastes within their boundaries and as close as possible to its source of generation, the internal pressures should provide incentives for waste reduction and pollution prevention. Parties are generally prohibited from exporting covered wastes to, or import covered wastes from, non-parties to the convention.

According to Article 12, Parties are directed to adopt a protocol that establishes liability rules and procedures that are appropriate for damage that comes from the movement of hazardous wastes across borders.

Besides, there are some party obligations under the Basel Ban Amendment (Decision III/1). The Basel Ban obligation states that all Annex VII Parties have a special obligation to “prohibit all transboundary movements of hazardous wastes which are destined for operations according to Annex IV A to States not listed in Annex VII.”

Other Related Regulations

1982 UNCLOS

The 1982 United Nations Convention on the Law of the Sea (UNCLOS) establishes for its States Parties an overarching framework of rights and obligations relating to maritime affairs whose provisions on the protection of the marine environment (Part XII), according to many states, reflect rules of international customary law. The convention’s overall significance is explicitly recognized by the Hong Kong Convention. As stipulated by the preamble of UNCLOS, its objective is to establish a form of constitutional framework for the oceans focusing on the role of sovereign states within and toward the marine environment with the main objective of preserving sustainability. The fact that UNCLOS is a framework convention means that it has to be filled with substance by other instruments, both regionally and globally (Engels 2013).

1998 Rotterdam Convention

The Rotterdam Convention on the Prior Informed Consent Procedure for Certain Hazardous Chemicals and Pesticides in International Trade promotes a regulatory regime concerned with transboundary movements of particular hazardous chemicals.

This convention is of particular relevance in the ship recycling context as it serves as an example of a specific set of relevant principles and concepts, such as prior informed consent, which have been agreed upon at the international level regarding transboundary trade with certain chemicals and pesticides. Hence, these considerations might also be relevant with regard to ship recycling, especially if one considers the principal PIC procedure as a means of safeguarding states’ sovereignty.

1972/1996 London Convention

The London Convention on the Prevention of Marine Pollution by Dumping of Wastes and Other Matter, which is planned to be replaced by its Protocol after successful ratification, is an international instrument which aims at preventing “the pollution of the sea by the dumping of waste and other matter that is liable to create hazards to human health, to harm living resources and marine life, to damage amenities or to interfere with other legitimate uses of the sea.” The London Convention is not directly applicable to ship recycling. The convention and its protocol may rather be considered as a complement to ship recycling initiatives insofar as both abandoning ships and dumping them on the high sea still is considered by some actors as a mean of manage ships at the end of their final life cycle.

International Standards and Guidelines

A number of standardization organizations have been providing sets of relevant international and technical instruments, thus inter alia contributing to “harmonization of environmental standards, primarily by facilitating corporate behavior changes.” The most well-known standards are those developed and issued by the International Organization for Standardization (ISO). It uses a similar procedure to develop standards, guidelines, and related information as the OECD and may therefore also be labelled an instance of “network governance.” For ship recycling, the series of standards under ISO 9.000, ISO 14.000, and ISO 30.000 are of particular relevance.

European Initiatives in the Fields of Ship Recycling

Applicability

The territorial scope of the treaty defined in Articles 52 and 355 TFEU³⁴⁸ limits the scope of direct regulatory measures available to the EU in the field of ship recycling. However, the flag state jurisdiction provides just such an example of extraterritorial jurisdiction, which is one of the fundamental exemptions. Therefore, where these competencies have been conferred to the EU, the EU law would be applicable, regardless of

whether the ship is located within member states' waters. Additionally, via port state jurisdiction, the EU legislation could also be made applicable to all ships of foreign registries calling at ports of EU member states.

Objectives

The overall objective of European initiatives in the field of ship recycling is to "contribute to safer and more environmentally sound treatment of end-of-life ships worldwide." In the present situation, and with special consideration for the Hong Kong Convention, the EU measures may thus be based upon provisions for the protection of the environment, provisions aiming at improving the working environment, and provisions on the safety of transport at sea.

Fundamental Principles

According to Article 191.2 TFEU, environmental measures of the EU shall be based on inter alia, the Precautionary Principle and the Polluter Pays Principle.

Potentially hazardous materials shall be identified ex ante as "potential sources of concern," they shall be listed in inventories which shall accompany each ship in a "properly maintained and updated" fashion throughout its operating life in order to facilitate later recycling operations and minimize related risks to the environment and/or human health, and, with regard to supplementary data regarding hazardous materials, upon request an evaluation shall be conducted "of the association between the Hazardous Material in question and the likelihood that it will lead to significant adverse effects on human health or the environment." All these aspects can only be fully understood in the context of the Precautionary Principle, as various legal obligations are established requiring action at a point of time when there is no imminent environmental threat. In addition, capacity building via transfer of knowledge might also "raise the standards of protection and precaution in the countries concerned." As a matter of course, the EU will take these considerations into account when enacting related legislation.

The principle of "Polluter Pays," namely, the 1992 Rio Declaration, Principle 16 states: National

authorities should endeavor to promote the internalization of environmental costs and the use of economic instruments, taking into account the approach that the polluter should, in principle, bear the cost of pollution, with due regard to the public interest and without distorting international trade and investment. Of far greater importance in practice, at least from a European point of view, the principle of "Polluter Pays" has found its way into numerous judgments of the European Court of Justice in recent years. Additionally, the European Commission explicitly is of the opinion that environmentally sound ship recycling is first and foremost the producer's responsibility and should therefore follow the Polluter Pays Principle.

Waste Shipment Regulation (No 1013/2006)

The EU fully transposed and implemented both the Basel Convention and the Basel Ban Amendment by adopting its "Waste Shipment Regulation" (WSR). According to this regulation, ship recycling capacity currently available to ships flying the flags of the EU member states is considerably restricted, effectively to the existing Turkish and British ship recycling facilities. Therefore, the practice is that most of European ship owners obviously make use of loopholes and prefer to contract with substandard Asian ship recycling facilities in order not to face the costs and/or administrative burdens resulting from the application of the WSR.

The EU Ship Recycling Regulation (No 1257/2013)

Structure

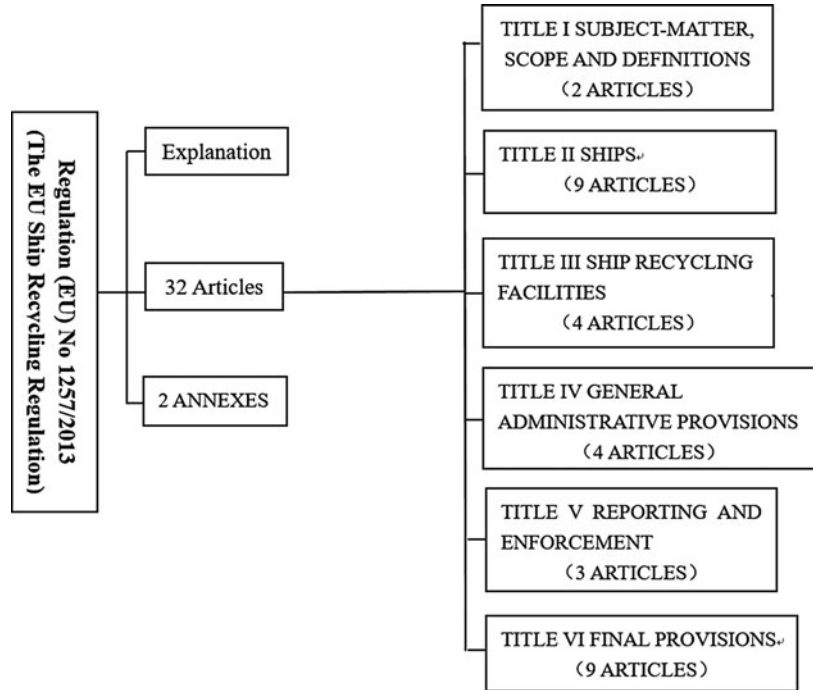
The EU Ship Recycling Regulation mainly contains 32 articles, comprising of 6 titles and 2 annexes which give the list of hazardous material and their control measures. Figure 2 demonstrates the basic structure of the EU Ship Recycling Regulation.

Applicability

The Ship Recycling Regulation applies to large commercial seagoing vessels flying the flag of an EU member state and to ships flying the flag of a third country calling at the EU ports or

Recycling Regulations,

Fig. 2 Structure of the EU ship recycling regulation



anchorages. In order to ensure legal clarity and avoid administrative burdens, ships flying the flag of a member state covered by the new legislation would be excluded from the scope of the Waste Shipment Regulation (EC 1013/2006).

According to the new rules, the installation or use of certain hazardous materials on ships, such as asbestos, ozone-depleting substances, PCBs, PFOS, and anti-fouling compounds, will be prohibited or restricted. Each new ship flying the flag of an EU member state (or a ship flying a flag of the third country calling at the EU ports or anchorages) will be required to have on board an inventory of hazardous materials.

Objectives

As is shown in Article 1, the main purpose of this Regulation is to prevent, reduce, minimize, and, to the extent practicable, eliminate accidents, injuries, and other adverse effects on human health and the environment caused by ship recycling. The purpose of this Regulation is to enhance safety and the protection of human health and of the Union marine environment throughout a

ship's life cycle, in particular to ensure that hazardous wastes from such ship recycling is subject to environmentally sound management.

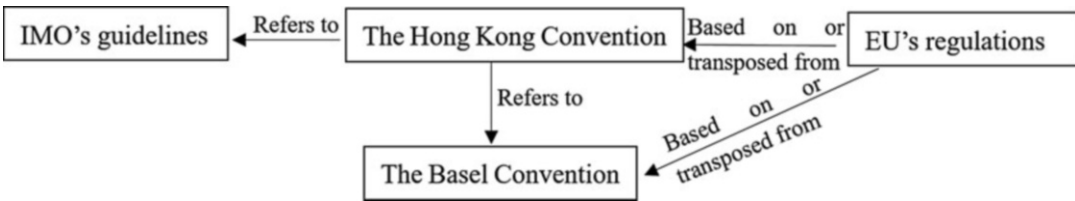
Specially, the Regulation aims to reduce significantly the negative impacts related to the recycling of the EU-flagged ships, especially in South Asia without creating unnecessary economic burdens (Explanatory note 1.2).

Major Obligations

Obligations of Owners of "the EU flagged ships" with respect to:

- (1) Recycling only at a ship recycling facility on the "European List" (which shall be initially published not later than 31 December 2016): see Articles 6 and 16.
- (2) Preparation of a ship for recycling: see Article 6.
- (3) Ship recycling plan: see Article 7.
- (4) Surveys and certificates: see Articles 8, 9, and 10.

For requirements for Ship Recycling Facilities to be included in the "European List," see Articles 13–16 of the Regulation.



Recycling Regulations, Fig. 3 Relationships between typical regulations

Entry into Force and Date of Application

Unlike the Hong Kong Convention, the EU Ship Recycling Regulation makes a distinction between the date of its entry into force and the date of application.

The regulation has entered into force on 30 December, 2013, 2 days after its publication in the Office Journal of the EU (Mikelis 2013).

The date of application is the date after which ships under the scope of the EU Ship Recycling Regulation will start having to have an Inventory of Hazardous Materials (IHM), to be surveyed, certificated, and recycled in line with the requirements of the EU. The EU Ship Recycling Regulation shall apply from the earlier of the following two dates, but not earlier than 31 December 2015 shall apply:

- (1) 6 months after the date that the combined maximum annual ship recycling output of the ship recycling facilities included in the European List constitutes not less than 2.5 million light displacement tones (LDT).
- (2) On 31 December 2018.

There are also some provisions of the European Regulations shall apply from different dates which are explained in Article 32(2–4). Please refer to the original text for more information.

Relationships Between Typical Regulations

Regulations are not independent of each other. Transposition from one regulation or its key mechanisms to binding obligations of another one is common in the process of developing better recycling regulations. Existed conventions and guidelines can be serviced as the basis to the new regulations by giving some pertinent

definitions and basic principles. For example, IMO's guidelines and the Basel Convention provide guidance to the Hong Kong Convention, facilitating both its understanding and implementation. The Basel Convention published the “Technical Guidelines for the Environmentally Sound Management of the Full and Partial Dismantling of Ships” in 2003 which is a predecessor to the Convention (de Larrucea and Mihailovici 2011). Similarly, parts of the EU legislation of recycling are transposed from the Hong Kong Convention and the Basel Convention as shown in Fig. 3.

As for the current existed and used regulations, coordination have been made between their conflicts by adding conflict clauses and other pertinent provisions. Besides, amendments or even some new guidelines and regulations are born from the conflicts and unreasonable places in original regulations.

In short, the make and amendment of one regulation are related to others which have similar scopes and conflicts to it.

Cross-References

- [Complete and Partial Dismantling](#)
- [Inventory of Hazardous Materials](#)
- [Ship Recycling](#)
- [Ship Recycling Facility Plan](#)
- [Ship Recycling Plan](#)

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RED - Reverse Electrodialysis

- [Salinity Gradient Power Conversion](#)

Red Tide

- [Harmful Algal Blooms](#)

Reefer

- [Transport Ship](#)

Reefing

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Definition

Reefing refers to a process of building artificial reefs. An artificial reef is a man-made underwater structure, typically built to promote marine life in areas with a generally featureless bottom, to control erosion, block ship passage, or improve surfing. The typical methods are sinking oil rigs (through the Rigs-to-Reefs program); scuttling ships, rocks, wood, metal; or deploying rubble or construction debris. Other artificial reefs are purposely built (e.g., the reef balls) from PVC or concrete. Shipwrecks may become artificial reefs when preserved on the sea floor. Regardless of construction method, artificial reefs generally provide hard surfaces where algae and invertebrates such as barnacles, corals, and oysters attach; the accumulation of attached marine life in turn provides intricate structure and food for assemblages of fish (Wikipedia 2019).

Scientific Fundamentals

History

The construction of artificial reefs can be traced to ancient times. Persians built artificial reefs for blocking the mouth of the Tigris River to thwart Indian pirates (Williams 2006), and Romans built reefs across the mouth of the Carthaginian harbor in Sicily to trap enemy ships (Hess et al. 2001). Artificial reefs to increase fish yields or for algaculture began no later than the seventeenth

century in Japan, where rubble and rocks were used to grow kelp. The earliest recorded artificial reef in the USA is from the 1830s, when fishermen used interlaced logs to build artificial reefs (Williams 2006). The interest in establishing coastal artificial reefs increased rapidly in the USA during the twentieth century due to the subsequent increase in recreational and commercial fishing and its expenditures (Bull and Love 2019). It was estimated that about 90% of the intentionally deployed wrecks to serve as artificial reefs correspond to the US locations (Ilieva et al. 2019).

The Colonization of Reefing

The colonization of artificial reefs is done progressively. In the beginning, devoid of any form of life, they are rapidly colonized by weed and animal classes like sponges, hydroids, bryozoans, bivalve shells, barnacles, acids, anemones, halcyons, and gorgons and finally, after a few years, by corals with calcareous skeletons like *Acropora*, *Montipora*, *Pocillopora*, and *Pavonia*. This encrusting fauna and flora provide food for sedentary and mobile animals like sea cucumbers, starfishes, sea urchins, crabs, and squids as well as small fishes like snappers, butterflyfish, *Priacanthus*, triggerfish, demoiselle, sergeant major, etc. In turn, these small fish attract pelagic and predator species such as hinds, morays, kingfishes, mackerels, barracudas, etc. Finally, it formed a marine biological system.

Rigs-to-Reefs

Common practice for platform removal uses explosives or mechanical techniques below the seafloor to sever the jacket so that it can be pulled out of water and taken to shore for recycling and/or a landfill (Bull and Love 2019). There are other options that entail reefing the submerged sections of the platform structure (BSEE 2017). Rigs-to-Reefs (RTR) is the process of converting end-of-life oil and gas platforms into artificial reefs. Such reefs have been created from oil rigs in the USA, Brunei, and Malaysia. Offshore oil and gas platforms first began functioning as artificial reefs in 1947 when Kerr-McGee, an American energy company, completed the world's first

commercially successful oil well 11 miles off Louisiana's shore. In the USA, where the practice started and is most common, Rigs-to-Reefs is a nationwide program developed by the former Minerals Management Service (MMS), now Bureau of Safety and Environmental Enforcement (BSEE), of the US Department of the Interior. And the vast networks of energy platforms in the Gulf of Mexico form what is widely regarded as the largest man-made reef in the world, so the Rigs-to-Reefs initiative first began in the Gulf of Mexico as a way to preserve the habitats that form on the rigs' steel scaffolding.

There are three methods of platform removal and reefing to have been used in the RTR process: (a) tow-and-place platform, (b) partial removal in place platform, and (c) topple-in-place platform.

The first and most common removal method, in deepwater applications, involves the use of explosives inside the jacket legs, 15 ft below the mudline. Once the jacket legs are severed by the explosives, the structure is toppled over in a horizontal position on the bottom. This method offers the donor lower costs and time savings. The disadvantage to the use of explosives is the potential mortality of sea turtles, marine mammals, and fish that might be associated with structures, though large numbers of fish are killed, the overall impact to the population was relatively small.

The second removal option involves the partial removal of the upper portion of the jacket and placing it on the seafloor next to the standing bottom portion of the jacket. This method is particularly beneficial with deepwater structures that are converted into reefs. The standing vertical portion of the structure, which must provide at least 85 ft of navigable clearance, remains in place and continues to provide beneficial habitat for a large number of pelagic and other reef fish originally associated with the platform. Also, the upper portion that is removed provides an equal or slightly lower profile to compliment the standing section and increases the overall surface area of the structure for habitat enhancement. This type of removal requires a waiver from the Minerals Management Service. The waivers are required, since existing regulations require severing of the 55-jacket legs 15 ft below the mudline. For

deepwater operations, this method significantly reduces the removal costs and risks for divers.

The third removal option involves divers cutting the jacket legs below the mudline using explosives. Once the legs are severed, the entire structure can be lifted from the sea floor using a derrick barge, towed to a new permitted location, and placed on the bottom in a horizontal or vertical position. Mechanical or abrasive cutters can also be used to cut the legs. This method is typically only used in water depths less than 100 ft. Water depths in excess of 100 ft would significantly increase the risks to divers. There, however, would be no adverse impacts to associated living marine resources. This method is also expensive, labor-intensive, and time-consuming. Although there was no monetary savings using this method, instead of using explosives, the turtles, fish, and encrusting organisms were transported along with the structures to the new reef site.

Benefits

First, oil and gas platforms have proven to be excellent artificial reef material. The National Plan cites five major characteristics or standards for artificial reef materials. These standards, together with siting and management, generally determine the success or failure of an artificial reef project. These include function, compatibility, durability, stability, and availability, and oil and gas platforms appear to possess all these characteristics.

Second, it is well documented that oil and gas platforms function well as artificial reefs by providing habitat for a variety of species, since many of these species are habitat limited. And the steel members of the platform provide the necessary hard bottom substrate for many of the encrusting organisms critically important in developing reef habitat.

Third, oil and gas platforms have proven to be compatible with the marine environment, since generally only the submerged jacket of the structure or that portion of the platform that has never come in contact with hydrocarbons is used.

Forth, oil and gas platforms are also very durable and stable, rarely if ever moving from where

they were placed. And these platforms also appear to be relatively durable and have a life span of 300 years.

Last, during the reefing, partial mechanical removal methods using divers or abrasive cutting tools have provided a method for transferring platforms into reefs with the highest profile in the water column with the least impact on the natural resource and decrease the dangers to sea turtles and marine mammals.

Drawbacks

There are several disadvantages to using oil and gas platforms as artificial reefs. One of the disadvantages is the expense in removing these structures. Derrick barge rates currently run between \$50 thousand to \$100 thousand a day depending on the lifting capabilities of the barge. The size of the structure to be removed determines the size of barge required. This, however, may be turned into a benefit if the savings realized, by not having to take it to shore, can be shared with the entity accepting the ultimate responsibility for the structure. To date, however, the oil and gas industry has dealt only with established state-recognized reef programs. In some areas permitting and meeting state law requirements and the ability to satisfy liability requirements have prevented fishing clubs and private individuals from acquiring platforms as reefs.

Another disadvantage is the method of removal. Currently, state-of-the-art techniques required to sever these structures from the seafloor involve the use of explosives. The concern over the use of explosives stems from their potential impact on endangered sea turtles and marine mammals.

Scuttling Ships

Reefing by scuttling ships is the deliberate sinking of a ship by allowing water to flow into the hull, which can provide an artificial reef for divers and marine life. This can be achieved in several ways – seacocks or hatches can be opened to the sea, or holes may be ripped into the hull with brute force or with explosives.

In the USA, scrap materials of opportunity, deployed without assembly or much modification,

still account for a large portion of reef construction materials. Vessels have served as components of most state artificial reef programs. Where available, and where depth conditions allow for deployment, vessels remain an important reef material to many reef managers, particularly on the Atlantic coast. The earliest record of intentionally sinking vessels for artificial reef fishing is 1935 when four vessels were sunk by the Cape May Wildwood Party Boat Association. Dozens of steel-hulled ships sunk in coastal continental shelf waters along the Atlantic and Gulf Coasts during World War II still provide commercial and recreational fishing opportunities and diving enjoyment more than 60 years later.

To prepare a hulk for sinking as an artificial reef, several things must be done to make it safe for the marine environment and divers. To protect the environment, the ship is purged of all oils, hydraulic fluids, and dangerous chemicals such as PCBs. Much of the superstructure is removed to prevent the hazard of it eventually caving in from corrosion. Similarly, the interior of the ship is gutted of all structures that corrode quickly and would be dangerous to divers if they came loose. The ship is thoroughly cleaned, often with the help of volunteers interested in diving. A significant part of the cost of preparing and sinking the ship comes from scrapping the contents of the ship, including valuable materials such as copper wiring. The hulk's suitability as a diving site is enhanced by cutting openings in its hull and interior bulkheads to allow divers access. The preparation phase removes a significant amount of weight, so the ship sits higher in the water than normal. The ship must be carefully weighed down by filling some sections with water as makeshift ballast tanks to prevent excessive rolling in port or during towing. The ship is towed to the sinking location, usually in shallow waters.

Benefit

First, vessels make interesting diving locations for both recreational divers and technical deep-diving mixed-gas users. Vessels are also regularly utilized as angling sites by recreational fishermen and the charter fishing industry.

Second, vessels used as artificial reefs can, alone or in conjunction with other types of artificial reefs, generate reef-related economic contributions to coastal counties. Especially steel-hulled vessels, when selected for sound hull integrity, are considered durable artificial reef material when placed at depths and orientations that insure stability in major storm events.

Third, due to high vertical profile, the artificial reefs built by vessels attract both pelagic and demersal fishes. Vertical surfaces produce upwelling conditions, current shadows, and other current speed and direction alterations that are attractive to schooling forage fishes, which in turn attract species of commercial and recreational importance, resulting in increased catch rates for fishermen.

Forth, like other artificial reef material, it can augment benthic structure which locally increases shelter opportunities and reef fish carrying capacity in locations where natural structure is sparse or create structure which is more preferable or attractive to certain fish species than locally less complex hard bottom.

Drawback

First, vessels were originally designed and utilized for purposes other than artificial reef construction. They can be contaminated with pollutants, including PCBs, radioactive control dials, petroleum products, lead, mercury, zinc, and asbestos. Hazardous wastes and other pollutants are difficult and expensive to remove from ships.

Besides, vessels typically provide proportionately less shelter for demersal fishes and invertebrates than other materials of comparable total volume. This is because the large hull and deck surfaces provide few, if any, holes and crevices. This lack of shelter from predation greatly reduces the usefulness of a ship as nursery for the production of fishes and invertebrates. Also, while a high vertical profile can be attractive to pelagic fish species, unless a vessel hull is extensively modified to allow for access, water circulation, and light penetration, most of the interior of the vessel is not utilized by marine fishes and macroinvertebrates.

Last, high vertical profile may render some vessels more prone to movement and/or structural damage due to ocean current and wave surge generated by severe storm conditions. And its durability may be compromised by salvage operations during the cleaning process or by the explosives sometimes used to sink these vessels.

Although there are some drawbacks in the reefing by them, they are the most common method of building artificial reefs, compared to its advantages.

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- [Decommissioning of Subsea Facilities](#)
- [Ship Recycling](#)

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Regular Profiling Float (RPF)

- [Profiling Float](#)

Reliability

- [Reliability and Safety in Offshore Engineering](#)
- [Reliability-Based Design \(RBD\)](#)

Reliability Analysis

- [Reliability-Based Design \(RBD\)](#)

Reliability and Safety in Offshore Engineering

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Synonyms

[Dependability](#); [Probability of reliability](#); [Reliability](#); [Responsibility](#)

Definition

Reliability refers to the ability of the product to complete the specified function under the specified conditions and within the specified time. *Safety* refers to the ability of structure to bear various possible actions under normal construction and normal use conditions and to maintain the necessary overall stability when and after accidental events.

Introduction

Offshore engineering is a field of engineering that principally deals with the methods for recovering hydrocarbon resources from deep beneath the seabed, from the installation stage and operation of fixed platform structures as well as floating platforms, their operation, and the laying of pipelines and supplementary oil/gas transport systems.

Recently, phenomenal advancement in science and engineering has been witnessed as a result of rapid growth of the offshore field, particularly in the exploration and development of offshore oil and gas fields in deep waters of the oceans.

A competition increase in the areas of development and installation of innovative offshore

structures, systems, and facilities has led to the question of reliability to be a matter of great interest. In addition during designing, installation and operation of offshore structures technical problems arise which relates to safety and environmental issues. However, offshore structures must be designed, built, installed, and operated with an aim that they work reliably, safely, and efficiently for designed periods of time without maintenance and with limited supervision.

Generally, reliability is taken to mean operation without failure. When an offshore system as a whole does not do what it was designed for, even though none of its components fails, but then it is still charged against the system reliability. A good example of reliability would be an offshore structure that can continue to operate safely when a system or component fails. It is good to know that a good engineering always involves a balance between safety and reliability.

A commonly used definition for a safety of an offshore structure is that it is a state of an offshore structure being safe from undergoing or causing hurt, injury, or loss. Safety is attained by using reliable structures, components, systems, and procedures. The comparative definitions can be summarized by defining reliability as a measure of confidence that the system produces accurate and consistent results, contrary to safety which can be defined as a measure of confidence that the system will not cause accidents.

Reliability as a Probability

De Carlo (2013) gives the definition of reliability as the probability that a component (or an entire system) will perform its function for a specified period of time, when operating in its design environment. Thus he gives his arguments that the elements necessary for the definition of reliability are an unambiguous criterion for judging whether something is working or not and the exact definition of environmental conditions and usage.

Parnami et al. (2012) also defines reliability as the probability that a component, device, system, or process will perform its intended function without failure for a given time when operated correctly in a specified environment. He also adds

that reliability deals with decreasing the frequency of failures over a time interval and is a measure of the probability for failure-free operation during a given interval.

From definition reliability can be mathematically expressed as

$$R(T) = \Pr\{T > t\} = \int_t^{\infty} f(x)dx$$

where $f(x)$ is the failure probability density function and (t) is the length of the period of time (which is assumed to start from time zero).

It is easy to represent “probability of failure” as a symbol or value in an equation, but it is almost impossible to predict its true magnitude in practice, which is massively multivariate. Therefore, having the equation for reliability does not begin to equal having an accurate predictive measurement of reliability.

Key Concepts of Reliability

In order to start a discussion on basics of reliability, coverage of the key concepts of probability becomes crucial. Three important independent concepts can be derived as follows from the definition:

1. Time duration
2. Environmental conditions of use that cause a failure or a fault
3. Functioning parameter value or range that consist the error concept

Fjeld (1978) elaborates that probabilistic reliability is the only meaningful concept which can be used to obtain a logical and objective distribution of risks and safety requirements in the offshore industry. He gives a clear description that probabilistic reliability is a tool for specifying criteria for how safe offshore engineering works should be and for evaluating design and operational requirements from probabilistic criteria for the installation as a whole.

Reliability is restricted to operation under stated conditions. This constraint is necessary

since it is impossible to design a system for unlimited conditions. For example, the deep-sea petroleum Gulf of Mexico oil spill, largest marine oil spill in history, caused by an explosion on the Deepwater Horizon oil rig as a result of failure of subsea blowout preventer stack. Thus, there are extremely strict requirements for safety and reliability in offshore and marine engineering.

Objectives of Reliability in Engineering

Simiu and Smith (1984) state that the objective of structural reliability is to develop design criteria and verification procedures aimed at ensuring that structures built according to specifications will perform acceptably from a safety and serviceability viewpoint.

Structural reliability is also applied to individual components, which are potentially applicable to offshore engineering problems. These include the estimation of failure probabilities, safety indices, and safety (or load and resistance) factors. It is also an essential tool in the risk analysis of the total installation of a structure.

However, it has been stated in Ersdal and Lengen (2002) that the most obvious choice for establishing quantitative probabilities of failure is to use a structural reliability analysis, taking into account the variability of the nature, uncertainties in load, uncertainties in capacities, and often inspections, maintenance, reserve strength, and air gap.

Applying the highlights of decreasing order of priority of the objectives of reliability in engineering from (O'Connor and Kleyner 2012), we can apply the objectives as follows:

1. To prevent or to reduce the likelihood or frequency of failures by applying engineering knowledge and specialist techniques
2. To identify and correct the roots of failures that occurs regardless of the efforts to prevent them
3. To determine ways of coping with failures that occurs, if their causes have not been corrected
4. To apply methods for estimating the likely reliability of new designs, and for analyzing reliability data

From their work more clarification is given on the need to emphasis on order of priority as the most effective way of working, in terms of minimizing costs and generating reliable offshore structures. The ability to understand and anticipate the possible causes of failures forms the primary skill and knowledge of how to prevent them. Having the knowledge of the methods that can be used for analyzing designs and data is of much importance.

Reliability Theory and Models

Gertsbakh (2000) states that an important step in the formal development of the reliability theory is introducing the notion of lifetime. It is defined as the time elapsed since the “birth” of the system till the appearance of certain event which we call “failure.”

Fjeld (1978) modern probabilistic reliability theory seems to be the logical tool to obtain an objective distribution of risks. Failure sources which cannot be considered random and therefore cannot be treated by reliability theory/must be eliminated by control and inspection. Random errors cannot be eliminated by inspection and must be covered by safety factors.

From the review of many researches, it is clear that for better understanding of reliability, we need to develop models that show the relation between various components of a system. This aids in tracing the cause of failure or faults of a given component in an event of a system becoming unreliable.

Parnami et al. (2012) emphasizes that in system reliability analysis, it is important to model the relationship between various items as well as the reliability of the individual items in order to determine the reliability of the system as a whole.

Barlow (2003) highlights that in order to study reliability, you need to transform reality into a model, which allows the analysis by applying laws and analyzing its behavior.

Reliability models are broadly classified into two:

1. Static models
2. Dynamic models

Static models assume that a failure does not result in the occurrence of other faults, whereas dynamic models assume that some failures, so-called primary failures, promote the emergence of secondary and tertiary faults, with a cascading effect.

A diagram-based model for reliability analysis gives a visual representation of the system's components thus giving a good understanding of the system.

Figure 1 below shows diagram-based models frequently used in reliability analysis. Consisting of the following:

1. Reliability block diagrams (RBDs)
2. Failure modes and effects analysis (FMEA)
3. Fault tree analysis (FTA)
4. Event tree analysis (ETA)
5. Decision tree approach (DTA)
6. Root cause analysis (RCA)

Design for Reliability

Design for reliability (DfR) is a process that incorporates tools and procedures to ensure that a system attain its reliability requirements, under its use environment, for the duration of its lifetime. DfR is mostly used as part of an overall design for excellence (DfX) strategy. DfR is implemented in the design stage of a structure to proactively improve the reliability of a given system.

There are mainly two approaches that can be used in the design for reliability. These are:

1. Statistics-based approach (i.e., MTBF)
2. Physics of failure-based approach

Statistics-Based Approach (i.e., MTBF)

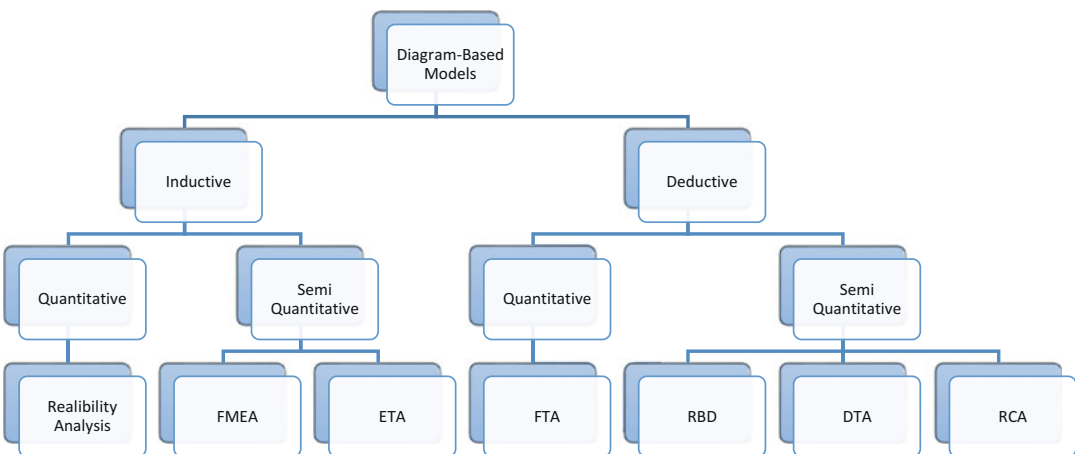
In this approach reliability design begins with the development of a (structure) model; it uses block diagrams and fault tree analysis models to provide a graphical means of evaluating the relationships between various parts of the system. These models use predictions based on failure rates taken from historical data.

Physics of Failure-Based Approach

Physics of failure is the method that depends on understanding the physical static and dynamic failure mechanisms. It accounts for variation in load, strength, and stress that lead to failure with a high level of detail, made possible with the use of modern finite element method (FEM) software programs that can handle complex geometries and mechanisms such as creep, stress relaxation, fatigue, and probabilistic design.

Reliability Testing

“Failure” definition is key aspect of reliability testing. There are many situations where it is not



Reliability and Safety in Offshore Engineering, Fig. 1 Reliability block diagrams

clear whether a failure is really the fault of an offshore structure. Probability estimation can be done from previous data sets or through reliability testing. The sole purpose of reliability testing is to discover potential hitches with the design as early as possible and, ultimately, provide confidence that the system attain its reliability requirements.

Reliability plays a key role in the evaluation of cost-effectiveness of offshore structure systems. Reliability is measured against the standard of requirements of a given offshore structure system's specification. Thus, the methodology used to calculate one offshore structure system's reliability depends on its configuration.

Reliability is not only achieved by mathematics and statistics. Reliability can be attained through process and reliability testing. Reliability testing involves exercising an application so that failures are detected and eliminated before an offshore structure is stationed. MTBF (mean time between failures) for repairable components of an offshore structure system and MTTF (mean time to failure) for non-repairable components can be used to calculate reliability.

Reliability tests can be done on the offshore structure system for the most likely situations under conditions of normal usage and validate that the expected service can be provided by a given offshore structure. As time progresses, more complicated tests can be applied to reveal subtler defects.

Benefits of Reliability in Offshore Engineering

Some of the benefits of reliability are listed as follows:

Safety: Some offshore structures fail due to unintended or unsafe conditions leading to injury or loss of life. Reliability engineering tools assist in identifying and minimizing safety risks before their occurrence.

Time: Failures which are not planned for cost time to resolve. Using reliability concepts we can reduce failures and loss of time.

Design: Enhancing the design team's reliability engineering capabilities through training and working closely with reliability professionals to make decisions having fully considered the impact on reliability of the offshore structures. This reduces the need for expensive redesign or rework costs to address reliability related design errors at a later stage.

Expectation: Offshore structures work under environmental and use conditions. Creating an offshore structure that matches the expectations imposed by the operators of the structures permits the structure to work as expected. Getting to know the conditions well allows the design to be met without over designing thus optimizing the structure cost and operator's satisfaction.

Distribution: With fewer failures and optimized maintenance work of offshore structure results in fewer spare parts in the overall logistics system. This minimizes the costs for logistics, transportation, and storage for spare parts and eventually service labor costs are minimized.

Liability: Offshore structure can cause the loss of property. Failure minimization and damage mitigation caused by any failure reduces the exposure to liability for the property loss.

Conclusion

Reliability in offshore engineering is a broad topic to be covered since the offshore structures are complex in nature. A lot of areas which need further research work by the researchers in the area of reliability of these offshore structures, more so in uncertainty quantification, failure analysis of complex systems and life testing of offshore structure and equipment.

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Reliability Based Design (RBD)

- ▶ [Concept Design](#)
- ▶ [Preliminary Design](#)

Reliability Based Design Optimization (RBDO)

- ▶ [Reliability-Based Design \(RBD\)](#)

Reliability-Based Design

- ▶ [Risk-Based Design for Ship and Offshore Structures](#)

Reliability-Based Design (RBD)

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Synonyms

[Probability theory](#); [Reliability](#); [Reliability analysis](#); [Reliability based design optimization \(RBDO\)](#); [Uncertainty theory](#)

Definition

In the development of a product, it is important to ensure its reliability in the design stage, which enables the concept of reliability throughout the full life cycle of product design, operation, maintenance, and decommission. Such a design method is called reliability-based design (RBD). RBD methods take safety factors into consideration to account for uncertainties, so that established target reliability levels for products can be achieved. The main research contents of RBD include uncertainty theory, reliability analysis, and reliability-based design optimization (RBDO).

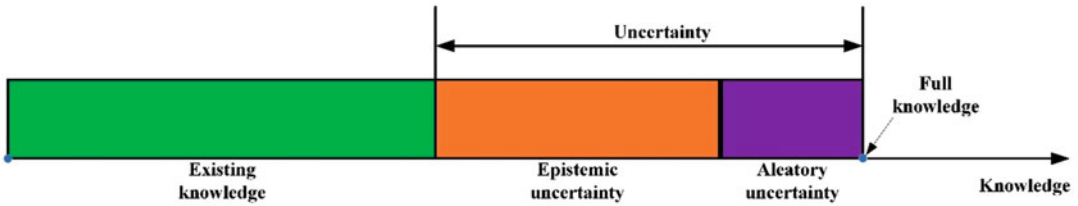
Scientific Fundamentals

Uncertainty Theory

At present, uncertainty is mainly divided into two categories, including aleatory uncertainty and epistemic uncertainty (Oberkampff et al. 2004). Aleatory uncertainty describes the nature of random variations inherent in physical systems or environments, and epistemic uncertainty is originated from the lack of knowledge or information. The relationship between mastery degree on knowledge and uncertainty is shown in Fig. 1.

With the development of science and technology, the understanding of a certain natural phenomenon becomes more and more profound, so the epistemic uncertainty will gradually decrease with the increase of knowledge. However, many natural phenomena have inherent contingency, and sometimes the phenomena are contrary to the conclusions obtained through the existing scientific observation, which reflects the inherent uncertainty of natural phenomena (i.e., aleatory uncertainty). The mathematical theory used to describe uncertainty is the basis for dealing with uncertainty in practical problems.

The most popular method of describing aleatory uncertainty is probability theory. The founder of classical probability theory (CPT) was the Swiss mathematician Bernoulli, who established the first limit theorem named Bernoulli law of large number in probability theory. In 1933,



Reliability-Based Design (RBD), Fig. 1 The relationship between cognition and uncertainty

A.N. Kolmogorov summarized the basic properties and relations of events and their probabilities with the ideas of set theory and measure theory, established the axiomatic system of probability, and laid the theoretical foundation of probability theory (Durrett 2019). For epistemic uncertainty, the widely used description methods are interval theory (Moore 1966), evidence theory (Shafer 1976) and possibility theory (Zadeh 1978), generalized information theory (Klir and Wierman 2013), etc. With the development of uncertainty theory, many scientists are working on a unified uncertainty theory that can be used to describe both aleatory and epistemic uncertainties. Cui and Blockley (1990) combined probability theory and interval theory to obtain interval probability theory (IPT) and employed it as a method to measure the credibility of evidence in knowledge system. Based on quantum decision theory (QDT), Papageorgiou and Kamperi (2017) proposed a quantum-based decision-making model which can be employed in engineering design; this model can also capture the complexity of human decision making under uncertainty. The proposed model is able to describe not just the aleatory uncertainty but also the subconscious intuitive feelings and cognitions of the decision makers (i.e., epistemic uncertainty).

Reliability Analysis

The ability of an engineering system or product to accomplish a specified task under certain conditions and specific time is referred to as reliability. In the design stage, the specified function of an engineering system can be expressed as a function of the input variable $X = (x_1, x_2, \dots, x_n)$, which is usually called the state function $g(X)$. The value of state function $g(X)$ can be used to judge whether

the function of engineering system can meet the design requirements. The most commonly used judgment logic is two-valued logic: when the state function exceeds a certain state, the engineering system cannot meet the requirements of a certain function stipulated in the design. This particular state is called the limit state of the function. If the engineering system can meet the functional requirements specified in the design, the system is reliable, otherwise, the system is unreliable. The reliability of the engineering system is usually divided according to the zero point of the state function:

$$\begin{cases} \text{reliable} & g(X) < 0 \\ \text{unreliable} & g(X) \geq 0 \end{cases} \quad (1)$$

Then the judgment function of two-valued logic can be expressed as:

$$I(g(X)) = \begin{cases} 1 & g(X) < 0 \\ 0 & g(X) \geq 0 \end{cases} \quad (2)$$

Reliability analysis is closely related to the uncertainty theory used to describe the uncertain variables. When the uncertain variable X is described by different uncertainty theories, the reliability analysis method of system state function $g(X)$ is also different. When uncertain variables are described as random numbers by classical probability theory, reliability analysis is called probabilistic reliability analysis. When uncertain variables are described as fuzzy numbers by fuzzy set theory, reliability analysis is named fuzzy reliability analysis.

There are two main reliability analysis methods, including probabilistic reliability analysis method and fuzzy reliability analysis method.

Probabilistic Reliability Analysis Method

In probability theory, the uncertain variable $X = (x_1, x_2, \dots, x_n)$ is considered as a random variable obeying a certain probability density distribution, and the value of state function $g(X)$ of the system is also a random variable. In this case, the failure probability P_f of the system can be expressed as:

$$P_f = \int \dots \int I(g(X)) f_X(x) dx_1 \dots dx_n \quad (3)$$

where $f_X(x)$ represents the joint probability density function of the random variable X .

It can be observed from eq. (1) that the calculation of failure probability is a multidimensional integration process, and it is very difficult to calculate this integral in practical problems with high dimension. Therefore, various approximate methods for reliability estimation have been established; these approaches are called reliability assessment methods (Moffitt 2010). At present, reliability assessment methods can be divided into four main categories:

- (1) Reliability Index Approach (RIA): First Order Reliability Method (FORM), Second Order Reliability Method (SORM), etc.
- (2) Monte Carlo method: Crude Monte Carlo (CMC), Improved Monte Carlo method (IMC), etc.
- (3) Performance Measure Approach (PMA): Mean Value (MV), Advanced Mean Value (AMV), Hybrid Mean Value (HMV), etc.
- (4) Approximation model method: Response Surface Method (RSM), Neural network (NN), etc.

Fuzzy Reliability Analysis Method

When the judgment logic function is fuzzy membership function or the uncertainty variable is described by fuzzy theory, the reliability analysis is called fuzzy reliability analysis. Fuzzy reliability analysis methods can be divided into two types:

- (1) Reliability analysis based on fuzzy logic
- (2) Reliability analysis based on fuzzy variables

Because the development of fuzzy theory is not mature and the calculation amount of the fuzzy theory is far beyond the acceptable range, the fuzzy reliability analysis method has not reached the level of practical engineering applications.

Reliability Based Design Optimization (RBDO)

The reliability of engineering products is closely related to the design, manufacture, assembly, and management in the process of product development, among which design plays a decisive role in the reliability of engineering products. RBDO adopts the uncertainty theory based on probability theory, and the research focus of RBDO is the influence of uncertain factors on the constraint function of the engineering product design optimization model. In the objective function, value of the uncertain variable is replaced by its mean value. In general, the optimal design points of engineering problems are located on the boundary of feasible region, and the uncertainty change of design parameters or design variables is likely to make the optimal design points fall out of feasible region due to the disturbance. As mentioned above, since most of the constraints that work are important mandatory index requirements such as safety, this means that there is a great possibility that the optimization design point cannot meet the mandatory index requirements of the actual engineering system, that is, the reliability of the optimization design is not up to standard, so it is important to research the reliability-based design optimization methods (Pan and Cui 2020). In RBDO, the optimization model can be formulated as:

$$\begin{aligned} & \min f(d, u_X) \\ & s.t. \quad P(g_u(d, X) < 0) < P_f^t \\ & \quad g_d(d) \leq 0 \\ & \quad d^l \leq d \leq d^u \\ & \quad u^l \leq u_X \leq u^u \\ & \quad \sigma \text{ or cov is known} \end{aligned} \quad (4)$$

where d is called design variables, X refers to the random variable and u_X refers to the mean value of X , g_u and g_d represent the uncertain constraint function and the deterministic constraint function,

respectively. $P(g_u(d, X) < 0) < P'_f$ is called probabilistic constraints. Probabilistic constraints are the biggest difference between RBDO optimization model and deterministic optimization model. There are three common RBDO algorithms to solve RBDO problems, including double loop reliability-based design optimization (DLRBDO), sequential optimization and reliability assessment (SORA), and safety factor based sequential optimization and reliability assessment (SFSORA).

DLRBDO

As shown in Fig. 2, DLRBDO algorithm (Allen and Maute 2004) includes the process of reliability analysis, so the algorithm includes two nested iterative processes. The external iteration is the optimization algorithm and the internal iteration is reliability analysis. Since DLRBDO algorithm will lead to a sharp increase in computation, the approximate model can be used to replace the original high-precision state function. DLRBDO algorithm based on approximate model has been widely used in engineering field (Gayton et al. 2003).

SORA

SORA algorithm was first proposed by Du and Chen (2002). In SORA algorithm, the relationship

between optimization and reliability analysis is no longer nested, but sequential. Reliability analysis is executed after the finish of optimization process, which can greatly reduce the calculation amount.

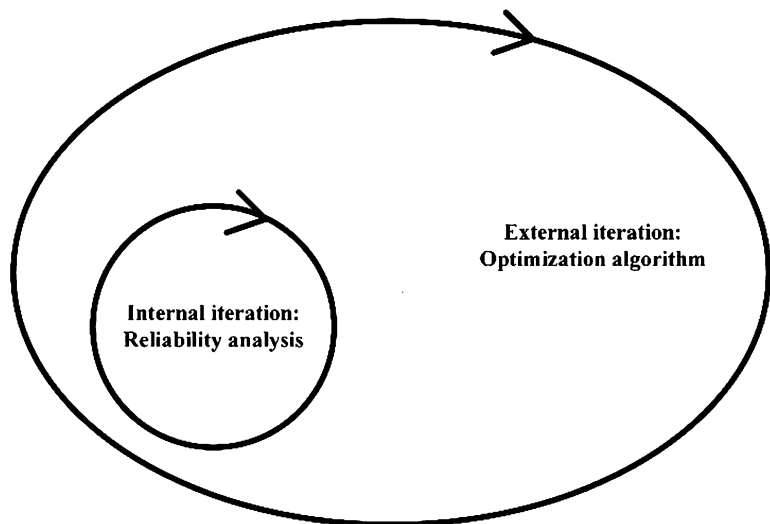
SFSORA

In engineering design, the most common method to ensure reliability is the safety factor method. Pan and Cui (2020) combined SORA algorithm with the safety factor method and put forward a new SORA algorithm based on safety factor, which is called SFSORA.

Key Applications

With the development of uncertainty theories and reliability analysis methods, reliability-based design methods have been widely applied to various fields. Based on probability theory, Ronold and Larsen (2000) presented a reliability-based design method and applied it to analyze the safety of a wind-turbine rotor blade against failure in ultimate loading. By using perturbation method, second moment method, and reliability-based design method, Zhang and Liu (2002) put forward a practical and effective method for the reliability-based design of automobile components. Zhang

Reliability-Based Design (RBD), Fig. 2 Illustration of double loop reliability-based design optimization



et al. (2003) presented a new reliability-based design methodology by combining the perturbation method and reliability-based design theory, and the proposed methodology was applied to the reliability-based design of gear pairs, and the theoretical formulae of reliability-based design of gear pairs were obtained. Phoon et al. (2003) developed a reliability-based design framework for transmission line structure foundations, and the role of the proposed RBD method is presented within the context of geotechnical limit state design. A design methodology that combines reliability-based design optimization and high-fidelity aeroelastic simulations for the analysis and design of aeroelastic structures is proposed by Allen and Maute (2004). In this work, to account for uncertainties in design and operating conditions, a first-order reliability method (FORM) is used to approximate the system reliability. To develop a strategy for solving reliability-based design optimization (RBDO) problems that remains applicable when the performance models are expensive to evaluate, Dubourg et al. (2011) proposed a surrogate model based RBDO method by using the kriging model, and the effective of the strategy is validated by comparing with other approaches available in the literature on three academic examples in the field of structural mechanics. A methodology for reliability-based design and assessment of an ageing fixed steel offshore structure was established by Mat Soom et al. (2016) to support detailed re-assessment applied to the management of the structure's safety, integrity analysis, and reliability by evaluating the loading acting on the structure. By integrating the coupled corrosion fatigue model into the reliability-based design optimization, Saad et al. (2016) applied the proposed RBDO method to the life cycle cost of the structure subject to deterioration processes of fatigue and corrosion. To maximize the characteristics of tailor rolled blank (TRB) structures, a multiobjective and multicase reliability-based design optimization (MOMCRBDO) was developed by Sun et al. (2017) to optimize the TRB hat-shaped structure, and the radial basis function (RBF) metamodel was adopted to approximate the responses of objectives and

constraints, the nondominated sorting genetic algorithm II (NSGA-II), coupled with Monte Carlo Simulation (MCS), was employed to seek optimal reliability solutions. Luo and Hu (2018) presented an efficient reliability-based design tool for stone-filled crib seawalls, in this work, the first-order reliability method is integrated into spreadsheet to facilitate the reliability analysis for practical application. Homaei and Najafzadeh (2020) proposed a reliability-based probabilistic evaluation framework to assess the effect of various parameters on the reliability for the scour depth to be less than a specific value.

Cross-References

► Design of Submersibles

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Remote Operated Vehicle (ROV)

- [AUV/ROV/HOV Propulsion System](#)

Remotely Operated Vehicle

- [Remotely Operated Vehicle \(ROV\)](#)

Remotely Operated Vehicle (ROV)

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Synonyms

[Remotely operated vehicle; ROV](#)

Definition

The **Remotely Operated Vehicle (ROV)** is a kind of submersible, remotely operated by a crew aboard a supporting vessel, floating platform, or on land. Generally, ROV works on the principle of teleoperation where slave system, in this case it is the ROV body, interacts with hazardous extreme environment and a master system, in this case human operator, is placed at safe and comfortable location (Agba [1995](#); Costa et al. [2012](#)). Therefore, the ROV can be remotely controlled to perform the underwater activities with the manipulators and other tooling capabilities. The umbilical cable can supply electrical power to the ROV body; meanwhile, the ROV body can also return the video or other data information to the supporting vessel through the umbilical cable.

Scientific Fundamentals

ROVs have the largest number in the world, which is the most widely used, most complex, and most powerful submersible as compared to other kinds. In addition, ROVs have the characteristics of strong work applicability, function flexibly extended, no constraint on the work endurance, no risk of the crews, etc. With the umbilical cable, ROVs can perform the

complicated tasks with higher strength and longer duration on a fixed point under the very severe deep sea circumstance (Tao and Chen 2016).

System Composition

The system composition of an ROV is closely related to its use and working depth. Generally, the ROV system includes eight subsystems, which are ROV body, tether management system (TMS), tether/umbilical cable, launch & recovery system, deck operation & control system, power supply system, manipulator & tooling package. Among all these subsystems, the ROV body, umbilical cable, deck or aboard operation & control system, and power supply system are compulsory (Lian et al. 2015). In addition, other configurations can be added depending on the system size and work demands, for instance, the drilling machine and sampler.

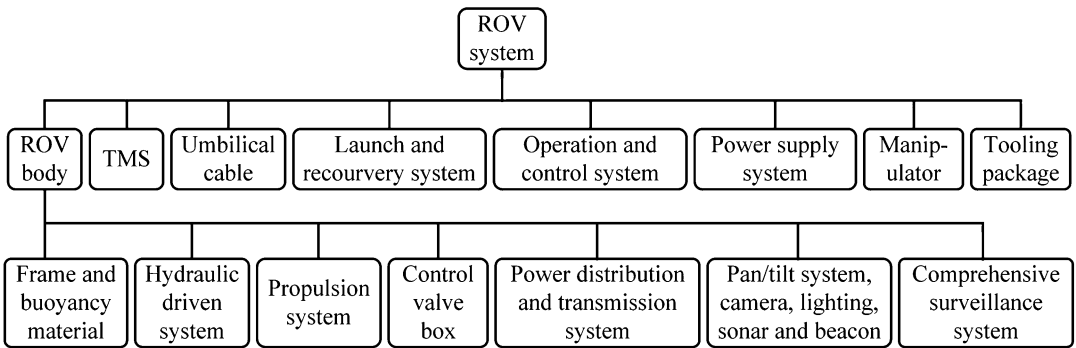
The whole ROV system can also be divided into the underwater and above-water parts. The above-water part consists of launch & recovery system, the deck operation & control system, umbilical winch and power supply system; the underwater part consists of the ROV body, tether management system (TMS), manipulator & tooling package (Lian et al. 2015). Undoubtedly, the two parts are connected by the umbilical cable, which can transmit the control signal and power from the supporting vessel to the ROV body. Meanwhile, the ROV body can also return the captured video and other information to the supporting vessel. In the comprehensive consideration of the water depth and work demanding, the ROV body can

be launched and recovered through either the single umbilical cable or the TMS. Diagram of the system composition for the ROV is shown in Fig. 1, and system configuration for the ROV body is shown in Fig. 2 (Lian et al. 2015).

ROV Types

As for ROV types, there are many different classifications. Depending on size and weight, ROVs can be classified into micro-sized ROV, small-sized ROV, medium-sized ROV, large-sized ROV, and super-sized ROV. According to uses, ROVs can be divided into observation-class ROV and work-class ROV. They can also be divided into work-class ROV and heavy duty work-class ROV depending on their work abilities and divided into free-swimming ROV and bottom-crawling ROV according to the motion mode; furthermore, they can also be divided into electric ROV and hydraulic ROV depending on the driving mode (Chen et al. 2018).

Small-sized ROVs are relatively simple to control and they are mainly used for underwater observation; therefore, they are observation-class ROV. Medium-sized ROVs commonly have the weight around hundreds of kilograms. Besides the observation function, they are also equipped with manipulators and sonar system; thus, they can dosimple works and are ability to fix their position. Large-sized ROVs always have the weight of several tons. They have very powerful propulsion system, and are equipped with a lot of devices, such as underwater TV, sonar, tooling package, multifunctional manipulators, etc. Super-sized ROVs have the weight of up



Remotely Operated Vehicle (ROV), Fig. 1 Typical system composition of ROVs



Remotely Operated Vehicle (ROV), Fig. 2 System composition of the ROV body (Lian et al. 2015)

to dozens of tons, which are typically used for special underwater works, for instance, the pipeline burial. It needs to be mentioned that large-sized ROVs are capable of performing very complex heavy duty underwater works; therefore, they are currently the most widely used kind of ROVs on the exploitation of the offshore oil & gas field. Figure 3 shows the images of the ROVs with different size.

Historical Development

ROVs are the earliest developed and most widely applied submersible among all kinds of submersible. The earliest ROV was designed in 1953 by Dimitri Rebikoff and named as “POODLE,” which was mainly used in the study of archaeology. It had little impact on the ROV development, but it was the origin. Due to limitation of technologies in the early day, the ROVs had a lot of problems, such as leakage, noise, unstable performance, frequent maintenance, hydraulic system being prone to closing down, and so on (Christ 2013).

In 1961, the USA army developed a flexibly operated underwater video system “XN-3,” which finally evolved into the cable-controlled underwater research vehicle (CURV). Later, the army improved CURV, which also made a lot of great events. For instance, the CURV II salvaged a

nuclear bomb at the water depth of 869 m offshore the Spanish sea (Fig. 4a). In 1973, the CURV III was emergently sent from San Diego, USA, to Ireland and performed the deepest underwater rescue in history when it rescued two men trapped at depths of 480 m in the sea for 76 h. After that, the US Army also developed a lot of more complex tethered vehicles, including the first small-sized portable observation-class ROV “SNOOPY” (Fig. 4b), and subsequently a fully electric ROV “Electric SNOOPY” equipped with sonar and other sensors. Subsequently, the US Army has input a huge of money into the Hydro Products in San Diego, USA, which promoted the ROV industry to grow in leaps and bounds. At this time, one of the famous ROV is the “TORTUGA.”

In the whole year of 1974, only 20 ROVs were produced in the world, including the Fenland “PHOCAS,” Norway “SNURRE,” UK “CONSUB 01,” as well as the Soviet Union “CRAB-400” and “MANTA.” The ROV industry was still insufficient to pry the market occupied by the HOVs (human occupied vehicles) and divers. However, the great change came in the period till to the end of 1982; there were about 500 ROVs all over the world. There was also a great change on the funding source to produce these ROVs. From 1953 to 1974, 85% of the



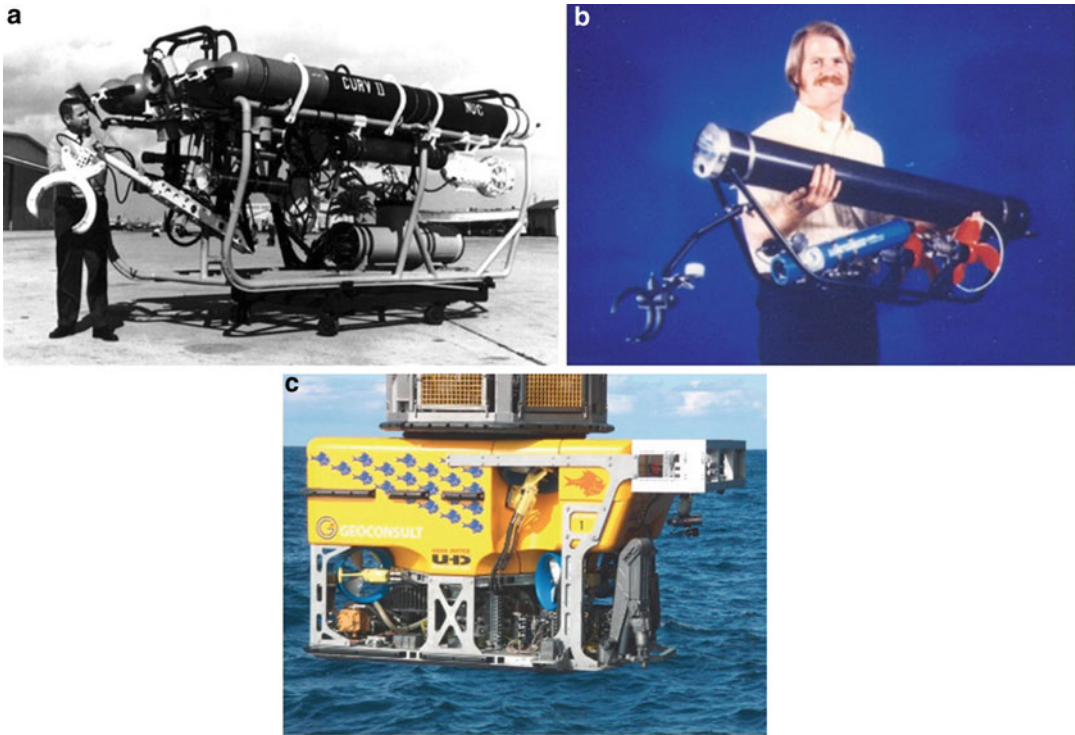
Remotely Operated Vehicle (ROV), Fig. 3 ROVs with difference size: (a) small-sized ROVs, (b) China “HAIMA”-4500 ROV (Lian et al. 2015), and (c) the world’s biggest subsea robot SMD ultra trencher 1 (Machin 2011)

fund was provided by the government, whereas 96% of this fund for about 350 ROVs was provided by the private companies from 1974 to 1982.

At this time, the technology of ROV became matured and stable. The most important thing was that the ROV was accepted by a lot of marine companies. In 1983, there was even an international conference of ROV’83 with the theme of “A Technology Whose Time Has Come!” More advanced countries, such as USA, Canada, France, Italy, Netherland, Norway, Sweden, Germany, UK, and Japan, entered the ROV industry and produced more kinds of ROVs, including the micro-sized observation-class ROV. Some of these ROVs were also

possessed by the society association or academic and research organization.

Till the 1990s in the twentieth century, the technology of ROV had become quite matured, and the ROVs acted positively in most of the oceans and performed many kinds of complex works. The US Army’s CURV III could reach the 6128 m water depth, and their advanced tethered vehicle (ATV) even reached 7012 m below the sea surface. The ROV “KAIKO 11000” designed and manufactured by the Japan Agency for Marine-Earth Science and Technology (JAMSTEC) even reached the deepest area of the seabed in 1994 (Barry and Hashimoto 2009), the Mariana Trench of the Pacific Ocean. However, in 2003, it was lost in an ocean



Remotely Operated Vehicle (ROV), Fig. 4 (a) US Navy's CURV II vehicle, (b) US Navy's hydraulic SNOOPY, and (c) Schilling Robotics UHD

investigation due to the break of the neutral-buoyance umbilical cable. The booming offshore oil industry increased the demanding for the advanced submersibles, especially when the works, such as ocean floor drilling and exploitation, could not be finished by the divers due to the high water pressure in deep sea region, which forced the ocean oil industry and ROV developer to cooperate together. For instance, Schilling Robotics UHD ROV could perform a lot of very complex works, such as underwater stalling, operation, and maintenance (Fig. 4c).

In the twenty-first century, many ROV manufacturers started to develop the observation-class ROV and medium-sized ROV. As for the ROV education, the European and North American countries have held some training plans to fulfill the demand of ROV operation with the booming ROV industry. Marine Advanced Technology Education (MATE) center aims to encourage the high school and college-aged young engineers into the field of subsea robotics. Today,

the center has already sponsored several international competitions.

Key Technologies in Developing the Deep Sea ROV

Typically, the ROV is an interdisciplinary system. As for the ROV design, the engineer needs to consider the launch and recovery system, layout configuration of ROV body, propulsion system, maneuverability, as well as the work tools. Among all these design procedures, some of them are called as key technologies and will be introduced as follows:

1. Navigation and position technology

With the development of the modern satellites, to lock on a target on land or on the sea surface is relatively easier, while it is quite difficult to get the position of an object underwater. Therefore, the navigation and position is the first problem that should be solved when operating the ROV for sea bottom works.

The operator needs to know the current position of the ROV in the deep sea and the position the ROV is going to. Then the operator can lead the ROV to the next position.

2. Comprehensive control technology

The comprehensive control technology is mainly used to distribute the power for the whole system, to control and monitor devices, direction, video surveillance, lights, pan/tilt system, leakage of electricity, leakage of hydraulic oil, and so on. The comprehensive control system is designed with high technology and complex configuration. It is also the head of the ROV system and the key selling point in the ROV industry. The specific key technologies include system control, real-time data inspection, warning, emergency isolation, multidata merging, real-time attitude inspection, touch control, data record and analysis, auto-heading, auto-depth and auto-altitude navigation, etc.

3. Umbilical cable

As the important part connecting the ROV body to the sea surface support vessel, the umbilical cable functions as power transmission, communication, control command transmission, data transmission, launch, and recovery. Generally, the umbilical cable consists of optical fiber, electric cable, mechanical part, filler, Kevlar, and other materials, which are integrated through the way of helical winding. Therefore, the umbilical cable is on a high comprehensive level, has large mechanical strength, has good corrosion resistance, and can be repeatedly released and recovered. In addition, the umbilical cable can be designed/chosen according to the working depths. Generally, there are two kinds of umbilical cables, armored cable and neutral buoyancy cable. As the armored cable is so heavy that the length limitation is generally no longer than 6000 m. Undoubtedly, the umbilical cable is one of the key technologies which restricts the development of the 11,000 m ROV system. Thus, to develop the light, high strength, and abrasion-resistant umbilical cable, which can be repeatedly released and recovered, is the hot spot and trend in the future.

Key Applications

The ROV has a wide range of applications; some of the specific areas include survey and inspection, deep sea oil & gas production, deep sea mining, military, educational outreach, underwater photography, fisheries and aquaculture, civil construction, and public safety (Tao and Chen 2016; Chen et al. 2014).

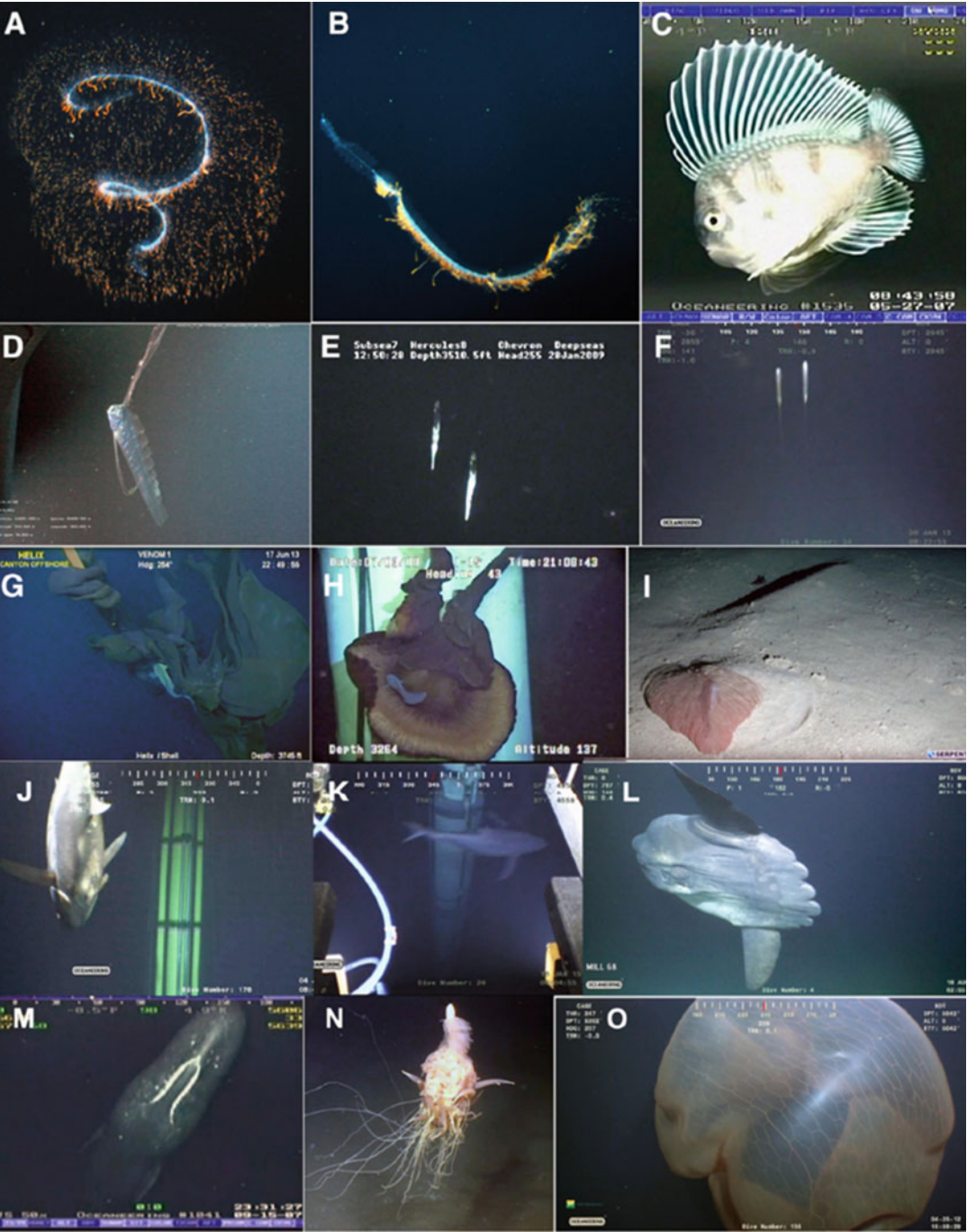
Survey and Inspection

ROVs used for the survey and inspection are clearly observation-class ROVs or observation-class ROVs with payload option. ROVs are our eyes and hands in the deep sea. They can be used for many science and geological surveys, for instance, capturing the images of marine lives and sea floor, sediment sampling, marine life collection, geological drilling and sampling, water sampling, gauging the health index of marine lives and environment, estimating pollution level, searching for new living species and minerals, and so on. Figure 5 shows the images of marine lives observed by the ROVs.

Observation-class ROVs can also be used to locate and position the beacon markers, lay the seabed observation network and underwater structures, inspect pipeline, jacket and even the marine hull of vessels. These ROVs usually have the ability to hold their position in strong currents, and are equipped with manipulators, lighting lamp, camera, sonar, and strobe flasher.

Deep Sea Oil & Gas Field

Mission of ROVs in the deep sea oil & gas field is a traditional and important application (Shukla and Karki 2016). With steep depletion of major onshore and shallow-water-offshore oil fields, new search of fossil fuel is moving toward deep-water and ultra-deep water offshore fields. Previously, companies can employ the divers to do some inspection and installing work in the shallow-water-offshore oil & gas fields, while it is impossible in the deep water. In addition, replacement of divers with robotic devices to accomplish complex tasks is an unavoidable trend with automatization of the overall offshore oil & gas industry. ROVs are used for many



Remotely Operated Vehicle (ROV), Fig. 5 Examples of ROV observations of marine life. (a) A galaxy siphonophore, (b) the same siphonophore with tentacles retracted, (c) a manefish, (d) an oarfish, (e) a pair of Paralepididae fishes, (f) a pair of Giganturidae fishes, (g–h) a fish

Thalassobathia pelagica, (i) a cusk eel, (j–k) yellowfin tunas, (h) an ocean sunfish, (m) a sperm whale, (n) siphonophore, and (o) deepstaria reticulum (Macreadie et al. 2018)

different purposes and the important tasks are listed as follows:

- **Site survey**
Observation-class ROVs are used to map the seabed before carrying out the activities of installation and drilling.
- **Drilling assistance**
Once oil & gas reserves are discovered, mobile drilling platforms such as drill-ship, jackup rig and submersible drilling rig are used to drill the exploratory well at reserve sites. Work-class ROVs are used to assist the drilling; the activities may include collecting fluid samples from seabed for analysis, retrieving a drilling riser connected to a blowout-preventer located on a sea bed, and real time drilling data interpretation for multiple holes in a single dive.
- **Inspection, maintenance, and repair**
Metallic offshore structures are naturally facing difficult environmental conditions (i.e., salty water and strong sea tides), inspection work of flaws, corrosion and related cleaning, grinding, installing, trenching, lifting, pulling, bolt handing, and equipment transport can be performed by work-class ROVs (Baker and Descamps 1999).
- **Mobile inspection**
Apart from submerged structures, the equipment on the sea surface (i.e., mobile floating platforms, permanent platforms, and ships) also need routine inspection by the observation-class ROVs to improve safety and efficiency of overall production facility. The specific works include inspecting the gauge reading, leakage and surface condition, etc.
- **Oil spill situations**
Oil spill can occur at any stage of petroleum industry (i.e., exploration, drilling, production, transportation, refining, and finally distribution) (Fingas 2012), which is one of the major crises for the oil & gas industry. It not only results in huge revenues losses but also severely damages ecological system of marine life. During these accidents, the ROVs with remote sensing, laser fluorosensors, microwave sensors, and other techniques can be

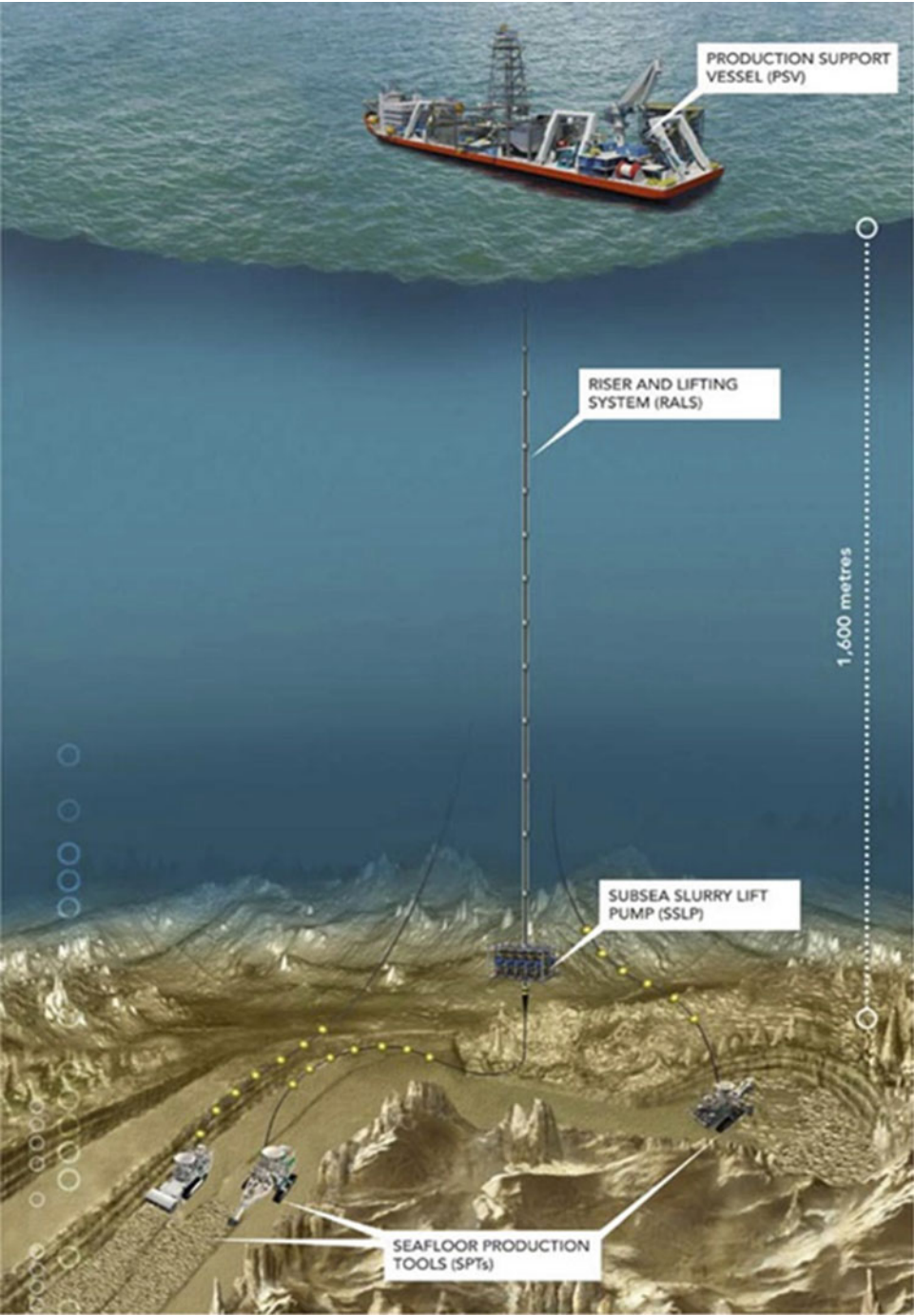
used for many activities such as regular surveillance, detection of spill, and helping in cleaning up operations after spill.

Deep Sea Mining

The deep sea mining has attracted more and more interests in the past decade, as the seabed is known to contain huge quantities of minerals. Recent discoveries have well illustrated that the rare earth deposits lied in the Pacific Ocean are estimated to be 80–100 billion metric tons. Take American Neptune Minerals for example, the company was established in 2011, which seeks to explore and mine the seafloor massive supplied deposit around volcanic arcs (Bogue 2015). It is known that exhaled supplied-rich mineralized fluids from the hydrothermal vents are often rich in gold, silver, copper, zinc, and lead around the volcano arc. Three unique seafloor production tools of an auxiliary cutter, a bulk cutter, and a collecting machine will be used to mine these massive deposits up to 1000 m. The auxiliary cutter and bulk cutter are used to flatten the rugged terrain close to the volcanic chimneys. Then the collecting machine sucks in the cut material, with the mine deposit harvested and pumped to a ship on the sea surface. A deep sea mining system schematic is shown in Fig. 6. The key components are the bulk cutter, a massive, tracked robotic device (Fig. 7) equipped with 4-m-wide cutting blades and driven remotely by two pilots in a control room on a surface vessel. Undoubtedly, they are also special kinds of ROVs.

Military

ROVs have been deployed by the navy more than half century, and a famous example as mentioned above was that the USA Navy's CURV II salvaged a nuclear bomb at the water depth of 869 m offshore the Spanish sea (Christ 2013). ROVs can help the navy to hunt and break the mines. Small-sized ROVs can also be used by the navy for coast guards and port authorities, and have been widely deployed for a variety of underwater inspection tasks, such as explosive ordnance disposal, meteorology, port security, mine countermeasures, maritime intelligence, surveillance, and reconnaissance.

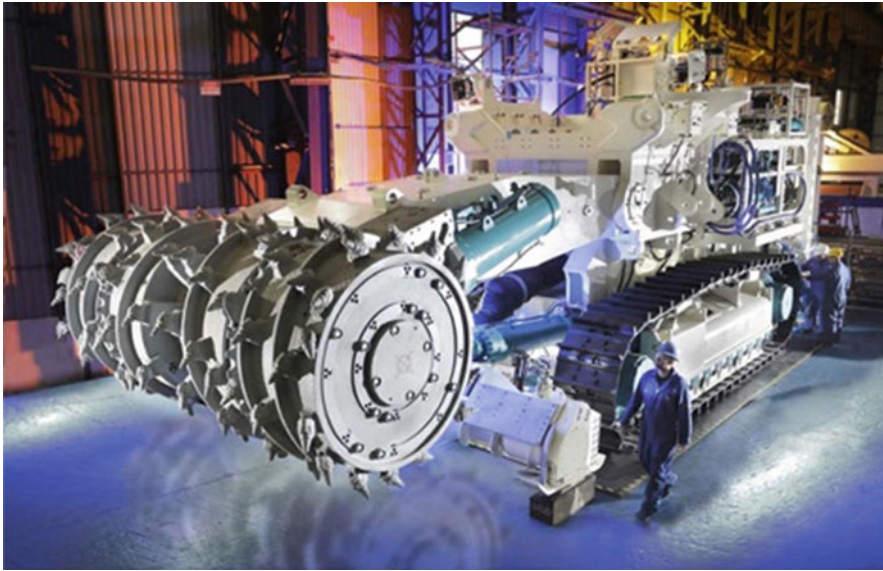


Remotely Operated Vehicle (ROV), Fig. 6 Deep sea mining system (Bogue 2015)

Underwater Broadcast and Photography

Note that the time-duration for ROVs working in the dangerous and complex sea circumstances is unlimited. If evolved with HD cameras, the

observation-class ROVs can help the filmmakers to capture the footage (The Dark Secrets of the Lusitania 2016). A famous example for using ROVs to film the documentaries is the BBC



Remotely Operated Vehicle (ROV), Fig. 7 Robotic bulk cutter (Bogue 2015)

Wildlife Special Spy in the Huddle. In addition, the scientist or tourists can also use the small-sized and medium-sized ROVs to take photos of marine lives, coral or plants.

Fisheries and Aquaculture

Small-sized and medium-sized ROVs can be used in the fisheries and aquaculture. One of the typical services provided by the ROVs is to inspect the fish cages, the specific works include checking nets for holes, assuring the integrity of moorings for farms, and retrieving dead fish from the cage for health purposes (Christ 2013). In the shallow-water-offshore region, like the Shandong province of China, farmers keep a lot of sea cucumbers. The current is very strong and the water is turbid due to the mixture of sand from the rivers in this region, but this region is quite suitable for the sea cucumbers. There is a limit for divers to collect the sea cucumbers in a day due to the visibility, physical fitness, as well as the safety consideration. Another reason is that it is also very expensive to employ the divers. If evolved with sonars and suction manipulators, the ROV can collect the sea cucumber more efficiently than the divers. If countering the overall cost, the ROV will be cheaper than the divers.

Civil Construction

ROVs used for subsea civil construction are typically higher powered and specification work-class ROV; they are equipped with high pressure 7-function manipulators and tooling package. The specific tasks performed by the ROVs may include moving heavy pieces, guiding large construction pieces into place, laying and burying cables and pipelines, setting and pulling rigging, and setting mattresses, etc. (Christ 2013).

Public Safety

Public safety is clearly the realm of observation-class ROVs equipped with cameras, sonars, and other tooling capabilities. This mission typically involved periodic inspection and sweep of pier recreational water, first responder to perform search and rescue the public need, such as recreational boater drown, drown person in the recreational water, river or sea.

Cross-References

- [Atmospheric Diving Suit \(ADS\)](#)
- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Deep Submergence Rescue Vehicle \(DSRV\)](#)
- [Deep Tow System](#)

- ▶ [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)
- ▶ [Glider](#)
- ▶ [Human Occupied Vehicle \(HOV\)](#)
- ▶ [Hybrid Remotely Operated Vehicle \(HROV\)/Autonomous and Remotely Operated Vehicle \(ARV\)](#)
- ▶ [Remotely Operated Vehicle \(ROV\) in Subsea Engineering](#)
- ▶ [Rescue Bell](#)
- ▶ [Submarine](#)
- ▶ [Submersible](#)
- ▶ [Underwater Lander](#)

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Synonyms

[Autonomous underwater vehicle \(AUV\)](#); [Remotely operated vehicle \(ROV\)](#); [Unmanned underwater vehicle \(UUV\)](#)

Definition

ROV is a kind of underwater robot used for underwater observation, inspection, and construction.

As the land resources are gradually exhausted, the ocean, which could provide the rich resources for human’s sustainable development, has been paid increasingly attention. The twenty-first century is recognized as the ocean century by the world. With the advantages of safety, economy, and high efficiency, underwater vehicles can replace human beings to complete high-intensity and heavy-duty underwater operations in a deep and high-risk environment. Hence, an underwater vehicle has become an indispensable equipment for understanding, developing, and utilizing the ocean. This paper mainly focuses on the classification, application, design, and development technologies of underwater vehicles.

Concept and Classification

Underwater vehicle is a mechanical and electrical device that can freely move in the water, equipped with a vision and perception system. Through the remote control or autonomous operation,

underwater vehicle utilizes manipulator or other tools to replace or assist human beings to complete underwater tasks (Christ and Wernli 2014).

Classification of Underwater Vehicles

Underwater vehicle can be classified into different types as follows:

- *Classified by functions:* According to different purposes, underwater vehicles can be divided into three categories, operation, observation, and measurement. Among them, the operation underwater vehicles with working manipulators are used for underwater operations such as rescue, salvage, cable laying, operation, and maintenance of offshore oil platform and other production systems. The observation underwater vehicles are nearly as same as the operation one, but they are mainly used to measure the parameters of the objects to be investigated. The measurement underwater vehicle uses cameras, video cameras, and sonar to observe the topography of sea floor or to search for underwater objects.
- *Classified by modes of motion:* According to different motion modes, underwater vehicles can be divided into three categories, floating, crawler, and walking. Most of the floating underwater vehicles are designed with zero buoyancy (or a little positive buoyancy), and the three-dimensional space motion underwater is relied on thrusters. Crawler underwater vehicles, which are driven by crawlers, are mostly used in seabed construction. The walking underwater vehicles use the walking mechanism to walk on the sea floor. At present, the walking underwater vehicle is still in the research and development stage, and few application cases are reported.
- *Classified by control mode:* Underwater vehicles fall into two basic categories: remotely operated vehicle (ROV) and autonomous underwater vehicle (AUV). The difference between ROV and AUV reflects in whether the underwater vehicle is tethered or not. ROV connects the mother ship of submersible by an umbilical cable, which not only transmits power downward but also controls signals

(from the mother ship to ROV) and data/images (from ROV to the mother ship) in real-time bidirectional transmission. However, there is no physical connection (no tether) between the AUV and the mother ship. The AUV relies on its own power, navigation and positioning sensing system, and intelligent control decision-making system to achieve autonomous navigation. Compared with ROV, AUV has many advantages, such as wide range of activities, deep diving, little risk of umbilical entanglement, ability of performing tasks in complex structures, no large deck support system required, and low operation and maintenance costs.

Design and Features of AUV and ROV

AUV: AUV is designed to withstand extreme pressure and maintain neutral buoyancy in the deep sea under extreme harsh conditions to reduce the required propulsion energy and improve the efficiency of the vehicle. Therefore, the overall structure of AUV is generally optimized to ensure streamlined shape design of AUV body and to find a balance point between limited volume and system power consumption and battery power consumption, carry sensors according to the design purpose, and maintain the longest endurance capability. In addition, AUV uses modular and standardized design to accomplish many different tasks. Since all mechanical and electronic interfaces inside the body are standard, the selected sensors can be quickly integrated into the load module, and the functions of the new sensors can be integrated into the software of the main system through a software toolkit to synchronize the data flow with the navigation status. As a result, the AUV has stronger endurance and load-carrying capabilities, more automated and safe release, and recovery systems.

AUV has strong endurance and load-carrying capacity, more automatic and safe release, and recovery system. The general basic components of AUV include (1) battery, (2) onboard detection equipment, (3) navigation and positioning system, (4) propulsion system, (5) communication system, (6) safety and security system, (7) release recovery system, and (8) deck workstation.

ROV: ROV system is mainly composed of support mother ship, deck console, deck power supply unit, ROV lifting system, umbilical cord cable, ROV body, robot, and working tools. It is powered and communicated by a surface mother ship through the umbilical cord cable connecting the ROV body; observed and operated by the mother ship staff through underwater video, sonar, and other special equipment; and can also operate underwater by a manipulator. Since it does not endanger personal safety and power is primarily provided by the mother ship, working underwater can theoretically be if possible. The ROV has hydraulic power system, propulsion system, cloud platform system, video and lighting system, navigation and positioning system, power supply and distribution system, and monitoring system. The underwater stability of ROV can be well maintained by hydrodynamic analysis, reasonable arrangement of installation position of each equipment, and adjustment of center of gravity by ballast block (Dan et al. 2010).

At present, the propulsion devices of underwater vehicles mainly include propeller thruster, hydraulic thruster, pump-jet thruster, magnetic fluid thruster, bionic thruster, track thruster, and so on. Propeller is generally used as the propulsion system of underwater vehicles. However, in some special environments, other types of propulsion systems are used, such as mining robots working under the sea, mostly crawler thrusters.

The characteristics of the two underwater vehicles are comprehensively compared as shown in Table 1.

Figure 1 shows two observable ROVs developed by author’s team. Figure 2 shows two AUVs developed by author’s team.

Application Fields of Underwater Vehicles

With the rapid ocean development, there are increasing number of underwater construction

Remotely Operated Vehicle (ROV) in Subsea Engineering, Table 1 The characteristics of AUV and ROV

Type Characteristics	AUV	ROV
Working principles	No umbilical cord cable required Autonomous decision-making	Need umbilical cord cable Human decision-making
Control systems	Auto-control Complex control system	Artificial control Simple control system
Communication systems	Wireless communication Underwater communication is not possible	Wire communication Underwater cable real-time communication
Design methods	Integrated approach Mechanism analysis method.	Integrated approach Direct similarity method



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 1 Two observation ROVs developed by author’s team



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 2 Two AUVs developed by author's team

and building projects. Underwater vehicles can perform observation, camera shooting, measurement, salvaging, and construction operations underwater, and hence they are widely used in ocean development. At present, underwater vehicles are widely applied in many engineering fields, such as underwater engineering, marine scientific investigation, salvaging, marine military, and fishery production.

Applications in Underwater Engineering

- *Underwater structure inspection:* Physical location detection and quality measurement of underwater structures, verification and inspection of laying conditions, damage and corrosion, inspection of drilling platforms, wellheads, dams, cracks, and safety conditions of pollution barriers and gates.
- *Underwater surveillance:* Monitor the operation and construction guidance of grouting, piling, trenching, felling, bulldozing, and other aspects of underwater civil engineering.
- *Submarine topography survey:* Map natural or man-made underwater objectives, seabed topography, and seabed profile.
- *Underwater facility cleaning:* Clean and repaint the surface of underwater facilities, clean oil rigs, remove rust, and paint hulls, pipelines, and underwater components.
- *Underwater target search and recognition:* Search and salvage specified objects underwater, sunken ships, abandoned equipment, equipment, etc.
- *Installation and maintenance of underwater facilities:* Install and maintain underwater pipeline cables, open valve, weld, cut and detonate, replace the sacrificial anode, etc.
- *Assist diver operation:* Assist diver operations, ensure the safety of divers, and transfer construction equipment to divers.

Applications in Marine Scientific Expeditions

- *Marine geological survey:* Record the microtopography of the ocean floor, draw a map of the sea floor, and collect sea floor patterns and rock samples. A 3500-meter ROV independently developed by China has observed a rare huge black chimney in the seamount region of the Eastern Pacific Ocean. During operation, a hydrothermal black chimney sample was obtained by a manipulator of ROV, which provides valuable information for ocean research.
- *Marine life expedition:* Measure the biological form on the seabed and collect biological samples.
- *Marine physics expedition:* Observe seawater aquifers, circulating water velocity, salinity, temperature, depth distribution, seawater specific gravity, etc.
- *Geophysical expedition:* Measure the earth magnetic field and investigate oil and gas deposits.
- *Marine acoustic survey:* Observe underwater acoustic characteristics, underwater reverberation, and acoustic models.

- *Geochemical expedition*: Observe the water temperature of the sea floor, the soil temperature of the sediment layer, and the pH value of the sediment.
- *Marine optics expedition*: Observe water transparency, natural light field distribution, light absorption intensity, etc.

Marine Military Applications

Military underwater vehicles may play a significant role in the naval battle, which may induce a new prospect in the future. Military underwater vehicles can perform a variety of tasks in maritime military confrontations, such as coastal reconnaissance, demining, anti-submarine, communications, and navigation. They make a possibility to quickly seize a military victory with minimal human casualties in naval battles. The functions of military underwater vehicles are listed as follows.

- *Underwater demining*: Underwater vehicle is an optimal equipment to perform anti-deminig operations. Compared with other anti-deminig devices, underwater vehicle has the advantages of low cost and good concealment.
- *Anti-submarine warfare*: Underwater vehicle can search and track submarines in coastal waters, narrow waterways, and straits. It has a better ability to search for submarines than surface ships and submarines. Underwater vehicles are also utilized to set traps for enemy ships in anti-submarine warfare (sonar confrontation, misleading direction).

ROV System Design

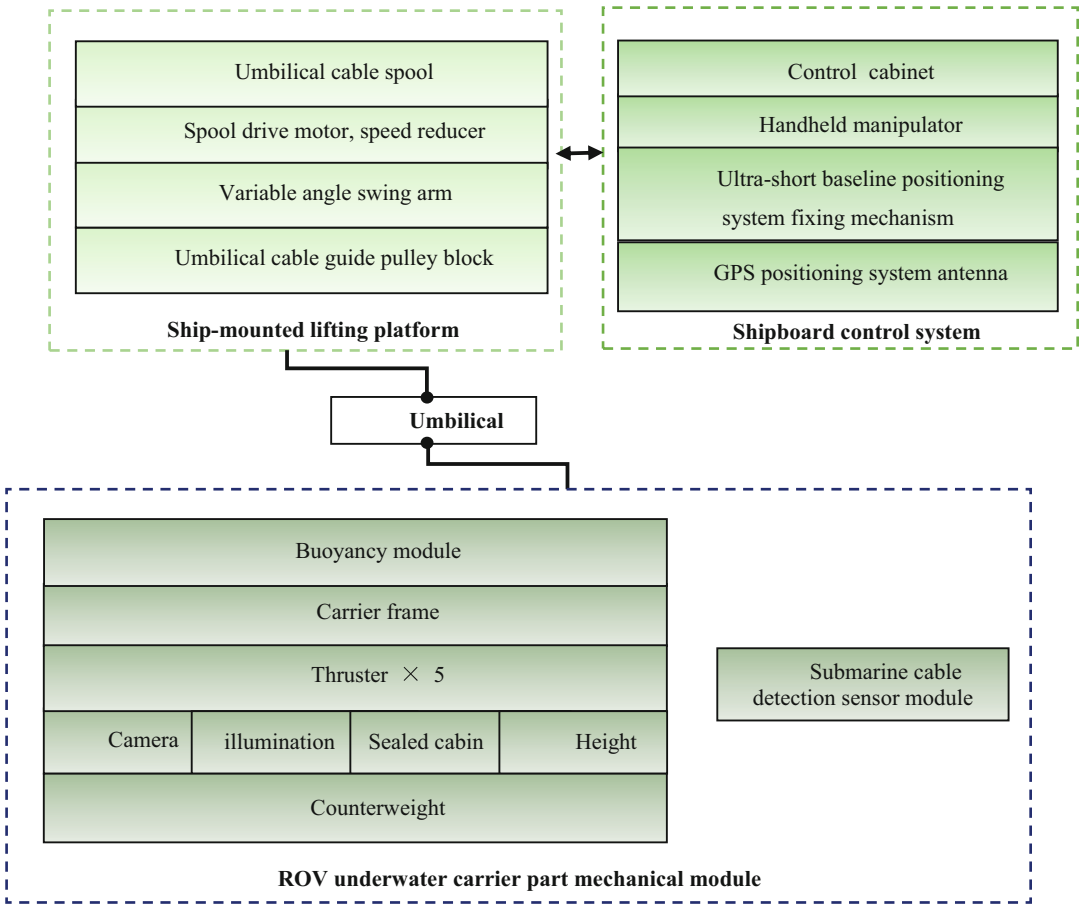
ROV System

ROV, a kind of unmanned underwater vehicle with cable remote control, generally consists of a deck unit (i.e., a control system), a winch retractable system, and a body. Its work flow is as follows: the operator sends orders through the control system, and the commands are transmitted to ROV through an umbilical cable. ROV makes corresponding actions according to the command.

Data collected by the camera and various sensors are also transmitted to the deck unit through an umbilical cable. The umbilical cable, which is a zero-buoyancy cable with certain strength, can guarantee the normal deployment and recovery of ROV. The specific structure of the system is shown in Fig. 3.

The ROV system is composed of seven main parts: main frame, floating material, control communication system, power propulsion system, observation and lighting system, manipulator system, and sensor detection system (Guan 2018).

- *Main frame*: The main frame is the carrier of the entire ROV system and its reasonable structural design based on the overall size of the ROV. The main frame should be firm and stable.
- *Floating material*: The common floating materials utilized in underwater vehicles are polymer-based solid material, which is characterized by low density (0.2 to 0.7 g/cm^3), low water absorption (less than 3%), high mechanical strength (e.g., compression strength is in the range of 1 to 100 MPa), corrosion resistance, and machinability. Polymer-based solid material could meet various requirements in the underwater environment. It is noted that floating body materials should be selected according to the maximum water depth.
- *Control communication system*: The control communication system is a comprehensive system, which includes collecting, transmitting, processing, and analyzing various information and commands of the ROV. The control system transmits real-time control commands to the ROV through the umbilical cable, and thus the ROV is controlled to perform some specified work tasks. At the same time, the information such as ROV's depth, attitude, and image is transmitted to the deck unit in real time through the umbilical cable for analysis and processing.
- *Power propulsion system*: The power propulsion system mainly includes two parts, the thruster and the power supply. The power propulsion system supports the mother ship deck to provide power to the ROV through



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 3 Block diagram of ROV system

- the umbilical cable and drives the thruster to achieve the command-controlled ROV movement.
- *Observation and lighting system:* The observation and lighting system are mainly composed of cameras, sonars, and LED lights. ROV can obtain underwater image through the observation and lighting system, which contributes to ROV in avoiding obstacles, improving the efficiency and safety during operation.
 - *Manipulator system:* The manipulator system is the key symbol of ROV’s operating capability, which is an indispensable part in an operating-oriented ROV. The manipulator system is mainly composed of a manipulator, a hydraulic pump, and a control valve. The manipulator has lots of basic parameters, among which the degrees of freedom,

movement space, load capacity, and selection of materials are important parameters for its design. During operating, the operator sends control signals from the control box on the deck of the mother ship, and the control signals are transmitted to the underwater control system through the umbilical cable, which controls the manipulator to complete certain tasks.

- *Sensor detection system:* Sensor detection systems mainly include altimeters, pressure sensors, attitude sensors, temperature sensors, ultrashort baselines, and cable detection sensors. Various underwater detection information collected by the sensors is transmitted to the deck unit through the umbilical cable. Based on the information, the operator analyzes and judges the status of the ROV, which is beneficial to achieve the ROV operation control.

ROV System Design

Overall Structural Design

In designing ROV, the engineers should make clear the design goal, the operation task, the working mode, and the working environment. The mature ROV technology in worldwide can be referenced in the design of overall structure framework. The ROV body is discretized into several blocks for modular design. The connection among different modules should be concerned during module design, in order to maintain the integration of each module. Some aspects should be carefully considered in the over structural design of ROV.

- Compact layout. The limited space in the ROV should be fully used without interference among each part of the ROV. The overall size of the ROV should be optimized for convenient transport and storage.
- The technical performance and requirements for the working environment of ROV should be fully taken into consideration, which ensures the normal use of the equipment and avoid interference with each other.
- Without interfere normal function, the design of the components and the connection among the modules should be simplified in maximum extent, which is beneficial to facilitate the maintenance of the ROV.
- Under full consideration of operational requirements, sufficient spare space should be

reserved for ROV system updating and modifications.

- The durability, reliability, and stability of control systems and mechanical components in ROV must be ensured.

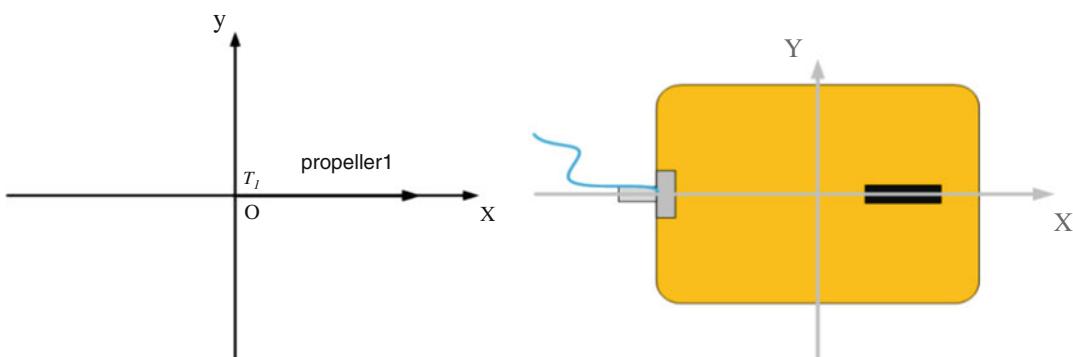
Design of ROV Propulsion System

The thruster is the main power source for the ROV. There are many kinds of thrusters, among which the propeller thrusters are widest utilized. The number of degrees of freedom of ROV in underwater movement is determined by the operation purpose and design requirement. Three degrees of freedom, i.e., forward and backward, up and down, and steering, are the basic mobility requirements of a ROV. The number of thrusters required for ROV is determined by the purpose and design requirements.

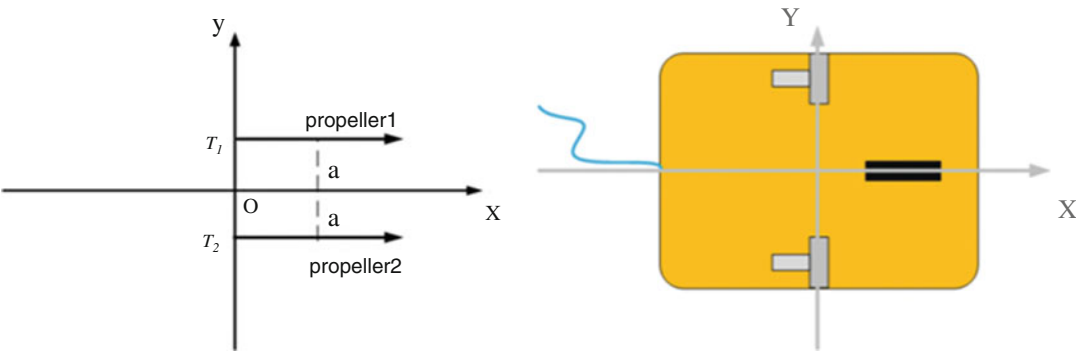
At present, the layout of thrusters can be divided into five types: single thruster layout, parallel layout of double thrusters, cross layout of double thrusters, annular layout of four thrusters, and conical layout, as shown in Figs. 4, 5, 6, 7, and 8.

For arranging thrusters, two points should be paid attention:

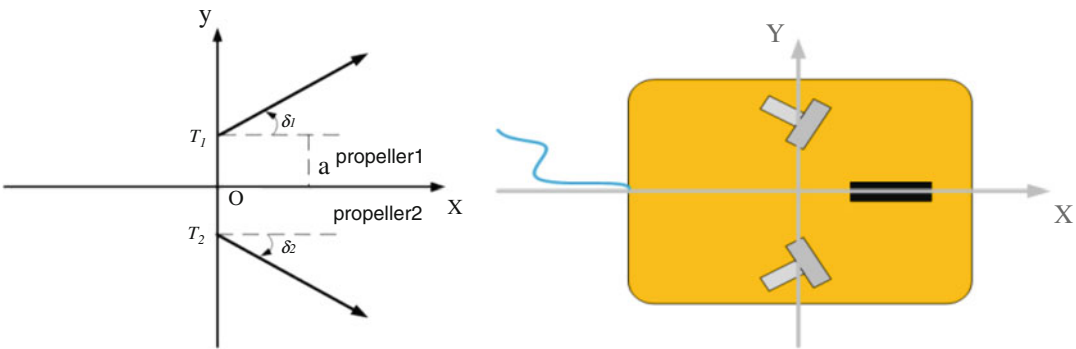
- The axis should be parallel to the moving coordinate system, which maximizes the efficiency of the thruster.
- The axis of three-axis thruster should be reasonably converged to one point. This point



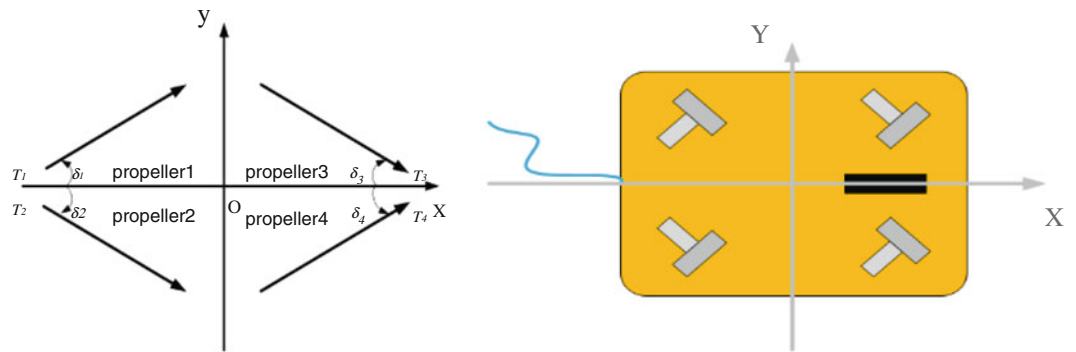
Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 4 Single-pusher arrangement



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 5 Parallel arrangement of dual thrusters



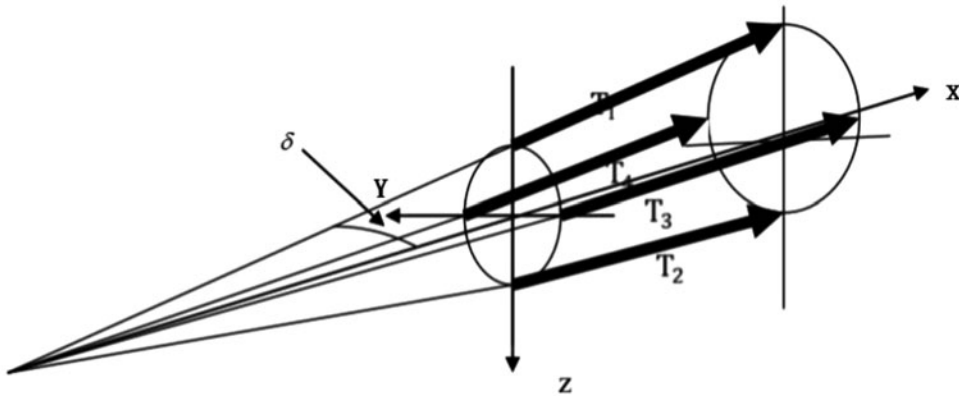
Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 6 Double-thruster cross arrangement



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 7 Ring arrangement of four pushers

should approach to the center of gravity of ROV in order to avoid some harmful movements of ROV and benefit to the system control.

Overall Balance Design
Due to many equipment on ROV, the weight symmetry of ROV should be taken into consideration, in order to maintain the gravity position and



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 8 Cone arrangement of thrusters

the metacentric position. In practical, it is very difficult to achieve the absolute balance of ROV. Therefore, the arrangement of ROV's internal equipment should follow some principles:

- The primary body should be firstly considered and then other parts.
- The global layout should be firstly considered and then local arrangement.
- Some certain amount of space should be left, which leaves the room for modification.

In order to ensure the stable motion of ROV, a certain stable height should be guaranteed. The metacentric height of general underwater vehicles should be greater than 7 cm, and the metacentric height of large underwater vehicles should be correspondingly increased. In order to determine the metacentric height, the coordinates of center of gravity and center of buoyancy need to be calculated according to the overall arrangement of ROV. Besides, the coordinates of center of gravity and center of buoyancy of ROV can also be calculated in 3D drawing software. It is necessary to ensure that the line between the barycentric coordinates and the center of buoyancy coordinates passes vertically through the underwater vehicle so that no overturning moment occurs when it is stationary. Otherwise, the general layout of ROV should be adjusted so that the two coordinate positions are equal as far as possible; even if they are not absolutely equal, the angle formed between the two

coordinate lines and the vertical line should be kept within the range of 0 to 15°.

Dynamic Analysis of ROV

For studying the motion law of ROV, its kinematic and dynamic modeling should be developed. Kinematics describes the relationship between location and time, while dynamics focus on the relationship between force and motion (Liu et al. 2015).

Force Analysis of ROV in Water

ROV is mainly affected by buoyancy, gravity, thrust of thrusters, hydrodynamic and umbilical cable interference forces, and the torque generated by these forces during underwater movement. The combined force and moment formed by these forces and moments make the ROV produce six degrees of freedom in space motion.

Gravity and Buoyancy: The buoyancy and gravity of ROV are always in the vertical direction. Gravity is represented by P , and buoyancy is represented by B . The gravity center is represented by G , and floating center is represented by C . It is assumed that the coordinates of the center of gravity and the floating center in the x and y directions are coincident (which can be achieved by balancing the weight and buoyancy), and the floating center is higher than the center of gravity in terms of h . Without considering the changes of buoyancy and load, the components of gravity, buoyancy, and moments

on the axes of the ground coordinate system can be calculated by corresponding formulas.

Thrust of the Thruster: The thrust generated by the propeller is the power that enables the ROVs to move in their respective degrees of freedom. The thrust generated by the propeller can be calculated from the propeller thrust formula.

Water Resistance: In underwater environment, the resistance of ROV mainly consists of two parts: the hydrodynamic force and the interference of the umbilical cable. The calculation of hydrodynamics is complex. There are around 100 parameters relative to hydrodynamic coefficient. Generally, the hydrodynamics is related to the geometry, motion state, and flow field properties of a ROV (Liu et al. 2020).

The calculation of the umbilical cable's interference force is also complex. Generally, it can be calculated by the concentrated mass method, that is, the umbilical cable is regarded as a chain model. In a chain model, the umbilical cable is assumed as a chain including a series of cylinders with finite length and concentrated mass at one point. According to the movement status of the mother ship and ROV, mechanical analysis and calculation are performed on each section of umbilical cable, and then the interference force of umbilical cable on ROV can be obtained.

In practical, an approximate estimation method can generally be used to calculate the resistance force of ROV in underwater environment. The resistance to which it is subjected can be estimated by the resistance formula (Fan et al. 2012).

ROV Dynamic Modeling and Analysis

According to rigid body dynamics theory and Newton-Euler equation, the six-degrees-of-freedom equation of ROV is obtained, and then the dynamic equation is established by combining the four forces of gravity, buoyancy, propeller thrust, and water resistance (Nakamura et al. 2013). The motion posture of ROV corresponding to the next moment can be calculated from the motion posture and external force of the previous moment, which is an indispensable basis for ROV motion control.

In order to verify the dynamic effect of ROV, CFD simulation method is often used (Yan et al.

2010). This method uses computer as the tool, uses discrete mathematical method, and uses simulation software to simulate and analyze the mathematical model.

ROV Control System

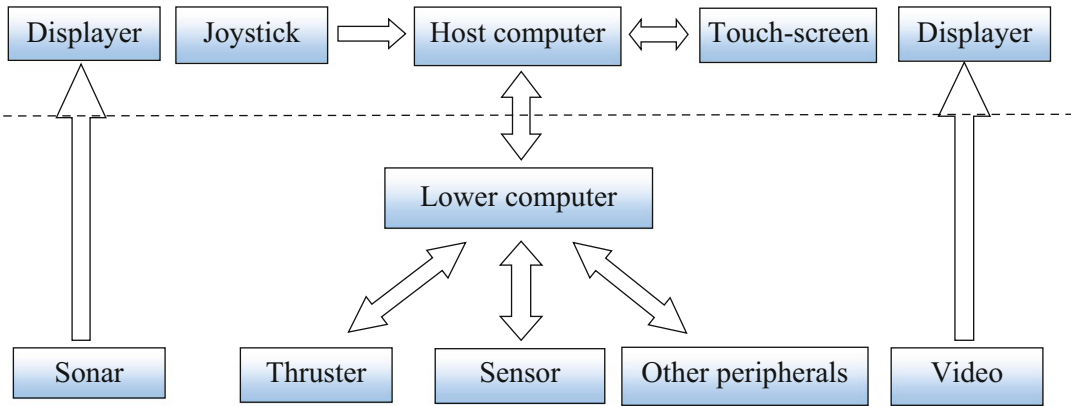
The control system is the core of the ROV, which mainly includes the hardware system and the software system. The hardware system is divided into two parts, the surface control system and the underwater control system. The software system is divided into two parts, an operation part and a monitoring part. The surface control system is responsible for collecting ROV's control instructions, including digital quantities and analog quantities. The underwater control system is responsible for responding to the control instructions of the surface control system and also feeding back the corresponding status and data in real time. The software system is responsible for sending operation instructions and receiving and processing data. The cooperation of the hardware system and the software system can ensure the normal operation of the ROV. The overall block diagram of the ROV control system is shown in Fig. 9.

ROV Control Hardware System

The overall system of ROV is complex and it involves many control targets. A large number of sensors and equipment are involved in its control system. Due to the difference among electrical interfaces and communication protocols, there still are different communication rates and specifications, even though the same the electrical standards are utilized. Hence, the integration of various interface standards and flexible configuration modes are compulsory in the design of the ROV's control system. Moreover, some certain degree of redundancy should be considered for later maintenance, expansion, and upgrade.

Common ROV underwater main control units are as follows:

- Based on the main control method of PC104. RS232, RS485, AI (analog input), DI (digital



Remotely Operated Vehicle (ROV) in Subsea Engineering, Fig. 9 ROV overall control system block diagram

input), Ethernet and other modules are configured on the interface, which are operated by Windows or Linux system.

- Based on embedded systems of single-chip microcomputers. RAM-based embedded hardware is commonly used, which is operated by Linux, uCOSII, and other operating systems, or without an operating system directly.
- Based on the PLC of the industrial automation industry, various common interface modules can be configured according to the ROV. Depending on different PLCs, the manufacturer can develop the operating system or use the general operating system.

ROV Control Software System

The ROV control software system is the core part of the ROV control system. The ROV control software system contains an operation part and a monitoring part. The operation part is the fundamental software, which is responsible for identifying the information of the joystick and operation buttons and communicating with the ROV in real time. Its software interface mainly completes the following functions:

- Configure the initialization information of the ROV software.
- Configure the initialization information of each sensor, such as communication mode, communication rate and so on.
- Control the enable status of the thruster and monitor the thruster speed.

- Feedback the current speed of ROV's movement in real time.
- Feedback the current depth of the ROV in real time, and estimate the ROV dive speed and dive time.
- Feedback the current ROV off-bottom height in real time, combined with the depth parameter of the ROV, and control the ROV movement speed.
- Real-time feedback of ROV electronic cabin temperature and monitoring of the operating system temperature of the electronic system.
- Real-time feedback of ROV water leakage detection status and monitoring the pressure-tight status and waterproof status of the electronic cabin.
- Feedback the status of underwater lights and cameras in real time, and check whether they are turned on correctly.
- Feedback the operating status of the manipulator in real time, and compare it with video monitoring to ensure the correct operation of the manipulator.
- Detect the hydraulic pump pressure in real time to ensure the correct operation of the manipulator, and eliminate errors.
- Configure an appropriate debugging window to detect PLC and other internal variables, and troubleshoot the failure rapidly.

ROV Motion Control Method

The main parameters of ROV motion control are depth (the vertical distance from the sea surface to

the center of gravity of the underwater vehicle), height (the vertical distance from the seafloor to the center of gravity of the underwater vehicle), heading angle, speed, position, and so on. In most ROV designs, in order to obtain good control performances without complicating the system, single-loop closed-loop control is used for these parameters, such as depth loop (auto-depth control), altitude loop (auto-height control), heading angle loop (auto-heading control), speed loop (constant speed control), position loop (automatic positioning), etc.

On the basis of the basic circuit of the motion control system, a certain control algorithm is also needed to achieve accurate control of the motion of the underwater vehicle. The common control algorithms include PID control, fuzzy control, adaptive control, and neural network control.

Sensing and Communication System in ROV

Common Sensor Used in ROV

There are three main sensors used in ROVs, navigation and positioning sensors, operating system sensors, and condition monitoring sensors.

Navigation and positioning sensors: Navigation and positioning system sensors include depth sensors, gyroscopes, Doppler Velocity Logs, acoustic positioning systems, inertial navigation systems, imaging sonars, etc. The function of navigation and positioning system sensors is collecting information on the position, attitude, and speed of the underwater vehicle in the water and detecting the vehicle's surroundings. The specific functions of each sensor are described as follows:

- Depth sensor: Measure the water depth in respect to vehicle's position.
- Altimeter: Measure the distance between vehicle and sea floor.
- Gyroscope: Measure the attitude of the vehicle in the water.
- Doppler log (DVL): The Doppler log is used to determine the speed of an underwater vehicle relative to the ocean floor.

- Acoustic positioning system: Acoustic positioning system is a technology that uses underwater acoustic equipment to determine the position and distance of the vehicle relative to the mother ship.
- Inertial navigation system: The inertial navigation system consists of a gyroscope and an accelerometer. By measuring and integrating the ROV's angular velocity and acceleration, the ROV's speed, yaw angle, and position in the navigation coordinate system can be obtained.
- Imaging sonar: Imaging sonar is an acoustic device whose principle is similarly to a radar. The sonar transducer, equivalent to a radar antenna, can emit sound waves and receive sound waves reflected from the surrounding environment and then produce acoustic images of the surrounding environment.

Operating system sensors: Operating system sensors refer to the various sensors that ROVs carry when operating underwater. Different operating systems use different sensors, including electromagnetic detection sensors and acoustic detection sensors.

Condition monitoring sensor: To ensure the safety of the ROV system, some sensors are needed to monitor the vehicle. For example, the battery voltage and power detection sensors can reflect the battery usage; the water leakage detection sensors in the sealed cabin can detect the sealing performance of the sealed cabin in real time.

Communication Technology

The core of the ROV communication system is data exchange between the aquatic equipment and the underwater equipment through a cable. The common methods are optical fiber communication and power line communication.

Optical fiber communication: A communication method that uses light waves as a carrier wave and uses optical fiber as a transmission medium to transfer information from one place to another is called *wired* optical communication. At present, optical fiber has been widely used in the communication system of underwater vehicles due to its

transmission frequency bandwidth, long transmission distance, high interference resistance, and small signal attenuation.

The principle of ROV optical fiber communication is as follows: the transmitted information (such as sensor data and video shot by an underwater camera) is firstly converted into an electrical signal at the transmitting terminal. Then it is modulated onto the laser beam emitted by the laser, so that the intensity of the light varies with the amplitude (frequency) of electrical signal. By transmitted through the optical fiber, the detector converts the optical signal into an electrical signal at the receiving end and restores the original information display after demodulation.

Power line communication: The use of power carrier communication in the ROV communication can combine the power supply line and the signal transmission line, which can reduce the number of cores in the zero-buoyancy cable. The principle of the power carrier in ROV is the following: the transmitted information is modulated onto the power line through the carrier module. After the power line transmission, it is demodulated on the carrier module at the other end to obtain the transmitted information.

Acoustic communication: Affected by seawater, electromagnetic waves diminish rapidly, and they cannot be used as a technical method for long-distance communication, while light waves are affected by the turbidity, absorption, and scattering in the water (Yong and Weigang 2019). Relying on current technology, only stable connections can be achieved over short distances. The propagation speed is fast; although it is affected by the noise, it can realize long-distance communication through filtering. The acoustic communication has stable and reliable applications on the *Jiao Long* and *Deep Sea Warrior* deep submersibles. It is assumed as one of the most promising directions in the development of acoustic communication in the future.

Conclusion and Prospect

Most underwater vehicles now have a certain degree of motion flexibility and superior navigation,

imaging, and sensing capabilities; however, they are still not reliable enough to integrate with marine life and to deal with unknown problems. Some challenges are still faced, including underwater communication, positioning measurement, mid-deep navigation, and unpredictable interference. With the continuous expansion of the application field, the underwater vehicle will develop toward smaller size, more compatibility, and higher intelligence, in order to break through the obstacles in the design of unmanned underwater vehicle, which may greatly improve its automation; in addition, some new technologies (e.g., multimedia technology and virtual reality technology) should be applied to underwater vehicles for future development.

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Remotely Operated Vehicles (ROVs)

► Hydrodynamics for Subsea Systems

Removal of Fixed Platform

► Decommissioning of Fixed Platform

Renewable Energy Propulsions

► Ship Construction and Operation Impact on Energy Efficiency

Rescue Bell

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Synonyms

Rescue chamber; Submarine rescue bell

Definition

Rescue Bell is one type of submarine rescue system, which can carry submariners from a disabled submarine to the surface. A typical Rescue Bell consists of four major components: a pressure hull, mating skirt, surface support system, and life support system. Some rescue bells also have propellers, imaging system.

Scientific Fundamentals

Historical Development

During the first two decades of last century, in the United States Navy Submarine Force, there were several accidents in which Navy submarines sank with the loss of life. These experiences led submariner Charles B. Swede Momsen to think of technical alternatives for rescuing survivors

from sunken submarines, which at that time was still a virtual impossibility. Momsen soon conceived a submarine rescue chamber that could be lowered from the surface to mate with a submarine's escape hatch and proposed the concept through official channels. While in command of the submarine S-1 (SS-105), in 1926, Momsen wrote to the Bureau of Construction and Repair (BuC&R) recommending the adoption of a diving bell for the purpose of rescuing entrapped personnel from submarines. But this idea was pigeonholed by the bureaucracy, even during his own subsequent assignment at BuC&R. The loss of S-4 with all hands put the Navy very much "on the spot" because of the loss of lives that might have been saved. The pressure of this incident forced favorable action and Momsen, using the aircraft hangar from S-1, designed and built a prototype submarine rescue chamber (Fig. 1).



Rescue Bell, Fig. 1 Momsen-McCann rescue bell prototype

During the first 3 months of 1928, divers and other salvage personnel were able to raise S-4 and tow her to the Boston Navy Yard, where she was drydocked and repaired. She returned to active duty in October 1928 and was employed thereafter as a submarine rescue and salvage test ship. Momsen went to sea in the reconditioned S-4 to carry out practical experiments and training with the rescue chambers.

Work with S-4 helped to develop equipment and techniques that bore fruit a decade later, when 33 men were brought up alive from the sunken submarine *Squalus*. The first diving bells for rescuing men from submarines were designed by the BuC&R in 1928. The diving bell went through a series of tests off the shores of Key West, Florida. Based on these tests, Momsen had several changes in mind for the bell, and after nearly 2 years of experimentation full of highly interesting results, the final bell was evolved and christened a "rescue chamber." This success was catalyst for gaining approval for the development of the submarine rescue chamber in 1930.

Lieutenant Commander Allan Rockwell McCann was put in charge of the revisions on the diving bell. From July 1929 to July 1931, McCann was assigned to the Maintenance Division, Bureau of Construction and Repair, where he developed the submarine rescue chamber. When the bell was completed in late 1930, it was produced as the McCann Submarine Rescue Chamber (SRC) (Navy designated the first 12 of these as YRC 1-12, YRC-4 was lost aboard the USS *Pigeon*, at Bataan, Philippines during the first days of WWII. YRC-5 was aboard the USS *Widgeon* (AM-22) during the Pearl Harbor attack). In 1931, a one-fifth scale model of a diving bell for submarine rescue work was built and tested. Design called for the bell to withstand the external pressure encountered at a depth of at least 300 ft (91 m) of water, and the test showed the model fulfilled this requirement with a factor of safety of about 3.5. The vessel was tested under external pressure, failure occurring in the shell at a pressure of 470 psi (3,200 kPa). Since the head of the vessel remained intact, it was decided to make a test of the head itself in order to determine its strength relative to that

of the shell, and if possible to obtain some measure of the stresses occurring under load. The head collapsed at a pressure of 525 psi (3,620 kPa), indicating its strength under external pressure was about 10% in excess of that of the shell.

The revised Submarine Rescue Chamber had improvements including a soft seal gasket for sealing the submarine/bell interface skirting, and a floor installed to maintain air-space in the bell during raising and lowering. Momsen in his speech to the Harvard Engineering Society on 6 October 1939 credits Allan Rockwell McCann with the improvements which made the bell operational, safe, and large enough to hold up to eight rescued crewmen and two operators.

After the development of the rescue bell, the US Navy was equipped with dozens of rescue bell. After 1970s, the Deep Submergence Rescue Vehicle (DSRV) replaced the rescue bell.

In recent years, China continues to develop the modern rescue bell. China has developed a type of movable rescue bell in 1990s successfully. This type of rescue bell has four thrusters. This rescue bell has better mating ability than McCann rescue bell. The working depth of the movable rescue bell is 200 m. The bell has umbilical line for power supply, with a rotary skirt. The mating angle can reach 20°. McCann rescue bell is a simple rescue chamber. Chinese rescue bell has the ability of manned occupied submersible, but the rescue principle is the same as McCann rescue chamber (Fig. 2).

In 2008, China has successfully developed a new type of rescue bell, Maneuvering Rescue Bell. The maneuvering rescue bell has a 45° rotary skirt. The rescue bell can be carried by truck, train, or airplane. The high-pressure seawater pump can drain off the skirt water quickly. The energy supply comes from the mother ship, and the rescue bell can work continuously (Fig. 3).

Key Technology in the Development of a Rescue Bell

In the development of the modern rescue bell, mating skirt and life support system are the key technology. Most equipments are the same as human occupied vehicle and ROV. For the mating skirt, big mating angle, light weight, and reliable

Rescue Bell, Fig. 2 China
movable rescue bell



Rescue Bell, Fig. 3 China
maneuvering rescue bell



R

sealing are the key technical specifications. For the life support system, rescue with pressure is a complicated rescue condition. Under this condition, mating and transfer are very difficult. And the life support system can support the rescuer under high air pressured environment.

The McCann bell suffers from severe limitations in strong currents and when dealing with a pressurized submarine or one lying at extreme

angles. The Rescue Chamber is air transportable to a Vessel of Opportunity (VOO) Mother Ship (MOSHIP), which requires little modification to use the system. Transfer Under Pressure (TUP) to and from pressurized environments such as submarines or hyperbaric chambers is not possible with this system, even though TUP is essential where being subjected to ambient pressure may be life-threatening.

The modern rescue bell such as the movable rescue bell and the Maneuvering Rescue Bell can deal with 2.5 knots currents. The extreme angle can reach 45°. The modern rescue bell can mate and rescue at pressurized environments under 50 bars.

Key Applications

In 1939, the McCann Rescue Chamber made its debut when it was used to successfully rescue 33 survivors from Squalus. At the time of Squalus' accident, Lieutenant Commander Momsen was serving as head of the Experimental Diving Unit at the Washington Navy Yard. The submarine rescue ship USS Falcon (ASR-2), commanded by Lieutenant George A. Sharp, was on site within 24 h. It lowered the Rescue Chamber, a revised version of a diving bell invented by Momsen and, in four dives over the next 13 h, recovered all 33 survivors in the first deep submarine rescue ever. McCann was in charge of Chamber operations, Momsen commanding the divers. Although there was no reason to believe anyone was alive in the aft part of the ship, a fifth dive was made to the aft torpedo room hatch on May 25. This run confirmed the flooding of the entire aft portion of the ship.

Although there are no other actual rescue cases, the Chinese navy carries out rescue training every year. The training includes mating and rescue submariner testing through actual submarine. Due to the low cost and simple rescue procedures, the rescue bell is still a simple and efficient submarine rescue equipment.

Cross-References

► [Human Occupied Vehicle \(HOV\)](#)

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Rescue Chamber

► [Rescue Bell](#)

Research Ship

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Synonyms

[Research vessel](#); [Survey vessel](#)

Definition

A research ship is a special ship or boat designed, modified, and equipped to carry out research at sea, which is carrying scientists and special equipment. Research ships are applied in marine natural science research such as geology, geophysics,

hydrology, meteorology, chemistry, biology, landforms, and so on.

Scientific Fundamentals

Development History

On the basis of great technological changes and typical characteristics of ship types, there are two main periods in the development history of the world research ships (Wu 2017).

First Development Period

The first development period of research ships is from the late 1950s to 1980s. Along with the application of electronic computers and the emergence of various kinds of advanced marine survey equipment, modern research ships are built gradually. Compared with the early refitted research ships, research ships in the first generation have qualitative improvements in performance, laboratories and special equipment, etc. (Griffin and Sharkey 1987).

There are a series of research ships that are built in the first development period, such as the 6000 t comprehensive research ship “Romenosov” built in the Soviet Union in 1959, the 3400 t hydrographic research ship “Surveyor” built in the United States in 1960, the research ship “ATLANTIS II” built in the United States in 1962, the 3100 t research ship “DISCOVERY” built in the British in 1962, the 4700 t reconstructive research ship “SONNE” built in Germany in 1977, etc.

The first reconstructively comprehensive research ship “Venus” is the pioneer of marine research ships in China in 1956 (Zhang 1998). Since that, there are typically representative research ships, such as the first meteorological research ship “Qixiang 1” designed and built independently in 1959, the comprehensive research ship “Dongfanghong” built in 1965, the 3000 t comprehensive research ship “Shijian” built in 1969, the 4400 t comprehensive research ship “Xiangyanghong 09” built in 1978, the ton class comprehensive research ship “Xiangyanghong 10” built in 1979, the 3000 t geological research ship “Haiyangsihao” built in 1980, the 3300 t

comprehensive research ships “Kexue 1” and “Shiyan 3” built in 1980, etc.

There are a few function shortages of research ships in this period (Li 1982). Loud noise and big vibration can’t meet the demand of acoustic experiment research. Survey equipment and laboratories are outmoded and defective. Besides, outdated detection method and small detecting depth and range are difficult to realize multidisciplinary research. And data transmission and processing are unable to complete in the field. What’s more, high speed results in wasting energy.

Second Development Period

The second development period of research ships is since the late 1980s. As the gradual improvement of the electric propulsion system and the dynamic positioning system, as well as a variety of upgraded research equipment, the design of research ships is more and more automatic and modularized (European science foundation 2009; Zhu et al. 2012). Therefore a series of new advanced research ships replace the overage ships.

There are a series of typical research ships in the period, such as the 18,000 t ocean drilling ship “JOIDES” built in the United States in 1984; the 3600 t “Atlantis III” built in the United States in 1997, which is the mother ship of Alvin DSV (Deep Sea Vehicle); the 6300 t ice comprehensive research ship “Maria S. Merian” built in Germany in 2006; the 3000 t comprehensive research ship “Sarmiento de Gamboa” built in Spain in 2007; the 5800 t comprehensive vessel “JAMES COOK” built in the British in 2007; the 57,000 t ocean drilling ship “CHIKYU” built in Japan in 2007; the icebreaker polar research ship “Akademik Tryoshnikov” built in Russia in 2012; the 6000 t new “DISCOVERY” built in the British in 2013; the 3200 t comprehensive survey ships “AGOR 27” and “AGOR 28” built in the United States in 2014; the 6000 t comprehensive survey ship “Investigator” built in Australia in 2014; the 8000 t new “SONNE” built in Germany in 2015; etc.

The building of Chinese research ships appears a little delayed from the middle 1980s to the end of the twentieth century (Li et al. 2012). After the

research ship “Dongfanghong 2” designed and built independently, the modified ships “Dayang 1” and “Snow Dragon,” and so on, the marine survey in China begins into the deep sea and the polar area. But the limited number of ships is not content with the actual demand of the development of marine survey in China. It enters into the peak development of research ships since the twenty-first century. There are, respectively, advanced research ships such as “Shiyan 1,” “Haiyangliuhao,” “Kexue,” “Xiangyanghong 10,” “Xiangyanghong 03,” “Xiangyanghong 01,” “Zhangjian,” “Jiageng,” etc.

Basic Type

Different countries have different strategic target, research area, foundation of parent ships, as well as capability of design and production. According to the characteristics and functional requirements of research ships, there are different classification methods (Meng et al. 2017; Wu 2017). Forms of research ships include mono-hull, catamaran, and special hull. According to the gradation standard of the US Federal Oceanographic Facilities Commission (shown in Table 1, FOFC 2001), gradations of research ships include global class, ocean class, regional class, and coastal class based on the ability of navigation and operation. Based on research tasks, they are divided into comprehensive research ships, subject research ships, and special research ships. However, common subject research ships are geophysical survey ships, geological survey ships, acoustic research ships, fishery research ships, meteorological research ships, environmental surveillance ships, buoy receive and release ships, etc. And special research ships are oceanographic drilling ships, polar research ships, space tracking ships, submersible carriers, archaeology ships, etc.

Comprehensive research ships regard multiple marine subjects and research fields as survey objective. They possess comprehensive survey capability of various functions and techniques. They can conduct marine geology, geophysics, geochemistry, biology, hydrometeorology, hydroacoustics, and fisheries and other marine survey. They can carry out comprehensive research from the upper atmosphere to the seabed through the air-sea interface. They mostly work in the ocean near their own sea or further area. Generally speaking, comprehensive research ships have more form values, cruising ability and self-supportability, and better properties, along with more complete survey equipment.

Fishery research ships are engaged in the scientific research of fishery resources and fishing ground environment, the experimental study of fishing gears and fishing methods, etc. They are capable of towing different types of fishing nets, collecting plankton or water samples, and carrying fish-finders. A fishery research ship is often designed and built along the same lines as a large fishing vessel, but with space given over to laboratories and equipment storage, as opposed to storage of the catch.

Geology research ships are applied in the research of marine geology, geophysics, and geothermics, by the means of seismic, gravity, and magnetic survey and sampling analysis. These studies can obtain the sedimentary and structure of seabed and make further information of the seabed mineral resources. These ships are equipped with special survey equipment as well as laboratories.

Oceanographic drilling ships can drill core samples of the seabed and obtain chemical and biological information of them. These ships are equipped with tall derricks, heavy lifting

Research Ship, Table 1 Gradation standard of research ships (FOFC 2001)

Gradation	Length/meters	Self-supportability/days	Endurance/nautical miles	Personnel
Global class	70	50	13,500	30–35
Ocean class	55–70	40	10,800	20–25
Regional class	40–55	30	8100	15–20
Coastal class	<40	<20	<5400	<15

equipment, and a full set of core sampling system, especially dynamic positioning system which can maintain a fixed position.

Acoustic research ships can conduct marine acoustic information acquisition for underwater acoustic intercept and acoustic research. Meanwhile, this information is applied in military affairs such as underwater target alert. Professional acoustic research ships must have lower noise and less vibration in order to satisfy the normal use of acoustic equipment.

Polar research ships are applied to investigate the polar information, such as hydrology, meteorology, and biology. They are constructed around an icebreaker hull, allowing them to engage in ice navigation and operate in polar waters. These ships usually have dual roles, particularly in the Antarctic, as polar replenishment vessels to the Antarctic research bases. Due to the specificity of working areas, polar research ships must have solid structure and larger values to support polar working for a long time.

Hydrographic research ships are designed to conduct hydrographic research and survey. Nautical charts are produced from this information to ensure safe navigation by military and civilian shipping. These ships usually mount equipment on a towed structure, for example, air cannons, used to generate a high-pressure shock wave to sound the strata beneath the seabed. This information is useful for detecting geological features which are likely to bear oil or gas. Naval hydrographic research ships often do naval research, for example, on submarine detection.

Key Technology

Comparing with regular ships, research ships have some specific technical requirements according to the demand of survey tasks.

Dynamic positioning system can propel and adjust the position of ships with sensors and thrusters automatically (Li 2014). Precise dynamic positioning system plays an important role in the operation process, such as clothing and recycling of survey equipment, as well as bottom sediment sampling (Zhang et al. 2015). In general, research ships are required to hold a

certain position and course, which are not far from a predesignated position.

Good maneuverability and steady slow propulsion performance are also important. The economical speed of research ships is 12–15 knots, while holding a low speed of 3–6 knots for a long time is necessary for measurement and trawling operation. Controllable pitch propellers, electric propulsion, and thruster systems can effectively realize slow sailing (Li 2014). To improve the maneuverability of research ships, they are always equipped with bow thrusters and stern thrusters, as well as active rudders.

In order to improve measurement accuracy, low noise and little vibration are necessary that strong hull can ensure low noise and little vibration of research ships. That is to say, they require good stability and surge immunity. It is effective to reduce noise and vibration by form and structure design of research ship as far as possible. Besides, advanced research ships are equipped with anti-rolling tank to reduce the freeboard and windward area.

Besides, research ships must have enough power load and electromagnetic compatibility. Certain redundancy and enough power load of the power station are necessary to guarantee the normal operation of different types of professional survey equipment. Instrument electricity must be separate from dynamics electricity and living electricity (Wu 2018). All electronic systems are mutually compatible and noninterfering to reduce and eliminate the electromagnetic interference. It is very important for ensuring security of the personnel and equipment on the research ship. In addition, acoustic research ships still need to set up undisturbed power supply.

In addition, deck area and laboratory area are the main workspace of field science research. Enough deck area can match the storage, fixation, and operation of different sizes of equipment, especially in a large comprehensive survey voyage. Laboratory area is generally divided into different cabins, such as universal dry or wet laboratory, thermostatic laboratory, chemical laboratory, data processing room, equipment room, sample storeroom, and so on. Laboratory

modularization becomes a development tendency of the design of research ships (Zhang et al. 2015).

Survey Equipment

The equipment of modern marine research ship consists of two parts, the operating control system and science survey equipment (Li et al. 2017).

Operating Control System

The operating control system referred to all the auxiliary equipment that support various research, including winch system, lifting system, air compressors, shipping network system, etc. The winch system generally includes conductivity, temperature, and depth (CTD) winches, optical-fiber cable winches, coaxial cable winches, steel rope winches, etc. Lifting and retraction system includes tail A frames, side A frames, flexible knuckle cranes, telescopic spreaders, etc. Air compressors provide compressed air for multiple seismic detection system. Shipping network system can carry out data management, ship management, web portal management, and so on.

Science Survey Equipment

Science survey equipment mainly includes underwater detection system, atmospheric sounding system, submarine exploration system, remote sensing information field verification system, and so on.

Underwater detection system is applied in comprehensive exploration of three-dimensional distribution features of marine hydrological, acoustic, chemical, and biological elements. We can study ocean circulation dynamic process, the distribution of fishery resources, and the regularity of marine acoustic propagation. The representative equipment includes conductivity, temperature, and depth (CTD), acoustic Doppler current profilers (ADCP), acoustic wave recorders, current meters, salinity meters, sound velocity profiler (SVP), fish-finders, wave buoys, underwater gliders, biological layered continuous sampling system, etc.

Atmospheric sounding system allows obtaining continuous real-time meteorological information to study three-dimensional air-sea

interaction and atmospheric sciences. The detecting equipment includes air-sea boundary layer observation system, shipborne automatic meteorological stations, aerosol time-of-flight mass spectrometer (ATOFMS), CO₂ measurement system, etc.

In addition, submarine exploration system is applied to detect the surface and formation of the seabed and then explore the sedimentary environment, geologic structural setting, and mineral resources below the seafloor. Common equipment includes multi-beam submarine geomorphic detection system, single-beam or multi-beam depth finders, sub-bottom profilers, marine gravimeters, marine magnetometers, submarine in situ observation system, multiple seismic detection system, deep-sea towing system, heat flow probes, side-scan sonar system, geological drilling sampling equipment, cone penetration test (CPT), underwater camera system, etc.

Furthermore, remote sensing information field verification system can obtain high-quality seawater information providing spot calibration and algorithm verification of satellite ocean remote sensing. The representative equipment includes waves and surface current detection system, underwater multispectral absorption and attenuation measuring instrument, offshore or underwater hyperspectral radiation measuring instrument, etc.

Key Applications

American Polar Research Ship “Sikuliaq”

“Sikuliaq” is an American polar research ship owned by the National Science Foundation and operated by the University of Alaska Fairbanks School, built in 2014 (shown in Fig. 1) (Sun and Liu 2018). The major research tasks include the effect of climate change and increased human use of Arctic regions on various issues such as ocean circulation and ecosystem dynamics.

It has a length of 79.6 m, a width of 15.85 m, a depth of 8.5 m, a draft of 5.715 m, a displacement of 3724 t, and a maximum speed of 14.2 kn. It can sail 18,000 nm at 10 kn. The ice class is Polar Class 5. It can break 1st-year sea ice up to 0.76 m

Research Ship,

Fig. 1 American polar research ship “Sikuliaq” (<https://www.wikipedia.org/>)



thick at a constant speed of 2 kn, which inspired her name. One of the most advanced research vessels ever built, it has extensive scientific facilities. In addition to 209 m² of built-in laboratories, it can accommodate two to four 20 ft scientific containers on the 405 m² aft deck. Designed for operations in ice-infested waters, it has a sloping icebreaker bow and a hull that is 2 ft wider at the bow than in the stern to reduce ice resistance. Besides, it also has dynamic positioning capability. It is equipped with diesel-electric propulsion system, powered by four MTU 4000 series high-speed diesel engines. It is one of the first vessels ever to be fitted with Icepod propulsion units, Wartsila brand of ice-strengthened azimuth thrusters that can be rotated 360° about the vertical axis. It is also fitted with a bow thruster to assist maneuvering at low speeds.

Chinese Polar Research Ship “Xue Long 2”

“Xue Long 2” is the first polar research ship constructed by China, scheduled to enter service in 2019 (shown in Fig. 2) (Xinhua Net 2017). She will serve both as a supply vessel for China’s research facilities in the Arctic and Antarctic regions and as a research vessel, with capabilities for geological and biological experimentation and surveying.

She measures 122.5 m long, with a beam of 22.3 m and a draft of 8.3 m at full load. She has a

displacement of 14,300 t. She has a diesel-electric propulsion system, with two 16-cylinder and two 12-cylinder engines, both Wartsila 32-series designs, powering two 7.5 MW Azipods that give her a speed of up to 15 kn in open water and 3 kn when breaking ice. She is classed as a PC3 icebreaker under the Polar Class of measurement and is able to break ice up to 1.5 m thick while traveling either ahead or astern. In addition, all of the generators are equipped with exhaust gas cleaning system to protect polar environment.

Japanese Ocean Drilling Ship “CHIKYU”

The largest ocean drilling ship in the world “CHIKYU” is a member of Japan Marine Association, built by Mitsubishi Heavy Industries shown in (Fig. 3) (Shen 2011). It has a length of 210 m, a width of 38 m, a depth of 16.2 m, a height of 130 m, a draft of 9.2 m, a displacement of 56,752 t, cruising ability of 14,800 nm under 10 kn, and a maximum speed of 12 kn. It can accommodate 200 passengers at a time.

The biggest feature is a huge riser drilling rig rising 121 m above sea level. It is capable of drilling 7000 m into the ocean floor in deep sea areas with a water depth 4000 m. It is equipped with six diesel generators with a total power of 35,000 kW, one 2550 kW bow thruster, and three 4100 kW azimuth thrusters and three same

Research Ship,

Fig. 2 Model of Chinese polar research ship “Xue Long 2” (<http://image.baidu.com/>)

**Research Ship,**

Fig. 3 Japanese ocean drilling ship “CHIKYU” (<http://image.baidu.com/>)



azimuth propellers in the stern. Besides, it also has dynamic positioning capability with DPS-B, with 360° omnibearing observation.

**Chinese Geophysical Research Ship
“Haiyangdizhi Bahao”**

“Haiyangdizhi Bahao” is the first geophysical survey ship that is designed and constructed in China (shown in Fig. 4). It is capable of 3D or 4D high-precision short track seismic survey, equipped with six seismic cables, up to eight cables. It meets the demand of natural gas hydrate survey, regional geological survey, and resource exploration.

The ship has a length of 88 m, a width of 20.4 m, a depth of 8 m, a draft of 6.2 m, and a maximum speed of 15.4 kn. It can sail 16,000 nm under 12.1 kn continuously. It is a double-skin ship equipped with double catheter controllable pitch propellers. The electric propulsion system is made up of four main generators and one berth generator, with double-layer vibration isolation system. In addition, it is satisfied with the environmental protection requirements of CLEAN class notation by China Classification Society (CCS). It is also qualified by COMF (NOISE3) and COMF (VIB 3) comfort class notation.



Research Ship, Fig. 4 Chinese geophysical survey ship “Haiyangdizhi Bahao” (<http://www.gmgs.cgs.gov.cn/>)

**Research Ship,
Fig. 5** Australian
oceanographic research
ship “Investigator” ([https://
www.csiro.au/](https://www.csiro.au/))



Australian Oceanographic Research Ship “Investigator”

“Investigator” is an Australian oceanographic research ship which was constructed in Singapore and is owned and managed by the Commonwealth Scientific and Industrial Research Organisation (CSIRO) (shown in Fig. 5). It is operated by the Australian Marine National Facility to undertake oceanographic, geoscience, ecosystem, and fisheries research (Australian Broadcasting Corporation 2014).

“Investigator” has a length of 93.9 m, a width of 18.5 m, a depth of 9.45 m, a draft of 6.2 m, and a maximum speed of 12 kn (Sabto BGM 2012). It is able to accommodate up to 40 scientists and go to sea for up to 60 days and 10,000 nm at a time. Special features of the ship are a “gondola,” similar to a winged keel, mounted 1.2 m below the hull, and two drop keels, which can be lowered to a maximum of 4 m below the hull, to carry scientific instruments below the layer of microbubbles created by the movement of the ship’s hull



Research Ship, Fig. 6 Germany fuel research ship “Atair II” (<http://image.baidu.com/>)

through the water (Brian et al. 2012). Such instrumentation includes acoustic mappers, a pelagic sediment profiler to produce maps of the sea floor, and two ADCP. The hull and the machinery of the ship have been designed to operate as quietly as possible to enhance its scientific capabilities.

Germany Fuel Research Ship “Atair II”

“Atair II” is the first research ship in the world with LNG as fuel, built by Germany Maritime and Hydrological Administration, expected delivery in 2020 (shown in Fig. 6) (Sun and Liu 2018). It is mainly applied in hydrological survey and wreck searching, as well as marine environment observation in the North Sea and the Baltic Sea. At the same time, it is the technical test platform for navigation and radar system.

“Atair II” has a length of 74 m, a width of 17 m, a draft of 5 m, and a capacity of 18 crew members and 15 scientists. It is satisfied with DNV GL classification and SILENT-R class notation. It also has one 6-cylinder Wartsila 20 diesel engine, two 1110 kW of Wartsila 20DF diesel engines, and two sets of exhaust gas cleaning system. The Wartsila LNGPac fuel system has the gas storage capacity of 130 m³, supporting the gas consumption of 10 days approximately. Besides, it also has dynamic positioning capability (Eworldship 2019).

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Research Vessel

► Research Ship

Resistance

► AUV/ROV/HOV Resistance

Resistance of Polar Vessel

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Introduction

When a ship is transiting in ice-covered water, there will be ice resistance on the hull due to the ship-ice interaction. The ice resistance components depend on the ice conditions, such as level ice or brash ice, and thus the magnitude of the components will vary. The total ice resistance is

given as summation of open water resistance and ice resistance, where ice resistance consists of breaking, submersion, and friction resistance. The breaking resistance depends on bow shape, ice thickness, and ice strength, while the submersion resistance will vary with the bow shape and ice thickness, and the friction resistance will depend on the hull form, speed, and thickness of ice and snow. The ice resistance can be found from full-scale measurements during ice trials, by model tests, or by application of semiempirical and numerical ice prediction resistance methods, which are applied in an early design phase to indicate the ice resistance for different ice conditions.

Definition

When sailing in ice-covered water, there will be a force subjected to ice (level ice and crushed ice) and fluid (water and air) acting on the ship which is opposite to the movement of the ship. We call the force as polar ice resistance.

According to Riska (2011a), the ice resistance can be divided into three main force components: breaking forces, submergence forces, and sliding forces. The breaking forces are related to the moment where the ice is breaking, which includes crushing, bending, and turning of ice, and the submergence forces are related to the submerging of ice floes. The sliding forces will include the frictional forces between ship hull and ice, and due to the frictional forces, the ice resistance will be speed dependent (Riska 2011b).

R

Empirical Formula to Calculate Ice Resistance

Spencer Method

According to Spencer (Spencer 1992), the ice resistance can be divided into four main force components: forces from breaking the ice, forces from submerging the broken ice, forces from friction along the ship hull, and hydrodynamic forces.

$$R_T = R_{OW} + R_B + R_C + R_{BR} \quad (1)$$

where R_T is the total resistance, R_{OW} is open water resistance, R_B is submersion resistance, R_C is ice clearing resistance, and R_{BR} is breaking resistance. That is, the total resistance is composed of open water resistance, submersion resistance, ice clearing resistance, and breaking resistance.

The open water resistance mainly refers to the water resistance. When a ship passes through the open water (without ice), the open water resistance can be measured directly by the general tests in the ship model towing tank.

It is difficult to measure the individual ice resistance components by the real ship tests. However, the individual resistance components can be separated by changing the test conditions in the ship model tests conducted on ice tank. At a certain speed, the pre-cutting ice resistance of the model R_{PS} is measured under the pre-cutting ice condition, and the breaking resistance R_{BR} can be separated by subtracting the resistance under the pre-cutting ice condition from the total resistance. Among them, the pre-cutting ice resistance is:

$$R_{PS} = R_{OW} + R_B + R_C \quad (2)$$

The submersion resistance R_B is caused by the fact that the broken ice blocks squeezing of the bottom of the ship due to buoyancy, which accounts for 10–15% of the total resistance. If resistance R_{OW} in open water is known, ($R_B + R_C$) can be separated. It is assumed that the submersion resistance R_B is independent of velocity and the clearing ice resistance R_C is related to velocity. Therefore, the model resistance is measured at a very low speed under the condition of the pre-cutting ice floe. At this time, it can be considered that the ice clearing resistance is equal to zero because the ice clearing resistance is related to the velocity. Similarly, open water resistance is also equal to zero. In this way, submersion resistance can be separated in the very low-speed pre-cutting ice condition:

$$R_B = R_{PS(V=0)} = C_B \Delta \rho g h_i B T \quad (3)$$

In the formula, C_B is the submersion resistance coefficient, $\Delta \rho$ is the difference between the water density and the ice density, g is the acceleration of

gravity, h_i is the ice thickness, B is the width of the ship, and T is the draft of the ship.

As the broken ice slides along the hull's surface, it creates ice clearing resistance. The ice clearing resistance is obtained by subtracting the submersion resistance and open water resistance from the pre-cutting ice resistance. Ice clearing resistance accounts for 10–15% of the total resistance. Therefore, the ice clearing resistance can be expressed as:

$$R_C = R_{PS} - R_{OW} - R_B \quad (4)$$

If the open water resistance R_{OW} is known, it is possible to separate submersion resistance and ice clearing resistance R_C according to formulas (2, 3, and 4).

Breaking resistance accounts for the largest part of the total resistance, which can be expressed as:

$$R_{BR} = R_T - R_{PS} \quad (5)$$

Colbourne Method

Colbourne from the Institute of Oceanographic Technology of Canada, aiming at the ship model resistance test under the crushed ice condition and taking into account the different concentrations of crushed ice conditions, developed an analytical method to analyze ship model resistance tests under crushed ice conditions.

Unlike the Spencer method, Spencer assumes that the resistance of ice floe is speed independent, and the resistance of clearing ice is related to the speed. Colbourne regards the submersion resistance and the clearing ice resistance as a whole, that is, the total resistance of the ships in ice area under the condition of broken ice is divided into open water resistance and breaking resistance, and the relationship between breaking resistance is square to the velocity; Spencer did not consider the impact of crushed ice concentrations on breaking resistance, but Colbourne thought that breaking resistance and crushed ice concentration have a powerful relation where the value of resistance is different for different ship types.

In the case of broken ice, assuming that the size of the broken ice is small relative to the size of the ship and when the ship passes through broken ice

area, the occurrence of broken ice breakage is so small that the resistance caused by this breakage can be ignored. Then, the total resistance under broken ice conditions can be divided into two components: one is open water resistance, and the other is called crushed ice resistance due to various movements of broken ice around the hull (including sinking, turning, sliding, etc.). Among them, the open water resistance can be measured by the resistance towing test of the ship model towing tank. And the breaking resistance caused by broken ice movement is obtained by subtracting open water resistance from the total resistance measured by ship model resistance test under the broken ice condition. The total resistance under broken ice conditions can be expressed as:

$$R_T = R_{OW} + R_p \quad (6)$$

where R_{OW} is the open water resistance and R_p is the breaking resistance.

Lindqvist Method

In 1989, Gustav Lindqvist (Lindqvist 1989) proposed a method for calculating ice resistance. He divided the total resistance into three parts: ice-breaking resistance, underwater resistance, and velocity-dependent resistance (in which the ice-breaking resistance is divided into two parts: extrusion force of stern and bending destructive force).

Breaking Resistance

It is difficult to measure the stern squeezing pressure. It can be approximated by the probability method. The force of the vertical component acting on the ice F_V is estimated as

$$F_V = 0.5\sigma_f h_i^2 \quad (7)$$

where the σ_f is the bending strength of ice and h_i is the thickness of ice.

By analyzing the squeezing process and considering the geometry, the squeezing resistance R_C can be obtained:

$$R_C = F_V \cdot \tan \psi + \mu \cdot \frac{\cos \phi}{\cos \psi} / 1 - \mu \cdot \frac{\sin \phi}{\cos \psi} \quad (8)$$

where $\psi = \arctan \tan \phi / \sin \alpha$, ϕ is stem angle, α is angle of waterline entrance, and μ is friction factor.

Bending destructive force R_B can be expressed as

$$R_B = \frac{27}{64} \cdot \sigma_f \cdot B \cdot h_{ice} / \sqrt{\frac{E}{12(1-\nu^2)g\rho_w}} \cdot \tan \psi + \mu \cdot \cos \phi / \cos \psi \sin \alpha \cdot (1 + 1/\cos \psi) \quad (9)$$

where h_{ice} is ice thickness, E is Young modulus, ν is Poisson ratio, σ_f is ice bending strength, B is the breadth of the ship, g is the acceleration of gravity, and ρ_w is the density of seawater.

Underwater Resistance

Underwater resistance R_S consists of two parts: potential energy loss and friction resistance; the resistance value can be calculated by the formula:

$$R_S = (\rho_w - \rho_i) \cdot g \cdot h_{tot} \cdot B \cdot K \quad (10)$$

$K =$

$$\left\{ T \cdot B + T/B + 2T + \mu \cdot \left[\frac{(0.7L - T/\tan \phi - B/4 \tan \alpha) +}{T \cos \phi \cos \psi \sqrt{1/\sin^2 \phi + 1/\tan^2 \alpha}} \right] \right\} \quad (11)$$

where ρ_i is ice density, ρ_w is water density, T is extreme draft, L is the length of the ship, and h_{tot} is the total thickness of ice and snow.

Velocity-Related Resistance

Resistance associated with speed is more uncertain than other parts. According to the research, the resistance increases linearly with the increase in velocity.

Based on the above four parts, using the empirical constants to obtain the total resistance sailing in ice can be expressed as

$$R_i = (R_C + R_B) \cdot \left(1 + 1.4v/\sqrt{gh_i} \right) + R_S \cdot \left(1 + 9.4v/\sqrt{gL} \right) \quad (12)$$

Keinonen Method

By analyzing the resistance model data of several kinds of parent ships, the ice resistance of ice-

breaker at a speed of 1 m/s is obtained by this method (Keinonen and Browne 1991). The main parameters of the method are ship size, inclination angle, hull condition, the thickness of ice and snow, bending strength, salinity, and temperature of sea water. The ice resistance is expressed as:

$$R_{ice} = 0.015 \cdot (F_{si} \times F_{sh} \times F_i) \quad (13)$$

where F_{si} is size factor; F_{sh} is form factor; and F_i is ice factor.

After that, Keinonen analyzed the situation of velocity exceeding 1 m/s and put forward the expression of total resistance sailing in the ice area as follows:

$$R(v)_t = R(v)_{ow} + R(1m/s)_i + R(> 1m/s)_i \quad (14)$$

where $R(v)_t$ is total resistance at velocity v ; $R(v)_{ow}$ is open water resistance at velocity v ; $R(1m/s)_i$ is ice resistance at 1 m/s; and $R(> 1m/s)_i$ is resistance increases at speeds above 1 m/s.

Riska Method

Riska's formula (Riska et al. 1997) for calculating resistance is also based on a series of empirical coefficients, in which the empirical coefficients are obtained from multiple sets of real ship tests of different types of ships in the Baltic Sea.

The main expression formula of Riska resistance is $R_T = R_{openwater} + R_{ice}$. The resistance sailing in the ice is divided into two parts: open water resistance $R_{openwater}$ and the resistance R_{ice} caused by ice. At present, the prediction method of open water resistance of ships is relatively mature. However, there are two parts for the resistance in ice R_i : inertial resistance related to velocity and direct resistance independent of velocity. The expression formula is as follows:

$$R_i = C_1 + C_2 V \quad (15)$$

where

$$C_1 = f_1^{1/2T/B+1} B L_{par} h_i + (1 + 0.0021\phi) \cdot (f_2 B h_i^2 + f_4 B L_{bow} h_i) \quad (16)$$

$$C_2 = (1 + 0.063\phi) \cdot (g_1 \cdot h_i^{1.5} + g_3 \cdot B \cdot h_i) + g_3 \cdot h_i \cdot (1 + 1.2T/B) \cdot B^2/\sqrt{L} \quad (17)$$

L , B , and T indicate the length, width, and draft of the boat, respectively; ϕ is bow inclination, h_i is ice thickness, and L_{par} and L_{bow} are the length of the parallel middle body and the length of the bow, respectively. The empirical coefficient f , g in the formula is given in Table 1 (ITTC 2017).

Enkvist Method

$$R_{TOT} = C_1 B h \sigma + C_2 B h T \rho_{\Delta} g + C_3 B h \rho_i v^2 \quad (18)$$

where R_{TOT} is ice resistance; B , T represents the width and draught of the ship, respectively; h is ice thickness; σ is ice bending strength; ρ_i is ice density; ρ_{Δ} is density difference between water and ice; g is the acceleration of gravity; v is the speed of the ship; and $C_1 C_2 C_3$ is the dimensionless factor (ENKVIST 1972).

In the test, the velocity-related terms can be isolated by comparing at low speed ($v \approx 0$) and normal speed; and the sinking items can be isolated by performing the pre-cutting ice ($\sigma = 0$) test. This experimental design method can determine the relative importance of the three resistance components in the equation.

Finnish-Swedish Ice Class Rules Method

The Finnish-Swedish Ice Class Rules (2002) provide a formula for the resistance of a ship sailing in the broken ice:

Resistance of Polar Vessel, Table 1 coefficient of Riska's formula

Variable	Value
f_1 (kN·m ⁻³)	0.23
f_2 (kN·m ⁻³)	4.58
f_3 (kN·m ⁻³)	1.47
f_4 (kN·m ⁻³)	0.29
g_1 (kN·s·m ^{-2.5})	18.9
g_2 (kN·s·m ⁻³)	0.67
g_3 (kN·s ¹ ·m ^{-3.5})	1.55

$$\begin{aligned}
R_{ch} = & \frac{1}{2}\mu_B\rho_{\Delta}gH_F^2K_p\left[\frac{1}{2} + \frac{H_M}{2H_F}\right]^2 \\
& \cdot [B + 2H_F(\cos\delta - \frac{1}{\tan\psi})] \\
& \cdot (\mu_h \cos\phi + \sin\psi \sin\alpha) \\
& + \mu_B\rho_{\Delta}gK_0\mu_hL_{par}H_F^2 \\
& + \rho_{\Delta}g\left(\frac{LT}{B^2}\right)^3H_MA_{WF}F_n^2
\end{aligned} \quad (19)$$

where $\mu_B = 1 - p$; p is porosity ($\mu_B = 0.8 \sim 0.9$); ρ_{Δ} is difference between ice and water density; g is acceleration of gravity; K_p is passivation stress (soil mechanics); δ is the inclination of the side-wall of broken ice; μ_h is coefficient of friction between ice and hull; L , B , and T are, respectively, represented the length, width, and draft of the ship; ϕ is the included angle between the waterline and the mid-longitudinal section line; ψ is angle of flare; α is waterline incident angle; K_0 is static lateral pressure coefficient; L_{par} is the length of a parallel middle body at the waterline; A_{WF} is waterplane area at the bow of the hull; F_n is Froude number; H_M is the thickness of broken ice in the middle of waterway; and H_F represents the thickness of the broken ice moving from the bow to both sides of the parallel middle body. It is a function of the width of the ship, the thickness of the channel, and the two slanting angles γ and δ depending on the internal properties of the broken ice (generally $\gamma = 2^\circ$, $\delta = 22.6^\circ$), as follows:

$$\begin{aligned}
H_F = & H_M + B/2 \tan\gamma + (\tan\gamma + \tan\delta) \\
& \times \sqrt{B\left[H_M + \frac{B}{4}\tan\gamma\right] / \tan\gamma + \tan\delta}
\end{aligned} \quad (20)$$

when $B > 10$ m and $H_M > 0.4$ m, the formula can be approximately simplified to

$$H_F = 0.26 + (BH_M)^{0.5} \quad (21)$$

Angle of flare ψ can be eliminated from the formula on the basis of the following triangulation:

$$\begin{aligned}
\sin\psi &= \tan\phi / \sqrt{\sin^2\alpha + \tan^2\phi} \\
\psi &= \arctan(\tan\phi / \sin\alpha)
\end{aligned} \quad (22)$$

It can be seen that the ship resistance in floating ice zone can be preliminarily estimated by the

Finnish-Swedish standard if the ship lines, principal dimensions, and the correlation property coefficient of floating ice are given.

Numerical Modeling of Ice Resistance

Discrete (Distinct) Element Method (DEM)

Discrete element method (distinct element method, DEM) already has a long history of deployment in the field of ice mechanics. Open source implementations are also available. For example, LIGGGHTS implements DEM (Kloss et al. 2012; Morgan et al. 2015; Morgan 2016; Yulmetov et al. 2017) and employed this software to simulate ice structure interaction.

Smoothed Particle Hydrodynamics (SPH) and Moving Particle Semi-Implicit Method (MPS)

Smoothed particle hydrodynamics (SPH) is one of the Lagrangian methods to handle continuum mechanics. It was originally developed for fluid simulations but is now extended to handle solid mechanics. Some commercial software such as LS-DYNA implements SPH to handle fluid and/or solid simulations.

Moving particle semi-implicit (MPS) method was separately developed and investigated with SPH as an implementation of incompressible fluid simulation but can be considered as an implementation of SPH with different spatial discretization schemes. Recently, it is also extended to handle solid mechanics. Ren et al. (2017) deployed this method to simulate the interaction of an ice beam and a large water droplet.

Fluid-Structure Coupling

Fluid-structure interaction (FSI) is one of the most important but difficult fields of investigation with numerical simulations. However, a few commercial packages implement FSI recently so that it becomes easier.

Song, Kim, and Amdahl (Song et al. 2015) employed LS-DYNA to investigate collision of the steel structure with an ice floe. Shigihara, Ishibashi, and Konno (2015) employed STAR-CCM+ to investigate collision of a ship with a single ice floe. Vroegrijk (2015) also employed

STAR-CCM+ to investigate ship navigation in brash ice channel.

Model Test of Ice Resistance

The model test of ice resistance is introduced in the Specialist Committee on Ice, Technical Committees and Group of the 28th ITTC.

Methods to Determine Ice Resistance Using Self-Propelled Model Test

An increasing share of self-propulsion model tests is one of the modern trends in test practices of ice basins. Most leading ice basins took to using self-propelled models rather than non-propelled models in ice resistance towing tests.

Method of Krylov State Research Center

(1) First Method

Traditionally, in the KSRC ice basin practices, the self-propelled models have been used to find ice resistance of model in astern mode. In this case, the captive test method is employed when a model is rigidly fixed to the towing carriage via dynamometer and outfitted with running propellers, as shown in Fig. 1. Propeller thrust is measured with special-purpose dynamometers.

For a propelled ship model running astern, the ice resistance R_I is found from the following force equation:

$$R_I = -(F_{means} + (1 - t) \cdot \sum_i P_i) \quad (23)$$

where F_{means} , force measured by dynamometer; t , thrust deduction factor; and $\sum_i P_i$, total thrust of propellers.

(2) Second Method

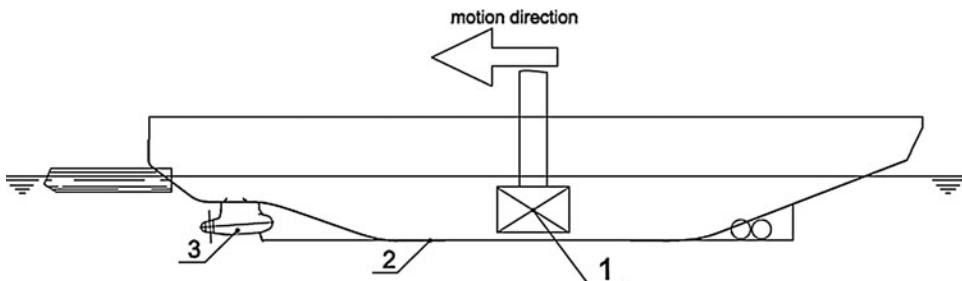
A model with running propellers is towed at a specified speed by towing carriage. The force of model-carriage interaction F_I is measured. The speed of propeller rotation is chosen to match the propulsive thrust with the model speed. In this case, a close to full-scale flow pattern around the hull is obtained. Experiments in a continuous ice sheet, when completed, leave a channel packed up with brash ice. This channel after being cleared of brash ice is used to repeat the self-propelled model tests, now in open water. These experiments are conducted at the same speed of propeller rotation and model speed to measure the force of model-carriage interaction F_W .

The pure ice resistance force R_I is calculated based on the test data obtained in ice and open water conditions using Eq. (24):

$$R_I = F_I + F_W \quad (24)$$

It should be noted that the forces F_I and F_W retain their actual signs when being summed up.

Rigid connection between the self-propelled model and towing carriage makes it possible to accurately set the model speed in ice. The force between towing carriage and model measured by dynamometer indicates whether the propeller



Resistance of Polar Vessel, Fig. 1 Astern test setup: 1, dynamometer; 2, ship model hull; 3, propulsion pod model (Specialist Committee on Ice, Technical Committees and Group of the 28th ITTC, 2017)

thrust is higher than the ice-plus-water resistance or not sufficient to overcome this total resistance. This force is defined by Eq. (25):

$$F_I = R_I + R_W - T_E \quad (25)$$

where R_W – model resistance due to water and T_E – effective thrust of model's propulsion system. Model tests in a channel cleaned of brash ice make it possible to determine the hydrodynamic resistance force F_W more accurately than it is done using the Finnish method because in these tests, the hydrodynamic resistance has practically no wave-making component. In ice basin, the hydrodynamic resistance should be determined following all requirements which are applied to the same kind of experiments when conducted in towing tanks. In these tests the force between model and towing carriage is determined, which is defined according to Eq. (26):

$$F_W = T_E - R_W \quad (26)$$

From Eq. (25) follows the ice resistance in Eq. (27)

$$R_I = F_I + T_E - R_W \quad (27)$$

Then the final formula for the ice resistance is Eq. (24).

Method of Aker Arctic

The ice basin of Aker Arctic is using a different technique involving a series of preliminary open water tests usually carried out in the ice basin itself. Under this method, ice resistance is determined using self-propulsion tests at the ship propulsion point.

Model ice resistance is determined in a number of steps. Self-propelled model is tested under bollard pull condition in open water. In this case, the propeller thrust is measured at different propeller speeds. These data are then used to find the speed which provides the specified thrust of propulsion system in bollard pull condition. Towing tests of the model with running propellers are performed in open (ice-free) water with the model fixed to the towing carriage via dynamometer. Propellers are

run at speeds corresponding to 80, 100, and 120% power consumption. In these tests the brake force (which is applied from the towing carriage side to model) is measured versus model speed (at constant number of revolutions that could be different for side propellers and middle propeller). The open water tests give brake force versus model speed.

Method of HSVA

HSVA ice basin is using both self-propelled and non-propelled models in ice resistance tests. Ice resistance tests with self-propelled models are performed in a captive setup. The model is towed at a given speed, while propeller revolution numbers are varied. Propeller revolutions range from practically zero to a certain value specified to assure that the force recorded by dynamometer changes its direction (model now pulling the towing carriage). As shown in Fig. 2, a mandatory test condition is practically zero number of propeller revolutions. Based on the results of this test, the ice resistance of model is found.

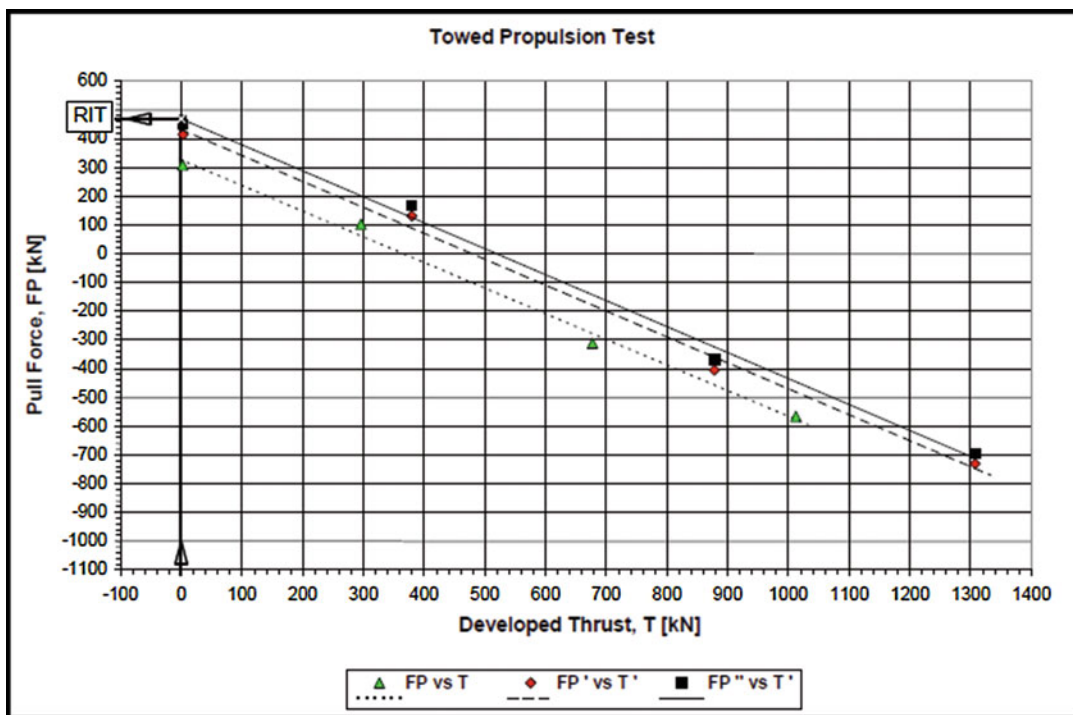
Model Ice in Ice Resistance Experiment

Type ice can be divided into two categories: frozen model ice and nonfrozen model ice. The frozen model ice is mainly used in the ice basin laboratory at home and abroad, and the non-freezing model ice is mostly used in the conventional towing tank laboratory without an ice basin.

The frozen model ice is divided into brine ice, urea ice, and EG/AD/S ice. Brine ice, also known as the first-generation model ice, is made from a certain concentration of sodium chloride solution (Schwarz 1977). Urea ice, also known as the second-generation model ice, is made up of a 1.3% concentration of urea solution (Timco 1979, 1980; Hirayama 1983).

The EG/AD/S ice, also known as the third-generation model ice, is made from aqueous solutions of three different additives including the mixture of ethylene glycol (EG), aliphatic detergent (AD), and sugar (S) (Timco 1986).

With the development of experimental technology and the requirements of ice zone tests, various kinds of nonfreezing synthetic model ice have come out one after another. The advantages



Resistance of Polar Vessel, Fig. 2 Determination of ice resistance by HSVA method (HSVA Report 2012)

of nonfreezing model ice are that it does not require low-temperature manufacturing and it's easier to control. At present, the main model ice used in ice ship resistance test are fragments of semi-refined paraffin wax and polypropylene material, as shown in Fig. 3.

Two Methods to Conduct the Towing Tests

The methods to conduct the towing tests can be found in the website of HSVA ice tank (<https://www.hsva.de/our-services/model-testing/tests-in-the-ice-tank.html>).

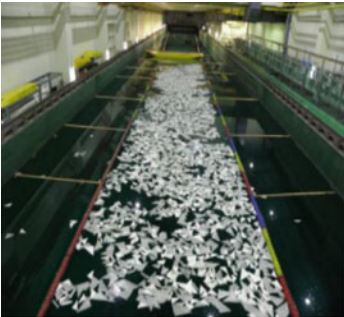
Towed Propulsion

During the towed propulsion test, the model is connected to the service carriage, which is pushed by the main carriage, via a rigid rod that itself is attached to a load cell at the bows of the model, as shown in Fig. 4. After being accelerated the model is towed at a constant speed. The propeller rpm is changed in four steps from near idling condition through a value close to the self-propulsion point up to a value well above the self-propulsion

condition. Both the thrust of propellers and the coupling force are measured, and therefore the actual propulsion point for the given speed can be determined throughout interpolation from the linear correlation between thrust and pull force. The resistance can be read from the intersection point of the same curve with the axis of ordinates.

Fixed Mode Testing

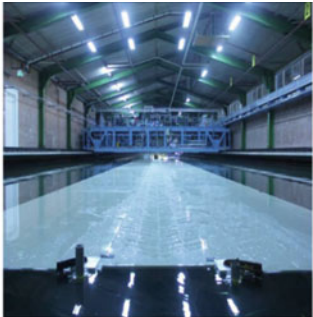
Regardless of the type, each model can be tested in fixed mode. This means the models' motion is blocked in a number of degrees of freedom, as shown in Fig. 5. This can be partially inferred that only certain directions of motion are blocked or completely – in this case all six degrees of freedom are blocked. Ice-induced vibrations can be investigated in a fixed mode where the model is rigidly connected to the basin floor, and only motions in one or two directions are allowed. Ice loads on a vessel may be determined in a fixed setup where the vessel is rigidly connected to the main carriage and pushed through the ice. A special rotation platform allows testing the



Artificial ice- frag-
ments of semi-refined
paraffin wax



Artificial ice- frag-
ments of semi-refined
paraffin wax



Artificial ice- polypro-
pylene material

Resistance of Polar Vessel, Fig. 3 Artificial ice (Kim et al. 2013; Guo et al. 2016; Werff et al. 2015)

Resistance of Polar Vessel, Fig. 4 Model towed by carriage using rigid pole (HSVA Report 2012)



Resistance of Polar Vessel, Fig. 5 Model directly under the carriage (HSVA Report 2012)



subject, vessel, or structure, under certain oblique angles. In all cases, forces in three axes and moments around these axes are determined by means of a six-component scale. In addition to the forces accelerations, velocities and movements of the structure/vessel will be measured.

Cross-References

<https://www.hsva.de/our-services/model-testing/tests-in-the-ice-tank.html>

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Resistance Test

► Towing Tank Test

Resonance Roll

► Dead Ship Condition

Resource Assessment

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Synonyms

ECMWF - European Centre for Medium-Range
Weather Forecasts; SWAN - Simulating Waves
Nearshore

Definition

Resource assessment is an evaluation of the long-term available offshore renewable energy resources (in terms of energy or power), including wind, wave, and marine current energy resources. This can be done from a global and regional perspective for site screening/selection or for a specific location for the development of commercial farms. Offshore renewable energy resource can also be distinguished as potentially available resource (assuming that all the energy can be converted into electricity) or technically available resource (which can be practically converted by a given device or technology and therefore is device- and site-specific).

Wind Energy Resource

Wind energy refers to the kinetic energy of air in motion, and the total wind energy during a period of t through an area of A normal to the mean wind direction is $E = \frac{1}{2} A \rho U^3$, where ρ is the density of the air and U is the mean wind speed. Then the power is the energy per unit time $P = \frac{1}{2} A \rho U^3$. Often, we use the wind power density $\frac{P}{A} = \frac{1}{2} \rho U^3$ for resource assessment and spatial distribution of wind power. In addition to the parameter of mean wind speed, turbulent intensity factor is often estimated based on measurements or simulations, and it is a very important parameter that is related to dynamic loads and responses of offshore wind

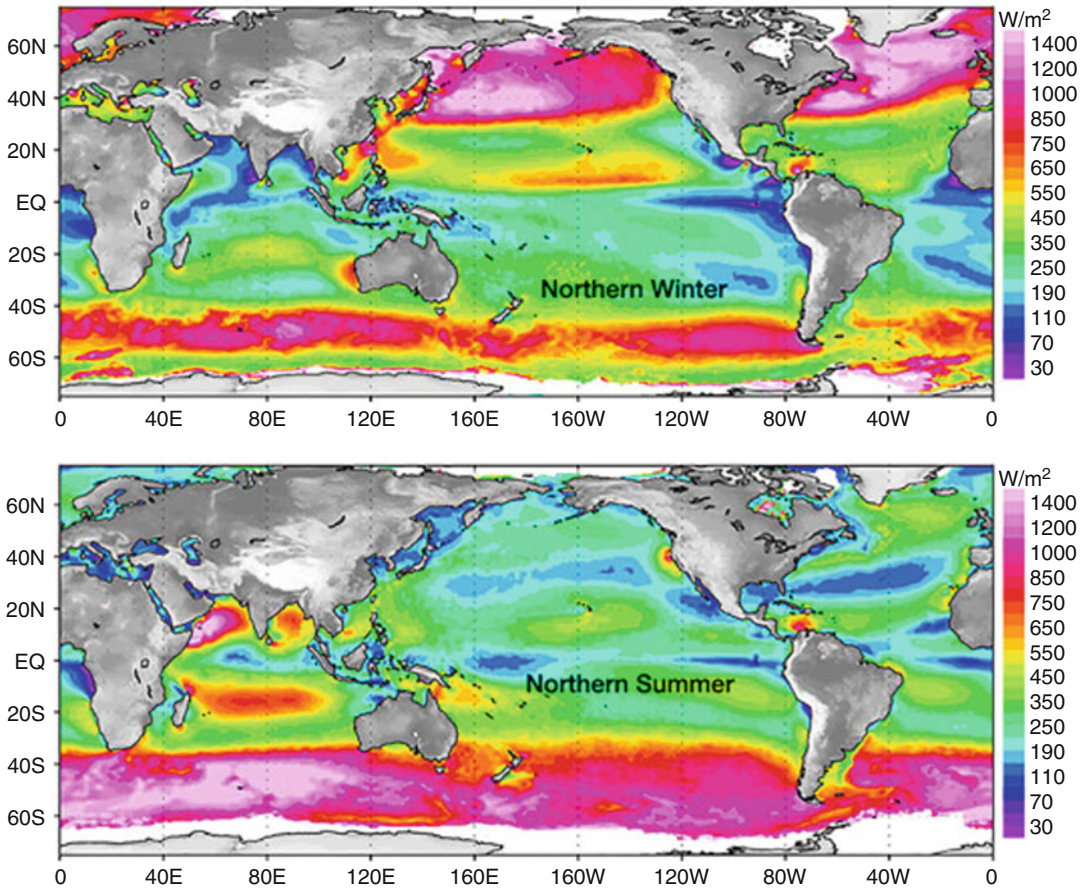
turbines, although it is not so important for power estimation.

Figure 1 shows the global spatial distribution of average offshore wind power density with respect to the winter and the summer seasons of the Northern Hemisphere (NASA/JPL 2008). It clearly shows the non-uniform distribution of the wind power density around the globe, with larger wind power in northern or southern areas than that near the equator. The equator area has less variation in wind power density (and therefore wind speed) among different seasons, while wind speed in winter is much larger than that in summer for the northern or southern area. Because of the considerations on economics and survivability of offshore wind turbines, it is not expected to see in the near future wind turbines in the middle of oceans, far from the shore. Offshore wind is now being developed from near shore gradually towards deeper waters.

The available power in wind is proportional to the cube of wind speed. Therefore, an accurate estimation of the mean wind speed is important for resource assessment. However, the mean wind speed also varies vertically, increasing when the height from the sea surface increases, which is referred to as wind shear. Figure 2 shows the annual mean wind speed distribution offshore Europe for five standard heights. Most of the offshore wind turbines today have a nacelle height between 70 m and 100 m. The mean wind speed at the height of 100 m is larger than 10 m/s for the northern North Sea area, while it is about 8.5–10 m/s for the North Sea and 6–8.5 m/s for the Mediterranean Sea.

If we consider a mean wind speed of 12 m/s as the rated wind speed of an offshore wind turbine and a maximum efficient of 50%, a rotor area of 11,600 m² and therefore a blade length of 60 m are needed to produce 5 MW of electric power.

On the other hand, the mean wind speed varies also in time. From a long-term perspective (i.e., 25 years of lifetime), a probabilistic distribution (for example, Weibull distribution) of mean wind speed for a given site is typically established using either measurement data or hindcast data (see Li et al. 2015). Such wind speed distribution can be used in combination with the wind turbine power



Resource Assessment, Fig. 1 Global distribution of offshore wind power density in winter and summer (NASA/JPL 2008)

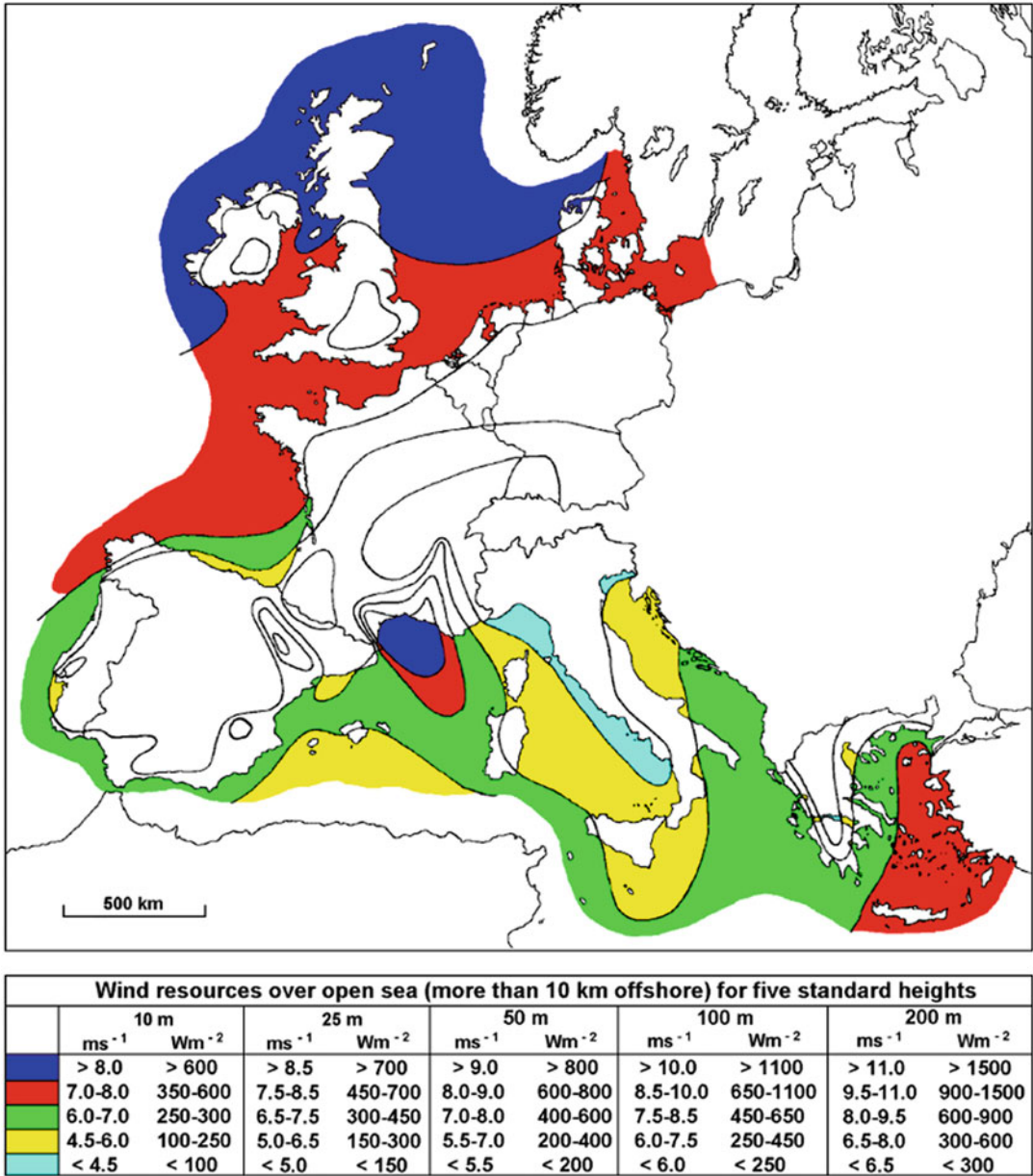
curve to estimate the annual power output from a turbine or from a farm.

As of today, there are many wind resource maps that are already developed at the global level by government or research agencies, as shown in Fig. 1. More detailed regional and local wind resource assessment, using either numerical models or in situ measurements (such as WASP – Wind Atlas Analysis and Application Program (Mortensen et al. 2005)), is needed for developing commercial offshore wind farms. Satellite-borne remote sensing observations, purpose-built meteorological masts, and ground-based remote sensing techniques (such as SODARs – Sonic Detection and Ranging; LIDARs – Light Detection and Ranging) are developed for specific wind speed measurement and resource assessment (Sempreviva et al. 2008).

Wave Energy Resource

Energy in propagating gravity ocean waves contains both kinetic and potential energy, which alternates when the waves propagate. It is the wave kinetic energy that can be absorbed and converted into electricity using a wave energy converter device.

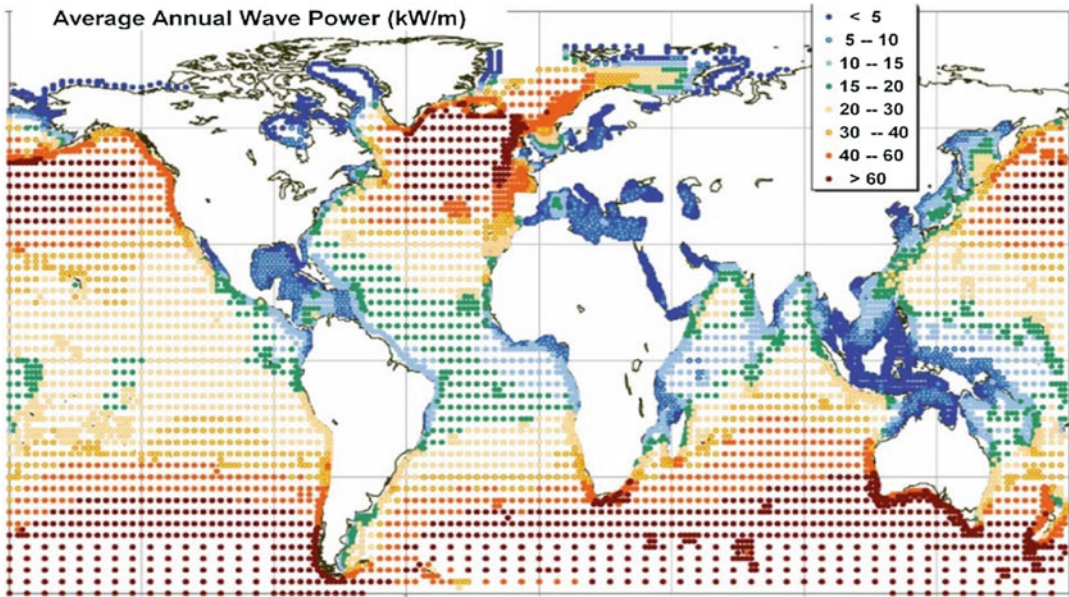
For a regular wave with a height of H , the average energy E stored on a horizontal water surface with a unit area is $E = \frac{1}{8} \rho g H^2$, where ρ is the density of the sea water and g is the gravitational acceleration (Falnes 2002). Wave power P , which is the wave energy flux through a vertical plane of unit width perpendicular to the wave propagation direction, is $P = c_g E = \frac{1}{32\pi} \rho g^2 T H^2$, where c_g is the group velocity and T is the wave period. For example, when $H=2\text{m}$, $T=10\text{s}$, we can get $P=40\text{kW/m}$. For an irregular



Resource Assessment, Fig. 2 Distribution of annual mean wind speed and wind power density offshore Europe (Troen and Lundtang Petersen 1989)

wave with significant wave height H_s and wave energy period T_e , the wave energy flux becomes $P = \frac{1}{64\pi} \rho g^2 T_e H_s^2$. For example, when $H_s = 2\text{m}$, $T_e = 10\text{s}$, we can get $P = 20\text{kW/m}$. It should be noted that the wave power is proportional to wave period and wave height squared. Although they are not sensitive as wind speed for wind power,

there exists a big difference in wave power for small and large waves. Figure 3 illustrates a map of global annual mean wave power (Mørk et al. 2008). This map indicates that the highest levels of average wave power are found in the areas between 40° and 60° in both hemispheres, which are consistent with the observations for global



Resource Assessment, Fig. 3 Global distribution of annual mean wave power (Mørk et al. 2008)

mean wind speed distribution as shown in Fig. 1. Like offshore wind resource, it is not possible to utilize all the wave energy in oceans based on the current technology and only the ocean areas close to the shore are relevant. High annual average wave power, typically 40–100 kW/m, is found along the coastlines of northern Europe, North America, South Africa, Australia/New Zealand, and Chile/Argentina.

Considering a site with annual average wave power of 40 kW/m and a device of 25% efficiency for electricity generation, the device needs to be at least 500 m long in order to produce 5 MW of electric power. This indicates that a large device is needed, which is particularly challenging when developing commercial wave energy converters since the cost will be high.

Wave power at a given offshore site also shows a variation in time. Since wave power is a function of both wave height and period, the long-term variation of wave power can be determined using the long-term joint distribution of significant wave height and spectral wave period (or wave energy period), as shown by Li et al. (2015). For a given wave energy converter device, the power absorption varies with respect to significant wave height and spectral wave period.

A combination of the wave power matrix of the device with the long-term distribution of significant wave height and spectral wave period can be used to derive the annual power output.

Global and regional distribution of wave power can be estimated based on wave buoy measurements, satellite remote data, or numerical models. The results in Fig. 3 were based on the data from the ECMWF (European Centre for Medium-Range Weather Forecasts) WAM numerical model, which is calibrated and corrected by Fugro OCEANOR against a global buoy and Topex satellite altimeter database (Mørk et al. 2008).

High-resolution numerical models are developed for resource assessment of a specific site, including the prediction of wave energy, heights, periods, and directions. Two types of approaches are used in these numerical models: (1) the deterministic approach, which describes the sea surface evolution accurately in both time and space; and (2) the spectral approach, which provides a statistical description of the wave conditions in space and time. The first approaches can capture all the relevant physics of wave/bottom/shoreline and nonlinear wave-wave interactions, but they are computationally expensive. The spectral

approaches provide an approximation of the physics of nonlinear wave-wave interaction, wind forcing, and wave breaking/bottom dissipation. But they are commonly used today and typically are open-source and user-friendly, such as MIKE21 (2008), SWAN (2009), TOMAWAC (2014). A comparison of the different numerical wave models was carried out and can be found in Venugopal et al. (2010).

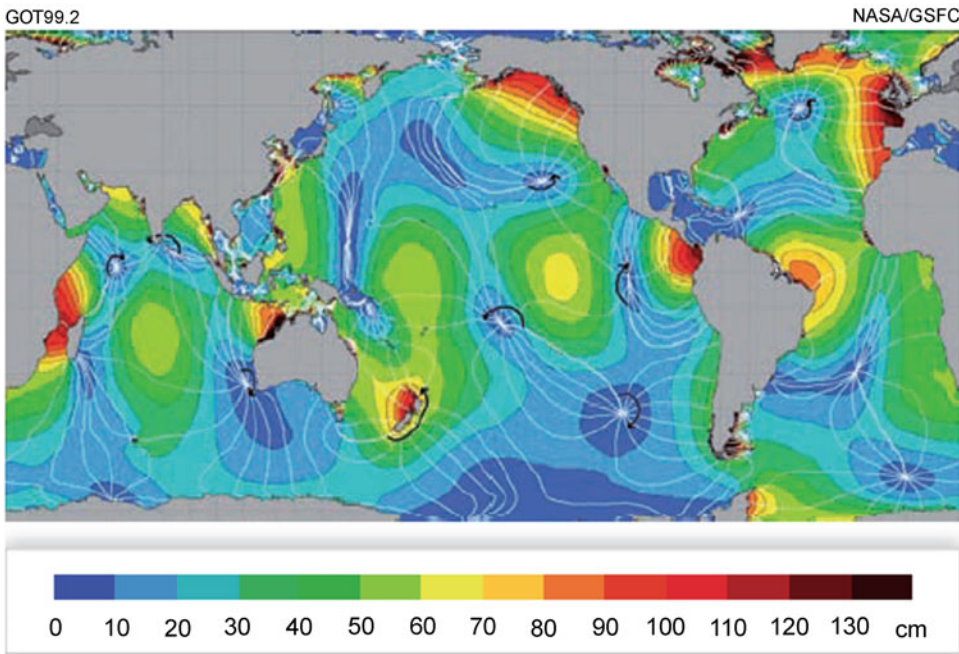
Ocean and Tidal Current Energy Resource

Ocean current might be generated due to many mechanisms, including tidal effect, ocean circulation, and wind generation. Here, we focus on tidal energy. Like offshore wind energy, ocean current energy refers to the kinetic energy in ocean current, which can be characterized using the current speed or velocity. However, since the density of sea water is much larger than that of air, one doesn't need a very large rotor area of a tidal turbine to achieve the same power production as compared to a wind turbine, although current

speed is typically much lower than wind speed. For example, considering a tidal current velocity of 3 m/s and 50% of efficiency of the turbine, a rotor with a diameter of 30 m (instead of 120 m for wind turbines at the rated wind speed of 12 m/s) can produce 5 MW of electric power.

Tidal energy resource can be very well predicted and shows a 12-h variation. It is strongly dependent on the tidal range at different locations. Figure 4 shows a global distribution of tidal range, which represents the tidal current speed and energy (NASA 2006). It shows that a strong location-dependent and places with abundant tidal energy are limited. In Europe, UK and Ireland have large resource, while some areas in the west coasts of Australia, New Zealand, Canada, and the east coast of the USA and China also have large potential to develop tidal energy.

Most of the tidal energy resource assessment is performed based on current speed measurements using for example Acoustic Doppler Current Profilers (Thomson et al. 2012) or Marine Radar (Bell et al. 2012), in particular in local straits or channels, where current speed can reach the minimum value for efficient use of tidal turbines and for



Resource Assessment, Fig. 4 Global distribution of tidal range (NASA 2006)

commercial development (Bahaj 2013). Numerical models based on 2D or 3D formulations have also been developed and used for resource assessment (Venugopal and Nimalidinne 2014). An offshore site with significant tidal energy resource often has large turbulence and large waves, which might be challenging for structural design of tidal turbines.

Cross-References

- [Offshore Wind Turbines](#)
- [Tidal and Ocean Current Turbines](#)
- [Wave Energy Converters](#)

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Responsibility

- [Reliability and Safety in Offshore Engineering](#)

Restoring Moment

- [AUV/ROV/HOV Stability](#)

Reynolds Number (Re)

- [Hydrodynamics for Subsea Systems](#)

Ribbon Bridge

- [Floating Bridge](#)

Rigid Connectors

- [Connectors of VLFS](#)

Rigid Module and Flexible Connector (RMFC)

► Connectors of VLFS

Risk-Based Design for Ship and Offshore Structures

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Synonyms

Reliability-based design

Definition

In engineering applications, as decision support tools, risk and reliability analysis methods are increasingly recognized worldwide. Risk-based design is a process integrating the risk and reliability analysis methods into product design. Risk-Based Ship and Offshore Structures Design introduces risk analysis into the traditional design process, aiming to deal with safety problems reasonably and effectively (Apostolos 2009).

Scientific Fundamentals

Principle of Risk-Based Design

The International Maritime Organization (IMO) has discussed the target based criteria and given rise to the term “Safety Level” designating the through-life level of acceptable risk associated with a particular ship concept, which may lead to a new guiding principle to attaining safety cost-effectively. The concept of “risk” is often associated with unexpected events and with shipping operations being undoubtedly “risky,” which

must be remembered when designing ship and offshore structures.

Effective risk management at the design stage is extremely important. At this stage, commencing with concept development or selection, there is the greatest scope to adopt or devise arrangements which will reduce, or even eliminate, particular risks. Risk assessment should be undertaken as an integral part of the design process, so that risk management decisions can be taken in good time. Much of the potential benefit of risk assessment of the design will be lost if it takes place after the design has been firmed up. It should not simply be used to retrospectively justify design decisions already taken.

Safety features incorporated into the design will be more secure, long lasting, and effective than those that rely upon the use of safety equipment or the adoption of operating procedures intended to compensate for design deficiencies. Furthermore, as the design and construction work progress, it will become increasingly difficult and costly to incorporate safety related features.

Safety considerations in design can be considered in the following hierarchy:

- (a) Eliminate the hazard
- (b) Reduce the likelihood of a hazardous event, giving preference to inherent safety over operational safeguards (e.g., procedures)
- (c) Reduce the consequences of a hazardous event, for example, by reducing combustible fuel inventory; by adopting fail-safe or fault tolerant systems; by protecting structure and equipment from fire.
- (d) Protect people from the effects; collective protection in preference to individual protection
- (e) Provide means of escape and evacuation

The application of risk-based design is more focused on the concept of high level innovation (shown in Fig. 1), so it is necessary to use knowledge of ship information in all its forms: best practice, engineering evaluation, latest tools and data, and all content of quantitative risk analysis.

Risk-based design is a comprehensive approach. The key to understanding this method

is the integration of risk assessment in the design process and decision-making towards achieving the overall design goals, as illustrated by framework of Fig. 2.

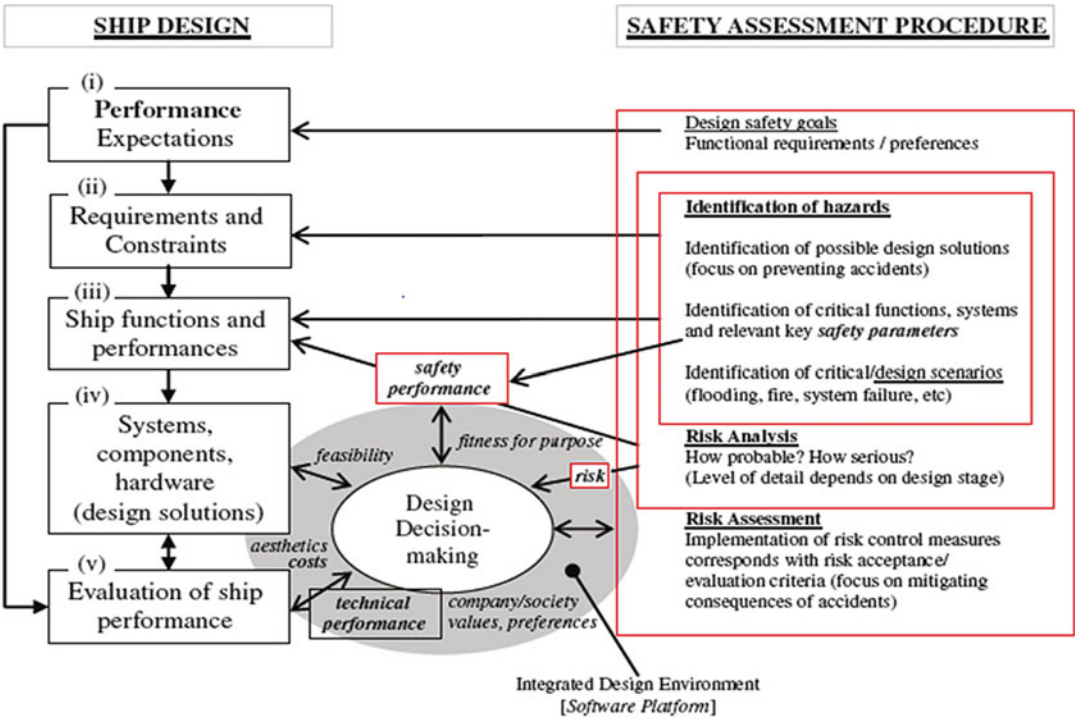
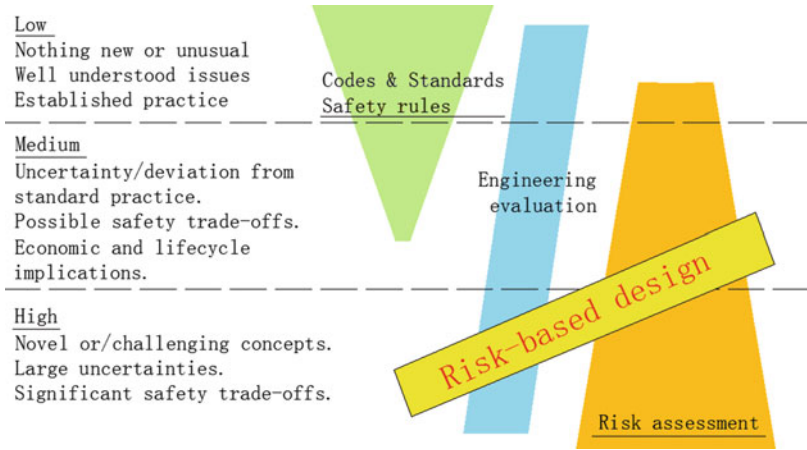
Safety Assessment

The hazards caused by risks of ships and offshore structures mainly include the following three

aspects: casualty, environment disruption, and economic loss (Zhang et al. 2003), shown in Fig. 3.

- (a) The risks of personnel include: potential casualties, fatal accident rates, average individual risk rates, etc.
- (b) The degree of damage to the environment can be measured according to its recovery time. It

Risk-Based Design for Ship and Offshore Structures, Fig. 1 Risk-based design and innovation (Apostolos 2009)



Risk-Based Design for Ship and Offshore Structures, Fig. 2 Framework for risk-based design (Apostolos 2009)



Risk-Based Design for Ship and Offshore Structures,
Fig. 3 Content of safety assessment (Zhang et al. 2003)

can be generally divided into: 1–12 months, 1–3 years, 3–10 years, greater than 10 years, etc.

- (c) Economic loss usually refers to material damage and production delay.

Security Goals

Security goals, like other design objectives, are related to the mission and purpose of ships. Clear security goals are already part and parcel of the design input. Examples of design objectives driven by safety factors include (Christian et al. 2012):

Top-Level Goals:

- (a) No accident can cause loss of the whole ship, such as collision, grounding, fire, etc. (Franz 2010).
- (b) No loss of human life due to ship related accidents.
- (c) Low impact to the environment (no air emissions, low noise).
- (d) When the ship has an accident, the impact will be minimized.

Specific Technical Goals:

- (a) Vessels remain upright and afloat in all feasible operational loads and environmental conditions.
- (b) Vessels remain upright and afloat in case of water ingress and flooding.
- (c) The ship structure can withstand all predictable loads throughout the life cycle.
- (d) Sufficient residual structural strength in the case of breakage.
- (e) High passenger comfort (no seasickness, low vibration level, low noise level).

In general, the ship can meet all kinds of requirements for its use.

Key Applications

Risk Measurement

The risk for ship and offshore structures can be expressed by probability and resulting value. In actual calculation, the basic expression of the risk is:

$$R = \sum_i p_i \times c_i$$

where p_i represents the frequency of occurrence of a single event and c_i represents the consequences of the event.

Design Phase Using Risk Criteria

When adopting risk based ship design method, different risk assessment criteria can be adopted in different design stages.

The feasibility design phase of ship engineering: usually, decisions need to be made under the premise of less information, such as project progress, cost budget, etc. Qualitative or semi quantitative risk assessment methods are mainly adopted. The risk matrix of qualitative assessment is shown in Fig.4.

The detailed design phase of ship engineering: with the clarity of design objectives and various design parameters and drawing on existing databases, probability based quantitative risk assessment methods can be adopted. At this point, the ALARP principle can be used for quantitative risk assessment of casualty risk, environmental risk, and property risk, as shown in Fig.5.

A qualitative ranking of both frequency (from frequent through to incredible) and severity (from catastrophic to negligible) of each hazard is deduced by consensus.

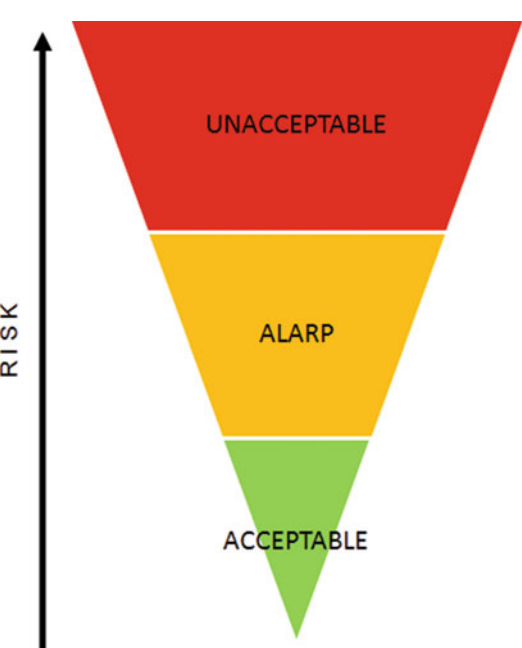
These can be amalgamated in matrix form where the risk categories are:

A = Intolerable

B = Undesirable and only accepted when risk reduction is impracticable

			Potential Consequences				
			L6	L5	L4	L3	L2
			Minor injuries or discomfort. No medical treatment or measureable physical effects.	Injuries or illness requiring medical treatment. Temporary impairment.	Injuries or illness requiring hospital admssion.	Injury or illness resulting in permanent impairment.	Fatality
			Not Significant	Minor	Moderate	Major	Severe
Like lihood	Expected to occur regularly under normal circumstances	Almost Certain	Medium	High	Very High	Very High	Very High
	Expected to occur at some time	Likely	Medium	High	High	Very High	Very High
	May occur at some time	Possible	Low	Medium	High	High	Very High
	Not likely to occur in normal circumstances	Unlikely	Low	Low	Medium	Medium	High
	Could happen, but probably never will	Rare	Low	Low	Low	Low	Medium

Risk-Based Design for Ship and Offshore Structures, Fig. 4 Risk matrix (<https://cn.bing.com/>)



Risk-Based Design for Ship and Offshore Structures, Fig. 5 ALARP principle (<https://cn.bing.com/>)

C = Tolerable with the endorsement of the project safety review committee
D = Tolerable subject to normal project review

Risk acceptance criteria may be employed in supporting a design for ship and offshore structures when either classification or statutory approval is sought (Ship Right Design and Construction 2018).

Linking Risk-Based Design and Acceptance

Risk-based design for Ship and Offshore Structures is considered an enhanced variant of the traditional design process, taking safety as an additional design objective. Hence, an additional constraint is entered into the design optimization, as shown below (Jan 2007):

$$R_{design} \leq R_{acceptable}$$

With R_{design} the risk of the considered ship or offshore structures and $R_{acceptable}$ the acceptable risk.

Usually, risk is determined by the frequency and consequences of events. Different risk categories need to be distinguished, such as human life, environment, or property.

The design risk R_{design} is typically the sum of some risks associated with different types of accidents, such as collision, fire, or grounding. With the help of risk model, such as fault tree and Bayesian networks, each partial risk can be calculated.

The acceptable risk $R_{acceptable}$ of human life and environmental protection is specified by the approval authority (flag state administration and/or classification society). There are two options for determining acceptable risk: relative or absolute. In the first case, a reference design is selected which complies with current rules. In the second case, IMO risk acceptance criteria are used or referenced.

Techniques of Quantitative Risk Assessment

Quantitative risk assessment is the term most often used to describe the use of statistical or more properly quantitative methods in risk-based design. Quantitative techniques widely used under the risk-based design for Ship and Offshore Structures include:

- (a) Failure Modes, Effects and Criticality Analysis (FMECA)
- (b) Fault Tree Analysis (FTA)
- (c) Event Tree Analysis (ETA)
- (d) Statistical analysis of historical accident data
- (e) Reliability analysis of component failure data
- (f) Data elucidation from structured expert judgment
- (g) Human error analysis
- (h) Cost Benefit Analysis (CBA)

Decision Making

So far the assessment and ranking of risk has been considered. The final stage involves hard decisions where the proposed safety improvements may be difficult and the cost to implement considerable.

The principle of As Low As Reasonably Practicable ALARP (see “[Design Phase Using Risk Criteria](#)”) shows that tolerable risk is not a simple

pass/fail test. The majority of risks are in the ALARP region where work should be done to reduce the risk further.

The owners of each risk must be satisfied that safety arguments, criteria, and decisions are sound and documented so as to provide an audit trail. Peer review and the use of independent safety assessment/audit are also valuable in giving strength to decisions and actions, particularly if later challenged.

By taking safety as the target in the design process, the new concept of “safety design” and the development of risk-based design method can deal with ship safety in a systematic and comprehensive way. Risk-based design opens the door to innovation and provides competitive advantages for the maritime industry by facilitating cost-effective safety. It is impossible to optimize the design solutions without risk-based design.

This method reflects the trend of target based standards and highlights the advantages of risk-based design methods. It is particularly useful in the face of innovative ship and offshore structures design concepts and alternative design and arrangements. In this case, quantitative risk analysis is the only reliable way to ensure a suitable safety level and set a safety target.

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RMRS, Russian Maritime Register of Shipping

► Moored Ship in Ice

Robust Control

► Intelligent Control Algorithms in Underwater Vehicles

Routing Layer

► Underwater Acoustic Sensor Network

ROV

► Remotely Operated Vehicle (ROV)

ROV Dynamic Positioning

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Synonyms

Dynamic positioning (DP); Dynamic positioning control; Dynamic positioning control system; Dynamic positioning of remotely operated vehicles; Dynamic positioning of underwater robotic vehicles; Dynamic positioning system

Definition

The remotely operated vehicles (ROVs) dynamic positioning (DP) is a method to automatically

maintain an ROV on its position and/or heading (fixed location or predefined track) by use of its thrusters. ROVs equipped with DP system allow operations close to the seafloor, including station keeping, trajectory tracking, path following, and low-speed maneuvering. Then the pilot can focus on monitoring and planning of operations that demand human intervention or decision making.

Scientific Fundamentals

Historical Development

DP system was born with the increasing demands of the rapidly expanding oil and gas exploration industry in the 1960s and early 1970s. In the 1960s, the first DP system was introduced for horizontal modes of motion (surge, sway, and yaw) using single-input single-output PID control algorithms in combination with low-pass and/or notch filter (Sørensen 2011). The first ship to fulfill the accepted definition of dynamic positioning was the “Eureka” (1961), as shown in Fig. 1, of about 450 t displacement and 130 ft. length. This ship was fitted with an analog control system, interfaced with a taut wire reference, equipped with steerable thrusters fore and aft in addition to her main propulsion (Fay 1989).



ROV Dynamic Positioning, Fig. 1 The world's first DP vessel Eureka (Steinbeck 2017)

DP system development came in the mid-1970 with the application of Kalman filters and linear quadratic (LQ) optimal controllers. After 1995, nonlinear PID control, passive observer design and observer back-stepping designs have been applied to DP (Fossen 2002). In 1980, the number of dynamic positioning capable ships was about 65, in 1985, the number had increased to about 150. Currently (2013), it stands at over 1,800 and is still expanding (► “Dynamic Positioning in Ice” from Wikipedia).

DP systems have become more sophisticated and complicated, as well as more reliable. The costs are falling due to newer and cheaper technologies, and the advantages are becoming more compelling as offshore work enters ever deeper water and the environment is given more respect. Computer technology has developed rapidly and some vessels have been upgraded twice with new DP control systems. Position reference systems and other peripherals are also improving and redundancy is provided on all vessels designed to conduct higher-risk operations.

DP of underwater vehicles like ROVs and autonomous underwater vehicles (AUVs) has lately received increasing interest from offshore contractors, vendors, and the research community.

DP Classification

Based on IMO – International Maritime Organization publication 645 (IMO 1994), the classification societies have issued rules for dynamically positioned ships described as Class 1, Class 2, and Class 3.

Equipment Class 1 has no redundancy. Loss of position may occur in the event of a single fault.

Equipment Class 2 has redundancy so that no single fault in an active system will cause the system to fail. Loss of position should not occur from a single fault of an active component or system such as generators, thruster, switchboards, remote controlled valves, etc. But the loss of position may occur after the failure of a static component such as cables, pipes, manual valves, etc.

Equipment Class 3 which also has to withstand fire or flood in any one compartment without the system failing. Loss of position should not occur from any single failure including a wholly burnt

fire subdivision or flooded watertight compartment.

Table 1 gives an overview of the IMO dynamic positioning system classification system and the roughly corresponding dynamic positioning system class notations, individual variations exist.

Elements of ROV DP System

A typical ROV DP system is shown in Fig. 2. The components and their relevance are explained below.

The controller computes control forces based on the error between the desired position and the actual position. The desired position is derived from guidance software. The actual position is taken from the navigation system. Control forces must be transformed to thruster commands by thrust allocation matrix (Fossen and Johansen 2006) and then send to thrusters.

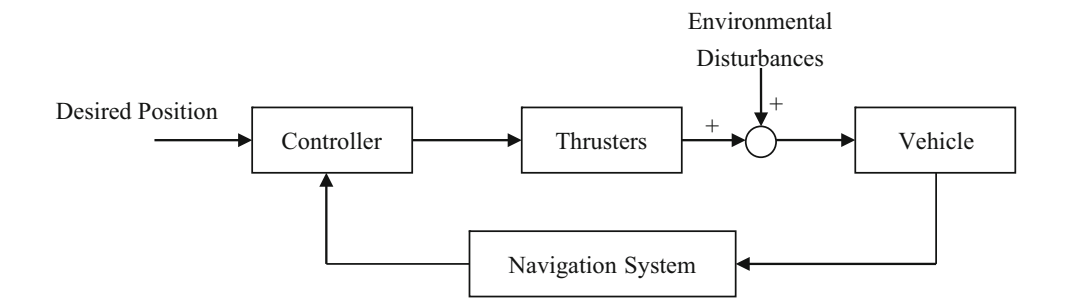
Thrusters provide DP capability for ROVs. Normally a DP ROV has an over-actuated vector thrust system. A single or multi failure of thrusters will not cause the system to fail according to the specific thruster configuration. Fault-tolerant control algorithms were proposed by researchers (Yang et al. 1998; Ni and Fuller 2003; Omerdic and Roberts 2004).

Navigation system indicates the position of ROVs. Ultra short base line (USBL), Doppler velocity log (DVL), or inertial measurement unit (IMU) is used to obtain position information. For some underwater application, integrated navigation combining the above sensors and pressure sensor, altimeter, digital compass, etc. is used to get a more accurate position. Raw data from sensors are processed in order to check for validity and consistency, and filtered by observers such as extended Kalman filter (EKF). The observer continues estimation of ROV status even if sensors fail to provide raw data, referred to as dead reckoning.

Environmental disturbances are mainly generated by ocean currents for ROV DP system. It's inconvenient to measure current forces. The traditional way is to design a robust controller or estimate current forces by an observer. Winds and waves can be neglected in underwater environment.

ROV Dynamic Positioning, Table 1 IMO DP classification

Description	IMO	Corresponding class notations					
	DP Class	ABS	LRS	DNV	GL	NK	BV
Manual position control and automatic heading control under specified maximum environmental conditions		DPS-0	DP (CM)	DPS0 DYNPOS-ATUS			
Automatic and manual position and heading control under specified maximum environmental conditions	Class1	DPS-1	DP (AM)	DPS 1 DYNPOS-AUT	DP1	DPS A	DYNAPOS AM/AT
Automatic and manual position and heading control under specified maximum environmental conditions, during and following any single fault excluding loss of a compartment (two independent computer systems)	Class2	DPS-2	DP (AA)	DPS 2 DYNPOS-AUTR	DP2	DPS B	DYNAPOS AM/AT R
Automatic and manual position and heading control under specified maximum environmental conditions, during and following any single fault including loss of a compartment due to fire or flood. (At least two independent computer systems with a separate backup system separated by A60 class division)	Class3	DPS-3	DP (AAA)	DPS 3 DYNPOS-AUTRO	DP3	DPS C	DYNAPOS AM/AT RS



ROV Dynamic Positioning, Fig. 2 Block diagram of ROV DP system

The desired position is derived from guidance software. Guidance software includes set-point regulation, trajectory tracking, and path following modules to accomplish different operations. Guidance software continuously computes the desired position according to predefined track or path. Guidance software can also take inputs from joysticks directly.

The ROV DP system comparing with ship DP system is listed in Table 2.

Scientific Problems to be Solved

It's hard to obtain a precise mathematical model of ROVs. The traditional way of modeling an ROV

is by computational fluid dynamics (CFD) and scaled model test. The mathematical model is just approximation in typical working condition.

Environmental disturbances are hard to be measured. There exists error in the estimation of environmental forces by observer. The error can be even more significant without a precise model of ROV. Besides, the forces from umbilical cable grow with water depth as a result of increasing cable length. Proper ways to measure or estimate disturbances need to be found.

ROV DP system needs to keep the station in both horizontal and vertical plane simultaneously, increasing complexity of the system. Robust

controller with good performance in hazardous underwater environment remains in laboratory. Performance of controller is affected by model accuracy, sensor configuration, and sea condition in practical application.

Application

Researchers from the Norwegian University of Science and Technology presented a dynamic positioning system for a small size ROV called Minerva in 2011 (Dukan et al. 2011), as shown in Fig. 3. Minerva is equipped with two horizontal

thrusters, two vertical thrusters, and one lateral thruster, accomplishing six degrees of motion. Navigation sensors consist of fluxgate compass, CRS03 silicon rate sensor, Teledyne RDI Workhorse Doppler velocity log (DVL), Kongsberg MRU6, and Kongsberg HiPAP500. The National Instruments (NI) CompactRIO (cRIO) programmed with LabVIEW is used for the implementation and development of the DP controller. Kalman filter (KF) and extended Kalman filter (EKF) are used as observer.

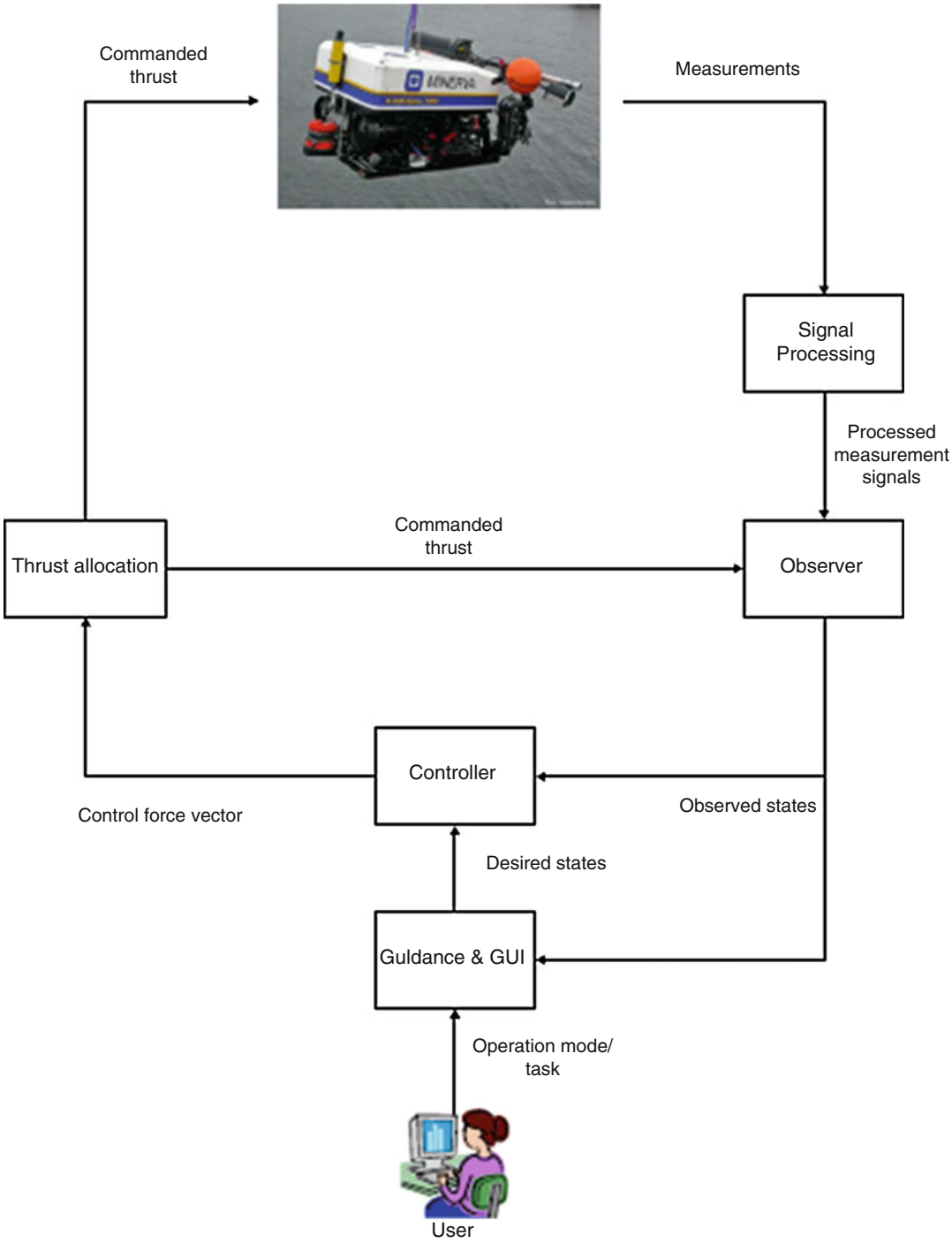
The DP system of Minerva consists of signal processing, observer, controller, guidance system, and thrust allocation modules as seen in Fig. 4.

ROV Dynamic Positioning, Table 2 Comparison with dynamic positioning of ships

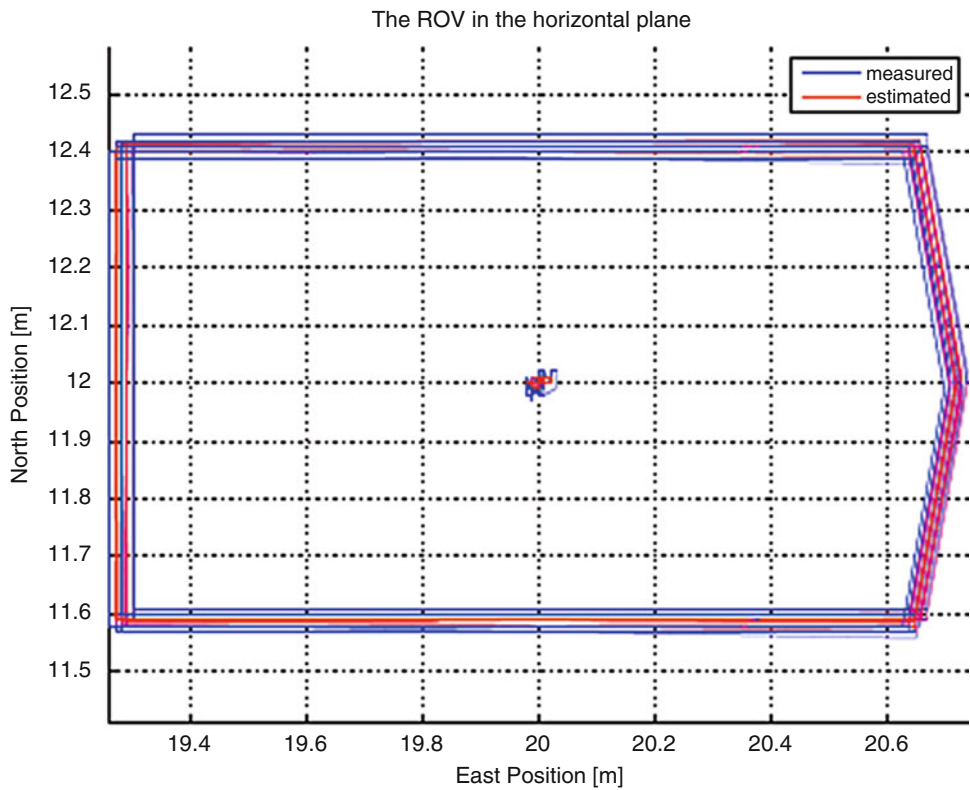
	ROV DP	Ships DP
Control parameter	Horizontal position Vertical position Heading	Horizontal position Heading
Environmental disturbance	Ocean current	Wind Wave Ocean current
Navigation sensor	DVL USBL IMU Digital compass Pressure sensor	DGPS Gyrocompasses Environmental sensor
Propulsion	Thrusters	Main propellers Tunnel thrusters Azimuth thrusters

ROV Dynamic Positioning,
Fig. 3 Minerva ROV
(Dukan et al. 2011)



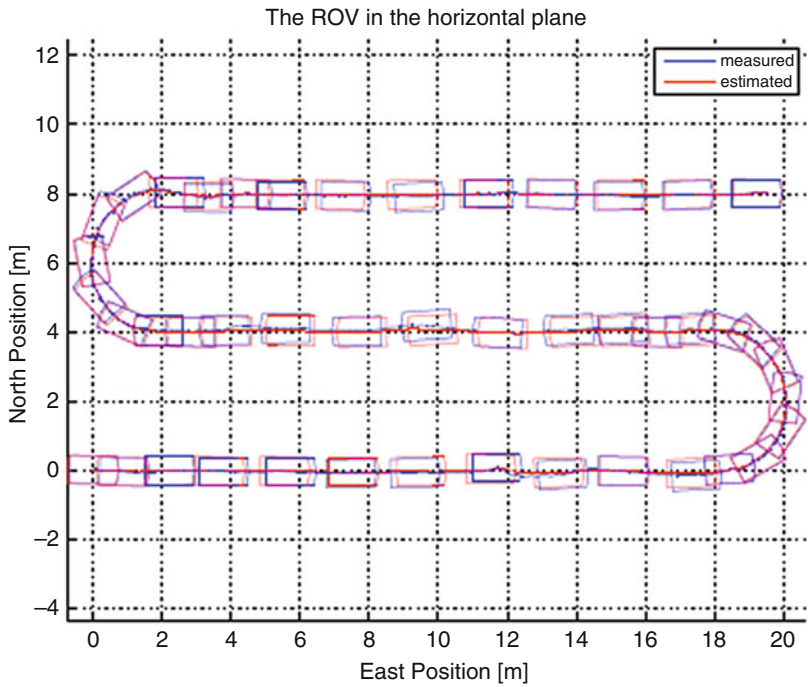


ROV Dynamic Positioning, Fig. 4 DP system of Minerva (Dukan et al. 2011)



ROV Dynamic Positioning, Fig. 5 Station keeping test of Minerva ROV DP system (Dukan et al. 2011)

ROV Dynamic Positioning, Fig. 6 Trajectory tracking test of Minerva ROV DP system (Dukan et al. 2011)



The signal processing module gathers signals from all sensors and checks validation of each signal data frame. Valid signals are passed to the observer module. Invalid signals are discarded and an abortion message is sent to observer module. The observer takes inputs from signal processing module and thrust allocation module. The outputs are estimated states of ROV. The controller module computes control force vector based on error between the desired states and estimated states. The thrust allocation module transforms control force vector to rotation speed of each thruster. The guidance module generates desired states for station keeping, trajectory tracking or path following.

Experimental results proved the validity and performance of the DP system. Result of the ROV DP system in station keeping test is shown in Fig. 5. The maximum measured deviation from desired position was 2 cm in North and 3 cm in East direction. The estimated deviation was within 2 cm in both North and East directions.

The results of trajectory tracking test are shown in Fig. 6. Maximum estimated cross-track error is 9 cm and measured is 20 cm.

Cross-References

- [Dynamic Positioning in Ice](#)
- [Integrated Navigation](#)
- [Remotely Operated Vehicle](#)

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Safety of Offshore Platforms

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Definition

The *safety* culture in the ocean engineering covers the safety of personnel, platform, and environment, and it is referred to the control of recognized hazards in order to achieve an acceptable level of risk.

Scientific Fundamentals

Introduction

In ocean engineering, the offshore engineering oil and gas industry is usually regarded as a high-risk sector, where the workers face not only process hazards associated with exploration, storage, and processing of hydrocarbons on the platforms but also other forms of hazards related to the harsh working environment and transportation (Broni-Bediako and Amorin 2010; Tang et al. 2017). It is generally agreed that the safety of offshore engineering can be grouped into two main categories, namely, personal safety and process safety (Swuste et al. 2016; Tang et al. 2018).

The personal safety emphasizes the health and safety of individual employees via minimization

of their exposure to radiations, chemical, noise, extreme vibration, abnormal temperatures, and mechanical, electrical, and ergonomic hazard with the measures such as industrial hygiene monitoring, chemical health risk assessment, medical surveillance, safety awareness program, and the work arrangement to reduce the fatigue of employee (International Labour Organization 2001; Mearns et al. 2003; Mearns and Hope 2005; Venkataraman 2008).

The process safety primarily concerns the hazards of offshore engineering related to the spills, fires, explosions, instability, capsizing, and destruction, which cause not only injuries and fatalities but property and environmental damage (Knegtering and Pasman 2009; Swuste et al. 2016). In many instances, the term process safety is usually interchangeable with asset integrity, where the asset integrity management usually covers the entire life cycle of an asset from design to decommissioning and includes management of processes, resources, people, and system (Hassan and Khan 2012).

The consequences of the process safety are usually potentially involving multiple injuries and fatalities, and they are more severe than those of the personal safety (Knegtering and Pasman 2009).

Typical Safety Issues in Offshore Engineering

Safety can be regarded as the absence of accidents or failures. The insight about safety features can be gained from detailed information of accidents

and failures. In the following, several typical catastrophic accidents in offshore engineering, which caused not only major economic and life losses but also serious environmental damage, are provided.

Structural Failure Due to the Extreme Environmental Loadings

Although the extreme environment condition occurs quite rarely during the life of offshore structure, the environmental loadings accompanied with this environment condition may exceed the design loading of offshore structure, and this may lead the structure into failure. As one of the extreme environment conditions, the hurricane with high storm surge and waves, heavy rains, and strong winds may have significant influence on the offshore structure.

The Hurricane Katrina entered the GOM on August 26, 2005. Prior to hurricane landfall, more than 700 platforms and rigs were evacuated making more than 90% of the oil and 83% of gas production in GOM offline (Feltus 2005). Statistical results have shown that the Hurricane Katrina completely destroyed 44 platforms and severely damaged 21 others (MMS 2006), including some large production facilities. The number of destroyed and severely damaged offshore platforms with different water depth is listed in Tables 1 and 2, respectively (Cruz and Krausmann 2008).

In addition, the drilling rigs were also affected. The hurricane destroyed four drilling rigs and severely damaged nine others, and six drilling rigs, including semisubmersible and jack-up units, were set adrift (Ghonheim and Colby 2005; Tubb 2005). In one case, a drilling rig (ENSCO 7500) was moved 193 km south of Louisiana, and Diamond's Ocean Warwick (75 m jack-up) was carried 106 km by the storm and washed up in Dauphin Island, Alabama. Furthermore, about 100 pipelines were damaged, and 36 of them were 0.254 m diameter or larger, and there were 211 minor pollution incidents of 500 barrels of oil or less in the outer continental shelf due to Katrina (MMS 2006).

In the same year, the Hurricane Rita entered the GOM on September 25, and it destroyed

Safety of Offshore Platforms, Table 1 Number of platforms destroyed by hurricanes Katrina and Rita

	Hurricane Katrina	Hurricane Rita
Water depth (m)	Number of destroyed	Number of destroyed
$d < 30$	16	30
$30 < d < 60$	14	27
$60 < d < 120$	14	11

Safety of Offshore Platforms, Table 2 Number of platforms damaged by hurricanes Katrina and Rita

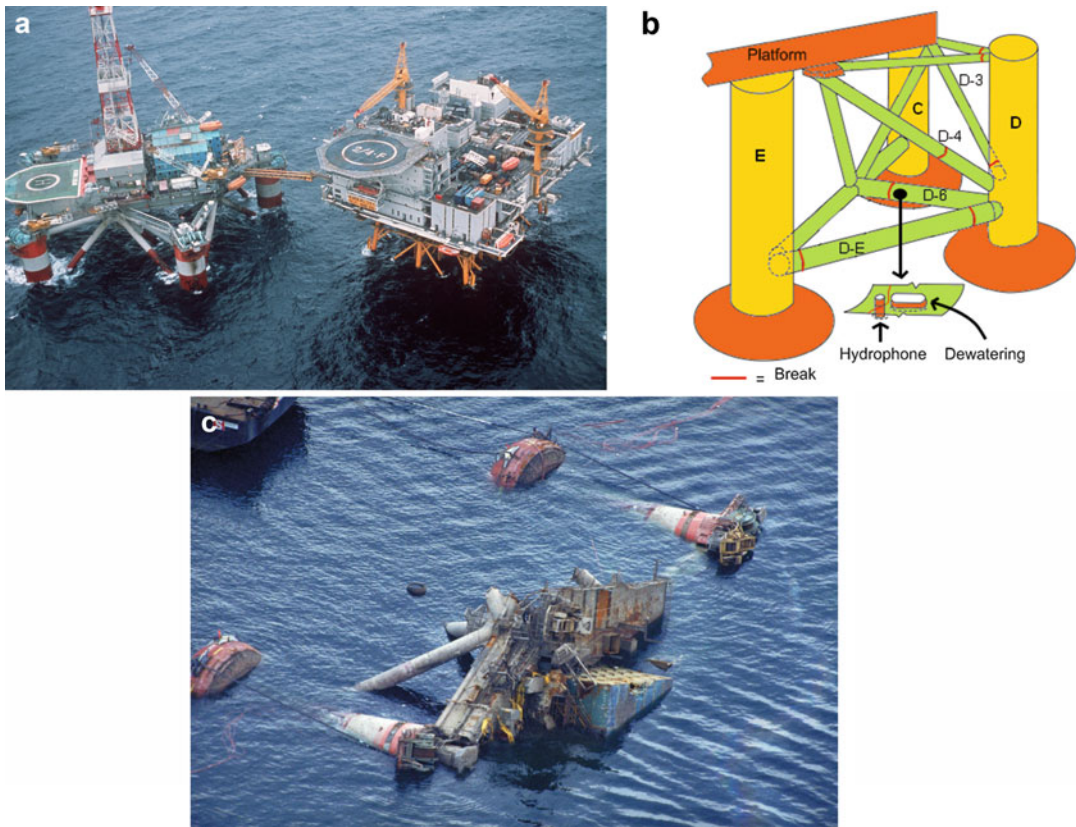
	Hurricane Katrina	Hurricane Rita
Water depth (m)	Number of destroyed	Number of destroyed
$d < 60$	6	7
$60 < d < 120$	6	18
$120 < d < 240$	5	5
$300 < d < 1000$	4	2

69 platforms and 1 rig. There were 32 platforms severely damaged, and 13 rigs were set adrift by the Hurricane Rita (Djamarani 2005; MMS 2006). As was the case during Hurricane Katrina, the pipelines were also affected by Rita. The MMS reported that there were about 83 pipelines damaged, and 28 of them were 0.254 m diameter or larger, and 207 minor pollution incidents of 500 barrels of oil or less in the outer continental shelf due to Hurricane Rita (MMS 2006).

More than 2000 people lost their life in hurricanes Katrina and Rita. The direct economic losses were about \$70–130 billion, and the reconstruction costs were expected to reach about \$200 billion. Following the hurricanes, changes have been proposed to operating and emergency procedures, maintenance requirements, and design practices including mooring practices for mobile offshore drilling units (Cruz and Krausmann 2008).

Structural Capsizing Due to Fatigue Failure

The offshore structures are exposed to multiple sea states during their service life, which may cause seriously accumulative fatigue damage



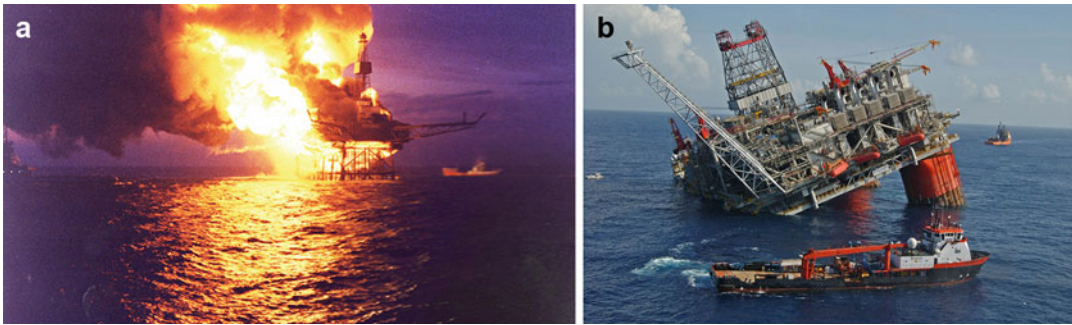
Safety of Offshore Platforms, Fig. 1 Alexander L. Kielland platform capsized in 1980 due to the fatigue crack (Norwegian Petroleum Museum)

and lead the offshore structures to fatigue failure even if the stresses in the critical regions are smaller than the elastic limit (Jia 2008; Song and Wang 2019). The fatigue damage becomes a major cause of damages that occurred on the offshore structures.

As illustrated in Fig. 1a, the Alexander L. Kielland platform capsized while working in the Ekofisk oil field in March 27, 1980, and there were about 123 crews killed, making the capsize the worst disaster in Norwegian waters since World War II (Bunn 2018). The accident started with one of the bracings (D-6) failing due to fatigue, thereby causing a succession of failures of all bracings attached to this leg (see Fig. 1b). It was discovered during the investigation that the weld of an instrument connection on the bracing

had contained cracks, which had probably been in existence since rig was built. The cracks had developed over time, and the remaining steel was less than 50%, and the strength of the bracing did not meet the serviceability requirement of the platform.

When the leg came loose, the rig almost immediately developed a severe listing. Within 20 min of the initial failure, it capsized completely, floating upside down with just the bottom of the columns visible in the sea (see Fig. 1c). Both the escape and evacuation operations were far away from orderly and had only limited success. Only one lifeboat was in fact launched successfully, one was totally unavailable due to the listing, and others were smashed against the platform during launching in high waves.



Safety of Offshore Platforms, Fig. 2 Platforms destroyed by explosions and fires. (a) Piper Alpha platform accident in 1988, (b) P-36 semisubmersible platform accident in 2001 (Moan 2005)

Structural Destruction Due to Explosion and Fire
Due to the operational errors or technical faults, the oil and gas leaked from the equipment may be ignited, and the offshore structure can be destroyed by the explosions and fires (Moan 2005).

The Piper Alpha platform got explosions and fires in July 6, 1988, and it was one of the costliest man-made catastrophes ever occurred either onshore or offshore (Ian 2012). The explosions and fires were initiated by a gas leak from the blind flange of a condensation pump that was under maintenance but not adequately shut down (PA 1990). As presented in Fig. 2a, the Piper Alpha platform was completely destroyed by the explosions and fires. The accidents resulted in the loss of 167 crew members, including 2 crewmen of a rescue vessel, and the economic loss of about \$ 3.4 billion. The investigative report pointed out that the main issue that caused the initiation of this accident was the lack of communication between the maintenance team and the control room operations.

The Petrobras 36 (P-36), the largest floating semisubmersible platform in the world in that time, experienced two explosions in March 15, 2001. The accident was initiated by the rupture of the emergency drain tank in the starboard aft column due to excessive pressure, and the rupture caused damage to various equipment and installations, leading to the flooding of water, oil, and gas into the column. After 17 min, the dispersed gas caused explosion and fire. At the time there are 175 people on the

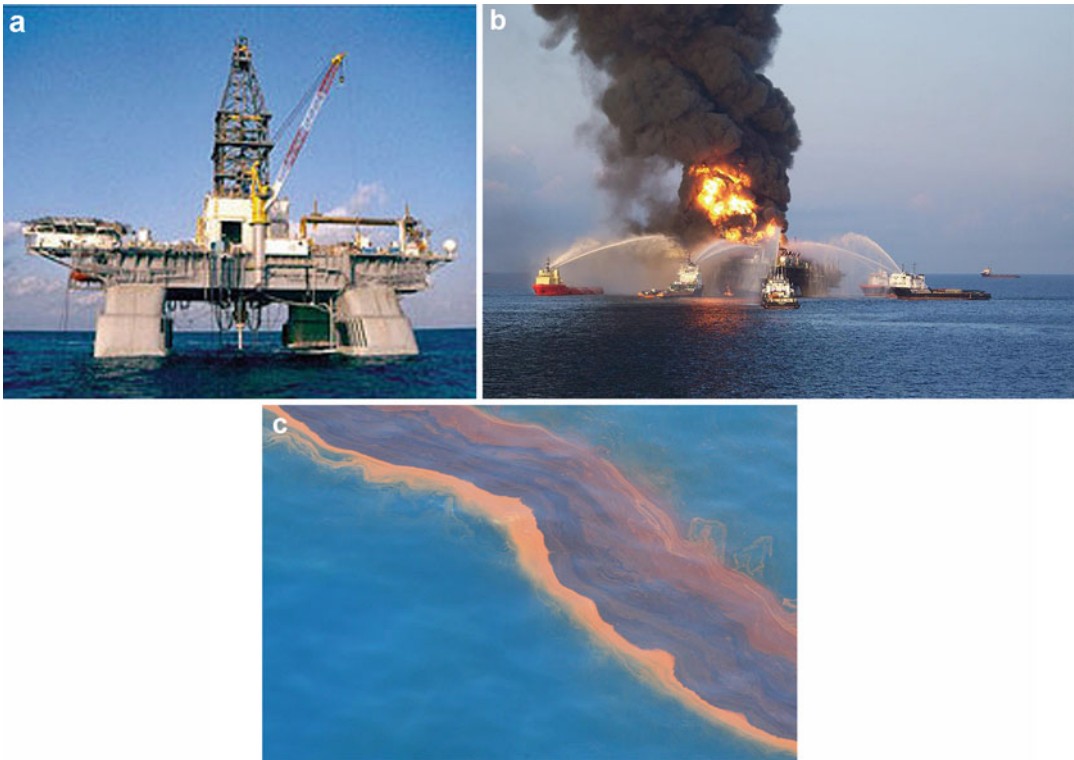
platform and 11 were killed. The platform sank 5 days after the explosion with an estimated 1700 t of crude oil remaining on board (Videiro et al. 2002). A series of operational errors were identified as the main cause of the first explosion and also the sinking.

Environmental Damage Resulted from Oil Spill

The oil spills accompanied with offshore engineering accidents have significant influence on the environment and ecosystem. In April 20, 2010, the Deepwater Horizon platform, which drilled the deepest oil well in history at a vertical depth of 10,683 m (see Fig. 3a), exploded and caught fire due to ignition of the high-pressure methane gas from the well (Schwartz and Weber 2010). There were 126 crew members on board when the platform caught fire, and 11 missing workers were believed to be dead from the explosion (see Fig. 3b).

The oil flowed for 87 days, and the total estimated volume of leaked oil approximated 4.9 million barrels with plus or minus 10% uncertainty, making it the world's largest accidental spill (Hoch 2010). According to the satellite images, the spill directly affected 68,000 square miles of the ocean (see Fig. 3c) and had affected 8332 species, including more than 1270 fish, 604 polychaetes, 218 birds, 1456 mollusks, 1503 crustaceans, 4 sea turtles, and 29 marine mammals (Biello 2010; Norse and Amos 2013).

Research found that the toxins from oil spills can cause irregular heartbeats leading to cardiac arrest and the tuna and amberjack that were



Safety of Offshore Platforms, Fig. 3 Deepwater Horizon oil spill (<https://en.wikipedia.org>)

exposed to oil from the spill developed deformities of the heart and other organs that would be expected to be fatal or at least life-shortening (Kerr 2010). In addition, the spill had a strong economic impact to BP and also the Gulf Coast's economy sectors such as offshore drilling, fishing, and tourism. Estimates of lost tourism dollars were projected to cost the Gulf coastal economy up to 22.7 billion through 2017. The GOM commercial fishing industry was estimated to have lost \$ 247 million as a result of post-spill fisheries closures (Sumaila et al. 2012). BP may be required to pay as high as \$ 90 billion.

Key Applications

Safety Design

Design Criteria

Safety design is referred as the design that effectively minimizes the likelihood of accidents/

damages and mitigates their consequences, which has long been a priority in the offshore engineering industries. As uncertainties in the structural resistance and the environmental loads to the structures for all kinds of reasons, the design approaches can be grouped into two main categories, namely the deterministic design approach and the probability design approach.

Deterministic Design Method

The deterministic design method, which is also known as working stress design (WSD) method, assumes that all kinds of environmental loadings are combined, and it may fit a certain probability distribution, and the resistance of structure is deterministic. In the design process, the stress of the structure under different environmental loadings should be less than the allowable stress, and a global safety factor is used to quantifying uncertainties of environmental load and structural capacity.

The deterministic design method is mainly considered as applicable to special case design problems, to calibrate the usage factors to be used in the WSD method and for conditions where limited experience exists. The detail of the deterministic design method can be found in API RP 2A (2000) for fixed platform and DNV-OS-C201 (2015) for floating platform.

Probabilistic Design Method

The probabilistic design method which is also known as load and resistance factor design (LRFD) method assumes that all kinds of environmental loads fit different probability distributions and the structural resistance/capacity is probabilistic and also obeys a certain distribution function as well. According to the probabilistic design method, the target safety level of the structure is obtained as closely as possible by applying load and resistance factors to characteristic reference values of the basic variables.

The basic variables are defined as loads acting on the structure and the resistance of the structure or the resistance of materials in the structure. The target safety level is achieved by using deterministic factors representing the variation in load and resistance and the reduced probabilities that various loads will act simultaneously at their characteristic values. The details of the probabilistic design method can be found in DNV-OS-C101 (2011).

Safety Assessment and Management

General

The offshore platforms are systems consisting of structures, equipment, and other hardware, as well as specified operational procedures and operational personnel. Ideally, the system should be designed and operated to comply with a certain acceptable risk level as specified. The safety assessment and management is the process to ensure that the risks of offshore platform are reduced to a level as low as reasonably practicable via hazard identification, risk assessment, and monitoring (Gupta and Edwards 2002). Typical elements of a safety assessment and management process consist of data collection and

management, risk assessment, inspection and monitoring plan, and strengthening and mitigation measures (International Labour Organization 2001; Li et al. 2012).

Data Collection and Management

The data of offshore structure is the foundation of safety assessment and management process, and it can be grouped into two main categories, namely, characteristic data and condition data.

1. Characteristic data: the data represents the structure's characteristic in initial condition, and it relates to the structure design, fabrication, transportation, and installation process, including the design drawing, analysis results, fabrication planning, transportation and installation report, and other related files.
2. Condition data: the data represents the structure's characteristics in operation condition, including the inspection data, monitoring data, maintenance and repair report, etc.
3. An effective data management system should be established to save, modify, and update the related data.

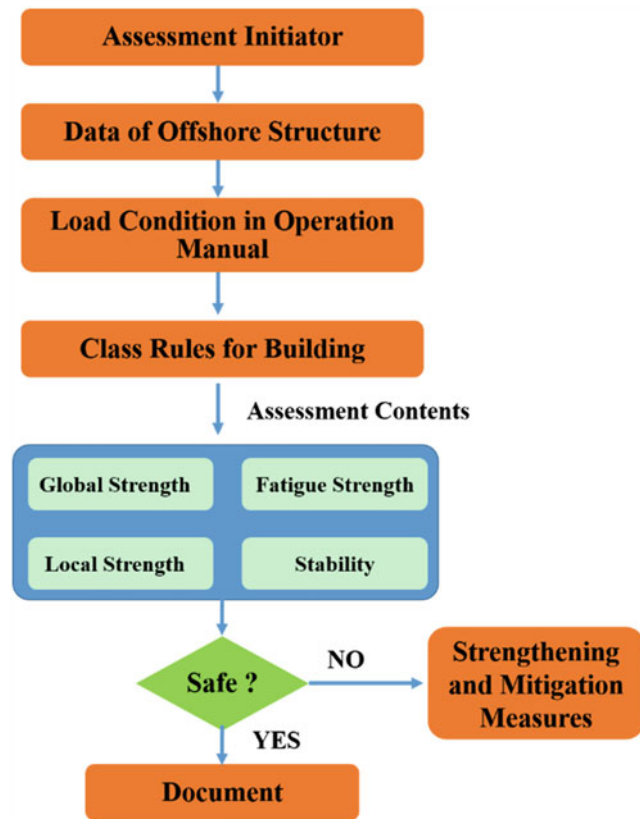
Safety Assessment

The safety assessment of offshore platforms during operation is necessary in connection with a planned change of platform function; extension of service life; occurrence of overload damage due to hurricanes, explosion, fires, and ship impact; updating of inspection plans; etc. (Moan 2005). The safety assessment should be based on updated data of offshore structure, and the corresponding results can be represented by the safety indicators, which can measure the safety performance of the offshore structure and provide input for the planning of inspection, monitoring, maintenance, and repairs. The procedure of safety assessment is illustrated in Fig. 4.

Inspection and Monitoring Planning

Inspection and monitoring are important measures for maintaining safety, especially with respect to fatigue, corrosion, and other deterioration phenomena. The inspection and monitoring data of a

Safety of Offshore Platforms, Fig. 4 The procedures of the safety assessment (Li et al. 2012)



given structure is actively incorporated in the planning of future activities (Moan 2005).

The inspection planning is usually formulated based on the results of safety assessment, which is the overall and strategic arrangement of platform inspection, including the inspection intervals and scope. Typically, the major inspections of offshore structures are carried every 4–5 years, while intermediate and annual inspections are normally less extensive. The optimal inspection interval can be determined with the probabilistic methods.

The following items should be included in the inspection planning:

1. Prioritizing which locations are to be inspected.
2. Selection inspection method (visual inspection, magnet particle inspection, eddy current) depending on the damage of concern.
3. Scheduling inspections.

4. Establishing a repair strategy (size of damage to be repaired, repair method, and time aspects of repair).

Strengthening and Mitigation Measures

If the safety of offshore structure did not meet the serviceability and safety requirements, some strengthening and mitigation measures should be carried out to strengthen and mitigate the structure. The permanent repairs are made by cutting out component and welding a new component, rewelding, and adding or removing scantlings, brackets, stiffeners, lugs, or collar plates.

However, the strengthening and mitigation measures are usually very expensive. The most relevant measure to alleviate the intensity of repair could be achieved by varying the inspection interval or improving the inspection technology. Many approaches regarding the repair cost as a criterion in reliability-based inspection, monitoring, maintenance, and repair planning have been developed

(Garbatov and Guedes Soares 2001; Moan 2005; Dong and Frangopol 2015).

It should be mentioned that the risk can never be absence, and the implementation of the safety assessment and management never ends, and it should be carried out through the life cycle of offshore structure.

Conclusions

The offshore engineering oil and gas industry has large investment and long-period and high-risk features, and the failure of the offshore facilities usually resulted in the loss of economy and life and subsequent pollution of the environment. Safety is a very important characteristic of the offshore engineering oil and gas industry, and it can be ensured by improving the design method and reducing the human and organizational errors and omissions.

The safety assessment and management process is a scientific management strategy to identify, measure, and analyze risk and, on this basis to deal effectively with risk, to achieve maximum security at minimum cost. In the future, the safety assessment and management process can be established with multiple data sources and advanced methods to account for complex interactions between the natural environment, the development of naval technology, and human behavior.

Cross-References

- [Dynamic Behavior and Fatigue](#)
- [Reliability and Safety in Offshore Engineering](#)

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Sagnac Interferometer (SI)

- [Fiber Optic Hydrophone](#)

Salinity

- [AUV/ROV/HOV Hydrostatics](#)

Salinity Gradient Power Conversion

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Synonyms

SGPC - salinity gradient power conversion; PRO - pressure-retarded osmosis; RED - reverse electro-dialysis; AEM - anion-exchange membrane; CEM - cation-exchange membrane

Definition

Salinity gradient power is the power available from the difference in the salt concentration between sea water and river fresh water and it is often called osmotic power or blue energy. This form of power is available when two solutions with different salinity levels are mixed together. Salinity Gradient Power Conversion (SGPC) is a technology that uses the osmotic pressure, i.e., the chemical potential of concentrated and dilute solutions of salt, to drive a turbine to generate electricity.

Salinity Gradient Power Resource

In order to utilize the salinity gradient power, a large amount of both sea water and fresh water are needed. River mouths, where fresh river water discharges to the sea, are naturally potential areas for SGPC development and are widely distributed around the world (Lewis et al. 2011). The estimated technical potential for SGPC is about 1650 TWh/year (Scråmestø et al. 2009).

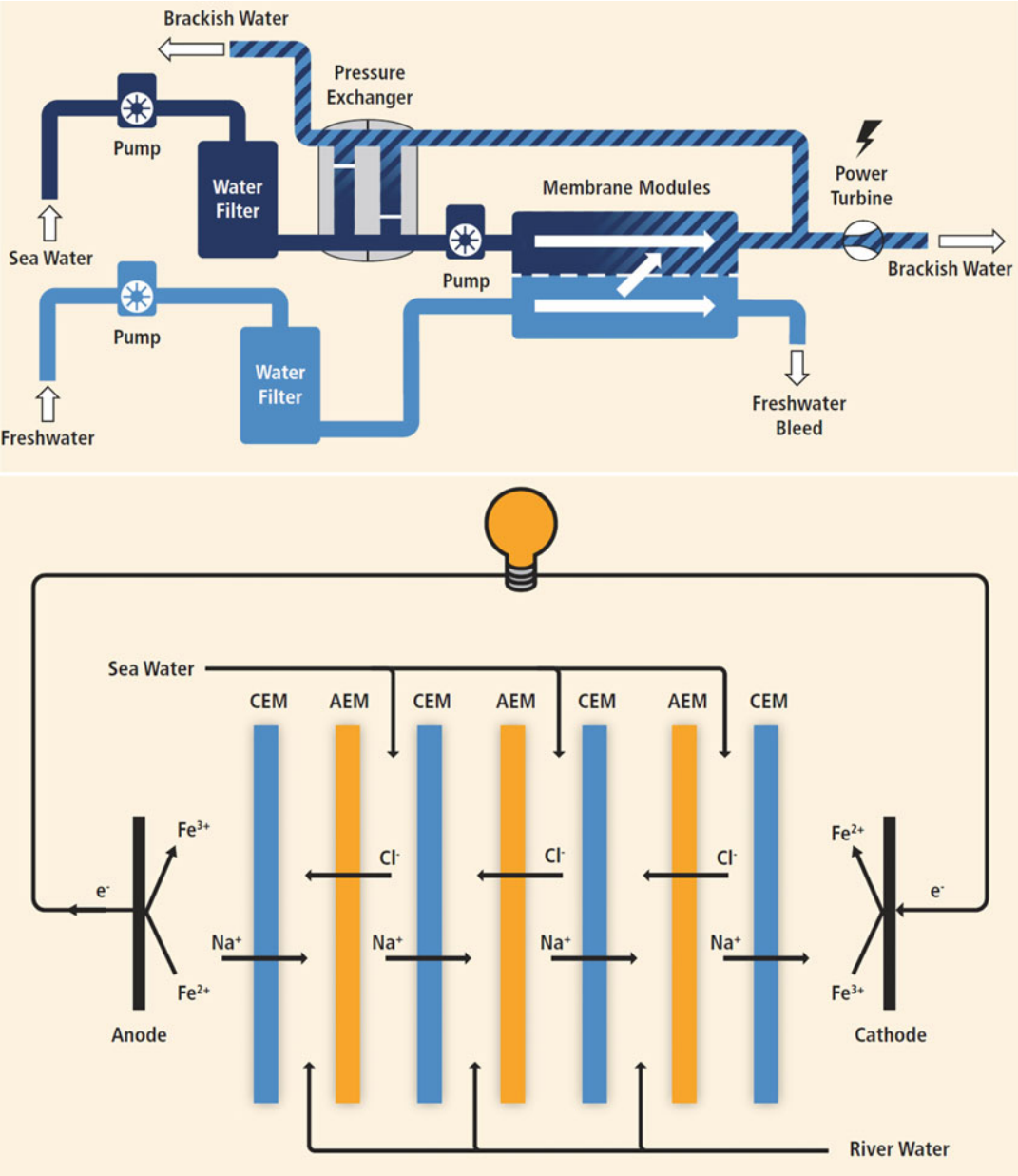
Technology for Salinity Gradient Power Conversion

Like ocean thermal energy, salinity gradient power is less developed than any other ocean

energy. No commercial plant based on salinity gradient power and only a few pilot plants exist today.

From the technology point of view, there are two types of SGPC, namely, Pressure-Retarded Osmosis (PRO) and Reverse Electrodialysis (RED), as shown in Fig. 1. In a PRO plant

(Scråmestø et al. 2009), both fresh water and sea water are pumped in two different loops to a chamber where semi-permeable membranes are placed between them and permit only water molecules to pass through. Water flows from the dilute solution (fresh water) to the concentrated solution (sea water) to bring the chemical potentials on



Salinity Gradient Power Conversion, Fig. 1 Illustration of PRO (left, Scråmestø et al. 2009) and RED (right, van den Ende and Groeman 2007) for ocean salinity gradient conversion

both sides of the membranes to an equilibrium, which will lead to high pressure on the sea water side and then drive a turbine to generate electricity. The osmotic pressure between sea water and fresh water is in the order of 2.4–2.6 MPa. The sea water is pressurized to about half of the osmotic pressure, before pumped into the chamber. As long as the pressure difference at the membranes between the two fluids is less than the osmotic pressure, the flows from fresh water to sea water continue. The remaining fresh water and the mixed fresh and sea water are then discharged to the river or the sea.

The RED approach (van den Ende and Groeman 2007) is essentially to create a salt battery, using ion-exchange membranes, which either let negatively charged ions in anion-exchange membranes (AEM) or positively charged ions in cation-exchange membranes (CEM) pass through. These AEM and CEM membranes are alternately stacked with water-based solutions of different concentrations in between to build up a considerable electrical potential that can be used for electricity generation.

Development and Challenges for Salinity Gradient Power Conversion

The concepts of PRO and RED have been proposed in 1970s. However, it is in 2009 that the first PRO power plant with a nominal power of 10 kW was built and started operation by the Norwegian company Statkraft. It uses about 2000 m² of membranes to eventually generate 5 kW of electric power (IRENA 2014). In 2014, the first RED plant with a nominal power of 50 kW was built at Afsluitdijk, the Netherlands, using the fresh water from Lake IJssel and the sea water of the North Sea (Schaetzle and Buisman 2015).

The developed prototypes of SGPC devices are very small, with a cost not competitive to other forms of ocean renewable energies. There exist many technical challenges for SGPC devices (Schaetzle and Buisman 2015). Both PRO and RED need a large amount of fresh and sea water, which should be pumped into specific chambers. This requires a large facility and increases the

overall cost. Both PRO and RED plants use membranes, which are very sensitive to fouling, i.e., the accumulation of minerals or marine growth on the membranes and require very clean water. This is a challenge in real world. In the case of a RED plant, the ion composition of the real fresh and sea water is much more complex than that is assumed in the lab tests with simple NaCl solutions. This will make the efficiency of a RED plant lower.

Cross-References

- [Economic Assessment](#)
- [Ocean Thermal Energy Conversion](#)
- [Power Take-Off System](#)

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SCF (Single Column Floater)

- [SPAR Platform](#)

Scouring

- [Iceberg Scouring](#)

SCR (Steel Catenary Riser)

► [SPAR Platform](#)

Screen-Force Model

► [Aquaculture Structures: Numerical Methods](#)

Sea Environment

► [Ship Operational Environment](#)

Sea Ice Management

► [Ice Management in Offshore Operations](#)

Seabed

► [Underwater Lander](#)

Seafloor

► [Underwater Lander](#)

Seakeeping Experiment

► [Aquaculture Structures: Experimental Techniques](#)

Seakeeping Tests

► [Towing Tank Test](#)

SeaStar TLP

► [Tension-Leg platform](#)

Second Generation Intact Stability Criteria

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Definition

Interim guidelines on the second generation intact stability criteria are approved by the International Maritime Organization (IMO) on 10 December 2020, and IMO Member States are invited to use the interim guidelines as complementary measures when applying the requirements of the mandatory criteria of part A of the International Code on Intact Stability, 2008 (resolution MSC.267(85)), and to bring them to the attention of all parties concerned, in particular shipbuilders, shipmasters, shipowners, ship operators, and shipping companies, and recount their experiences gained through the trial use of these interim guidelines to the organization. The second generation intact stability criteria include five stability failure modes as follows: dead ship condition, excessive acceleration, pure loss of stability, parametric rolling, and surf-riding/broaching with Level 1, Level 2 vulnerability criteria, and direct stability assessment for each stability failure model. The main purpose of these criteria is to enable the use of the latest numerical simulation techniques for evaluating the safety level of a ship from an intact stability viewpoint (IMO, Msc.1/Circ.1627, 2020).

Historical Development

The IMO intact stability criteria (resolution A.749(18)) were developed based on the

experience with ships designed and built quite some years ago – in the 1930s, and the empirical criteria do not and cannot represent the physical properties of modern vessels and a set of performance-based criteria is proposed to be developed (Germany, IMO SLF 45/6/2, 2002).

The intact stability code applies to all ships based on two different concepts: general criteria addressing the geometry of the calm water lever arm curve and the weather criterion consisting of a set of empirical formulae. The weather criterion is only based on beam wind and beam waves scenario, and the other stability failure models in quartering, following, and head seas cannot be included. Due to the restriction to the static calm water lever arm curve, dynamic problems endangering ships cannot be assessed. The scenario-based criteria including parametric rolling, pure loss of stability, and excessive acceleration in case of resonance are proposed to be developed (Germany, IMO SLF 46/6/6, 2003). A long-term approach should cover dynamic criteria (Germany, IMO SLF 46/6, 2003).

The development of dynamic stability criteria may have the following structure: criteria to avoid large rolling angles (minimum stability requirement), criteria to avoid large acceleration (maximum stability limit), criteria to guarantee sufficient roll damping in dead ship condition (minimum damping requirement), and criteria to avoid broaching (minimum course keeping limit) (Germany, IMO SLF 47/6/4, 2004).

The working group on the review of the intact stability code agreed that for three identified phenomena, i.e., pure loss and parametric excitation, behavior in dead ship condition and maneuvering-related problems (e.g., broaching-to), and necessary preliminary precautionary provisions should be in the code (as new paragraph 1.2 of part A of the draft revised Code) (Germany, SLF 48/4/7, 2005; SLF 48/WP.2, 2005).

In order to back up the long-term work tasks, Germany further gave out a proposal of probabilistic intact stability criterion (Germany, SLF 49/5/2, 2006), and Japan gave out proposals on the methodology of direct assessment for stability under dead ship condition and capsizing due to broaching (Japan, SLF 49/5/5, 49/5/6, 2006).

Japan, the Netherlands, and the USA believe that the motivation for the development of performance-based criteria for intact stability is derived from the appearance of new types of vessels. These vessels may have unconventional hull particulars or geometry or modes of operation for which there is insufficient operational experience. Performance-based criteria should be applicable to unconventional vessels while the existing intact stability code still be applicable to “conventional” vessels. The performance-based criteria also could be used as an alternative procedure for “conventional” vessels. A framework for the development of new generation criteria for intact stability is proposed by Japan, the Netherlands, and the USA, and parametric rolling, pure loss of stability, stability under dead ship condition, and broaching-to are considered (SLF 50/4/4, 2007).

The new generation intact stability criteria will be applicable to unconventional types of ships, assessed by vulnerability criteria (Germany, SLF 51/4/1, 2008). Japan gave out a proposal of new generation intact stability criteria as an example (Japan, SLF 51/4/3, 2008). The working group agreed to the framework for the new generation intact stability criteria and the draft terminology for the new generation intact stability criteria (SLF 51/WP.2, 2008).

In considering the best way forward on the necessary work for the new generation intact stability criteria, the working group, based on the report of the correspondence group (SLF 52/3/1, 2009 and SLF 52/INF.2, 2009), taking into account the relevant documents submitted to this session, developed a summary of methodology considered for the stability failure modes. The framework of the new generation stability criteria is confirmed including Level 1, Level 2 vulnerability criteria, direct stability assessment, and operational guidance for stability failure models, and the stability failure modes include pure loss of stability, parametric rolling, surf-riding/broaching, and dead ship condition (SLF 52/WP.1, 2010).

The name of “the new generation intact stability criteria” is changed as “the second generation intact stability criteria” in the conference SLF 53, and the structure of the second generation

intact stability criteria is updated with five stability failure models such as pure loss of stability, parametric roll, surf-riding/broaching, dead ship condition, and excessive acceleration, and each failure model has multiple levels including Level 1, Level 2 vulnerability criteria, direct stability assessment, and operational guidance (SLF 53/WP.4, 2011).

Following an in-depth discussion, the working group prepared an updated version of the draft vulnerability criteria of Level 1 and 2 for pure loss of stability, parametric roll, dead ship, and surf-riding/broaching (SLF 54/WP.3, 2012).

The working group prepared a revised plan, identifying the priorities, time frames, and objectives for the work to be accomplished. The working group agreed on the draft vulnerability criteria of Levels 1 and 2 for the failure mode parametric roll, and agreed to the selection of a wave environment for the first and second level vulnerability criteria for parametric rolling. The working group discussed the direct assessment methodologies considering their importance for ships which fail vulnerability levels 1 and 2 and to develop operational guidance when necessary (SLF 55/WP.3, 2013).

Subcommittee of ship design and construction (SDC) is established including the working of subcommittee on stability and load lines and on fishing vessels safety. The working group reviewed the second generation intact stability criteria on the base of the correspondence group and prepared a revised plan, identifying the priorities, time frames, and objectives for the work to be accomplished (SDC 1/WP.5, 2014).

The working group finalized the draft amendment to the 2008 Intact Stability Code regarding vulnerability criteria and the standards of levels 1 and 2, and further developed the direct stability assessment procedures (level 3) for the five stability failure models. The working group was the view that each criterion should have a single option. On the other hand, the working group was also of the view that the calculation of some parameters may include different options (SDC 2/WP.4, 2015).

For the finalization of the second generation intact stability criteria, the working group further

considered the draft criteria in levels 1 and 2 for the five stability failure modes, further developed the draft explanatory notes for all five failure modes, and further developed the draft guidelines of direct stability assessment procedures and operational limitation/guidance (SDC 3/WP.5, 2016).

The working group further considered the draft criteria in levels 1 and 2 for the five stability failure modes, further developed the draft guideline for the specification of direct stability assessment with a view to preparing draft interim guidelines, further developed the draft explanatory notes for all five failure modes with a view completing the draft interim explanatory notes, and further developed the draft guidelines for the preparation and approval of operation limitation and operational guidance with a view to preparing draft interim guidelines (SDC 4/WP.4, 2017).

A provisional work plan was drafted with a view to ensuring the finalization of the second generation of intact stability criteria (SDC 5/J.6, 2018).

With a view to presenting outcomes ready for finalization, continued progress made by the experts' group on intact stability including the draft interim guidelines on the specification of direct stability assessment procedures, the draft interim guidelines for the preparation of operational limitation and operational guidance, and the draft interim guidelines on vulnerability criteria for the second generation of intact stability criteria (SDC 6/WP.6, 2019).

The draft group on intact stability discussed the best possible method of work to finalize the draft interim guidelines on the second generation intact stability criteria and agree to elaborate on the matters pending in square brackets as a matter of priority, followed by a complete editorial review of the draft guidelines. The draft group agreed to remove all references to the draft explanatory note (SDC 7/WP.6, 2020).

Interim guidelines on the second generation intact stability criteria is approved by the International Maritime Organization (IMO) on 10 December 2020 (IMO, Msc.1/Circ.1627, 2020).

The explanatory note is under discussion by the correspondence group of the second

generation intact stability criteria and will be submitted to SDC 8 (Jan., 2022).

Application Logic

Figure 1 shows the simplified scheme of the application structure of the second generation intact stability criteria. As the simplest options, the vulnerability criteria are presented in two levels: Level 1 and Level 2. The assessment of the five stability failure modes should begin with the use of these levels. Level 1 is an initial check and then, if the ship in a particular loading condition is assessed as not vulnerable for the tested failure mode, the assessment for that failure mode may conclude, otherwise the design would progress to Level 2. If the ship in a particular loading condition is assessed as not vulnerable for the tested failure mode in Level 2, then the assessment would conclude, otherwise the design would progress to the application of direct stability assessment, application of operational limitations, revising the design of the ship, or discarding the loading condition. If the ship in a particular loading condition is not found acceptable with respect to direct stability assessment procedures, then the logic is that the design would then

progress to the application of operational measures or operational guidance, revising the design or discarding the loading condition (IMO, Msc.1/Circ.1627, 2020).

Vulnerability Criteria of Five Stability Failure Modes

Parametric Rolling

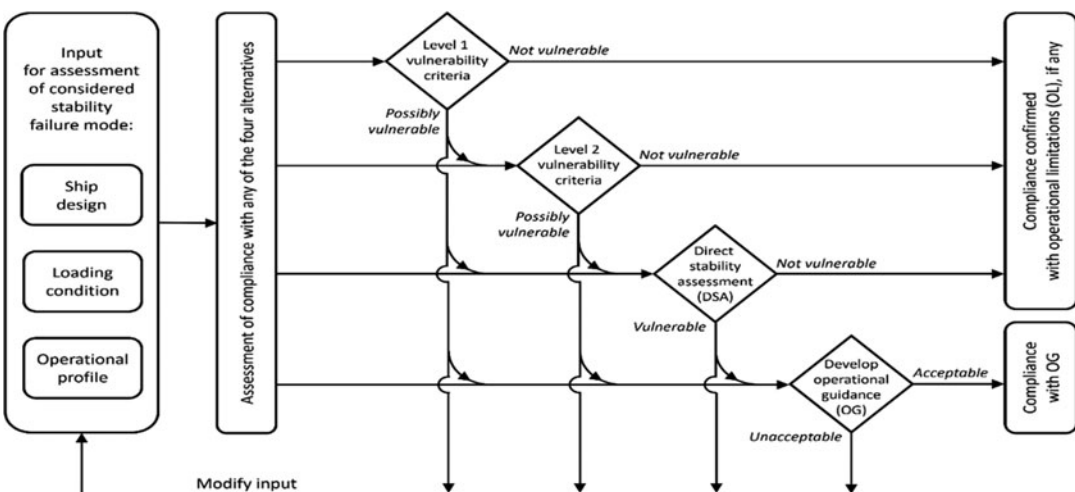
Parametric rolling is a dynamic stability phenomenon in which an amplification of roll motion is caused by periodic variation of transverse stability in waves. The vulnerability criteria of parametric rolling are shown as follows.

Level 1 Vulnerability Criteria of Parametric Rolling

For each loading condition a ship is considered not to be vulnerable to the parametric rolling failure mode if

$$\frac{\delta GM_1}{GM} \leq R_{PR} \quad \text{and} \quad \frac{V_D - V}{A_W(D - d)} \geq 1.0 \quad (1)$$

In which $R_{PR} = 1.87$ if the ship has a sharp bilge and otherwise:



Second Generation Intact Stability Criteria, Fig. 1 Simplified scheme of the application structure of the second generation intact stability criteria. (IMO, Msc.1/Circ.1627, 2020)

$$\begin{cases} R_{PR} = 0.17 + 0.425 \left(\frac{100A_k}{LB} \right), & \text{if } C_{m,\text{full}} > 0.96; \\ R_{PR} = 0.17 + (10.625 \times C_{m,\text{full}} - 9.775) \left(\frac{100A_k}{LB} \right), & \text{if } 0.94 \leq C_{m,\text{full}} \leq 0.96; \\ R_{PR} = 0.17 + 0.2125 \left(\frac{100A_k}{LB} \right), & \text{if } C_{m,\text{full}} < 0.94; \end{cases} \quad (2)$$

$\left(\frac{100A_k}{LB} \right)$ should not exceed 4; δGM_1 is amplitude of the variation of the metacentric height (m) calculated by Eq. (3), and $S_W = 0.0167$.

$$\delta GM_1 = \frac{I_{TH} - I_{TL}}{2V} \quad (3)$$

where I_{TH} , I_{TL} are transverse moment of inertia of the waterplane at the draft d_H (m) and transverse moment of inertia of the waterplane at the draft d_L (m).

$$\begin{aligned} d_H &= d + \delta d_H \\ d_L &= d - \delta d_L \\ \delta d_H &= \text{Min} \left(D - d, \frac{L \cdot S_W}{2} \right) \\ \delta d_L &= \text{Min} \left(d - 0.25d_{\text{full}}, \frac{L \cdot S_W}{2} \right) \end{aligned} \quad (4)$$

Level 2 Vulnerability Criteria of Parametric Rolling
For each condition of loading a ship is considered not to be vulnerable to parametric rolling if

$$\begin{aligned} C1 &\leq R_{PR1} = 0.06 \text{ or} \\ C2 &\leq R_{PR1} = 0.025 \end{aligned} \quad (5)$$

The value for $C1$ is calculated as a weighted average from a set of waves specified in (IMO, Msc.1/Circ.1627,2020).

$$C1 = \sum_{i=1}^N W_i C_i \quad (6)$$

In which W_i is weighting factor according to the wave data specified in (IMO, Msc.1/Circ.1627,2020).

If the Eq. (7) is satisfied, $C_i = 0$, otherwise $C_i = 1$.

$$\begin{aligned} &\left(GM(H_i, \lambda_i) > 0 \text{ and } \frac{\delta GM(H_i, \lambda_i)}{GM(H_i, \lambda_i)} < R_{PR} \right) \\ \text{or } &\left(\text{when } GM(H_i, \lambda_i) > 0, V_{PRi} = \left| \frac{2\pi_i}{T_r} \cdot \sqrt{\frac{GM(H_i, \lambda_i)}{GM}} - \sqrt{g \frac{\lambda_i}{2\pi}} \right| > V_s \right) \end{aligned} \quad (7)$$

The value for $C2$ is calculated as a weighted average from a set of waves conditions specified in (IMO, Msc.1/Circ.1627,2020)

$$C2(Fn_i, \beta) = \sum_{i=1}^N W_i C_{S,i} \quad (9)$$

$C2 =$

$$\left[\sum_{i=1}^{12} C2(Fn_i, \beta_h) + \frac{1}{2} (C2(0, \beta_h) + C2(0, \beta_f)) + \sum_{i=1}^{12} C2(Fn_i, \beta_f) \right] / 25 \quad (8)$$

$C2(Fn_i, \beta)$ can be calculated by Eq. (9).

In which W_i is weighting factor for respective wave cases specified in (IMO, Msc.1/Circ.1627,2020). Also, N is total number of wave cases for which the maximum roll angle is evaluated for a combination of speed and ship heading; C_i is 1 if the maximum roll angle in head and following waves according to one degree of rolling motion exceed 25 and 0 otherwise.

Pure Loss of Stability

The provisions given hereunder apply to all ships, except for ships with an extended low weather deck, for which the Froude number, Fn , corresponding to the service speed exceeds 0.24.

Level 1 Vulnerability Criteria of Parametric Rolling

For each loading condition a ship is considered not to be vulnerable to the pure loss of stability failure mode if

$$GM_{\min} \geq R_{PLA} \quad \text{and} \quad \frac{V_D - V}{A_W(D - d)} \geq 1.0 \quad (10)$$

where $R_{PLA} = 0.05$; GM_{mi} is minimum value of the metacentric height (m) calculated as provided in Eq. (11); $S_W = 0.0334$.

$$GM_{\min} = KB + \frac{I_{TL}}{\nabla} - KG \quad (11)$$

I_{TL} are transverse moment of inertia of the waterplane at the draft d_L (m).

$$d_L = d - \delta d_L$$

$$\delta d_L = \text{Min} \left(d - 0.25d_{\text{full}}, \quad \frac{L \cdot S_W}{2} \right) \quad (12)$$

KB : height of ship buoyance; KG : height of ship gravity; ∇ : displacement (m^3).

Level 2 Vulnerability Criteria of Parametric Rolling

For each condition of loading a ship is considered not to be vulnerable to pure loss of stability if

$$\max (CR_1, CR_2) \leq R_{PL0} = 0.06 \quad (13)$$

Each of the two criteria, CR_1 and CR_2 in 2.4.3.1, represents a weighted average of certain stability parameters for a ship considered to be statically positioned in waves of defined height, H_i , and length, λ_i , obtained according to (IMO, Msc.1/Circ.1627,2020). CR_1 and CR_2 are calculated as follows:

$$CR_1 = \sum_{i=1}^N W_i C1_i; \quad (14)$$

$$CR_2 = \sum_{i=1}^N W_i C2_i;$$

In which W_i is weighting factor for respective wave cases specified in (IMO, Msc.1/Circ.1627,2020). Also, $C1_i$ and $C2_i$ are calculated as follows:

$$C1_i = \begin{cases} 1 & \varphi_V < K_{PL1} = 30^\circ \\ 0 & \text{otherwise} \end{cases}$$

$$C2_i = \begin{cases} 1 & \begin{matrix} \varphi_{sw} > K_{PL2} \\ K_{PL2} = 15^\circ \text{ for passenger ships} \\ K_{PL2} = 25^\circ \text{ for other ships} \end{matrix} \\ 0 & \text{otherwise} \end{cases} \quad (15)$$

In which φ_v and φ_{sw} are angle of vanishing stability and angle of heel under action of heeling lever $l_{PL2} = 8(H_i/\lambda)dF_N^2$ specified in (IMO, Msc.1/Circ.1627,2020).

Surf-Riding/Broaching

Broaching is a violent uncontrollable turn that occurs despite maximum steering efforts to maintain course. As with any other sharp turn event, broaching is accompanied with a large heel angle, which has the potential effect of a partial or total stability failure. Broaching is usually preceded by surf-riding which occurs when a wave, approaching from the stern, captures a ship and accelerates the ship to the speed of the wave. Surf-riding is a single wave event in which the wave profile does not vary relative to the ship. Because most ships are directionally unstable in the surf-riding condition, this maneuvering yaw instability leads to an uncontrollable turn – termed “broaching” (SDC 3/WP.5/Annex 5, 2016). Broaching is considered as one of the most dangerous phenomena in following and stern-quartering waves for high-speed ships, such as destroyers and fishing vessels.

Level 1 Vulnerability Criteria of Surf-Riding/Broaching

For each loading condition a ship is considered not to be vulnerable to the surf-riding/broaching failure mode if

$$L \geq 200\text{m} \quad \text{or} \quad Fn \leq 0.3 \quad (16)$$

In which Fn is Froude number corresponding to ship service speed; L is ship length.

Level 2 Vulnerability Criteria of Surf-Riding/Broaching

For each loading condition a ship is considered not to be vulnerable to the surf-riding/broaching failure mode if

$$C = \sum_{H_s} \sum_{T_z} \left(W2(H_s, T_z) \sum_{i=0}^{N_\lambda} \sum_{j=0}^{N_a} w_{ij} C2_{ij} \right) < R_{SR} = 0.005 \quad (17)$$

Where $W2(H_s, T_z)$ is weighting factor of short-term sea state as a function of significant wave height H_s and zero crossing wave period T_z . Also w_{ij} is statistical weight of a wave with steepness $(H/\lambda)_i$ and wave length to ship length ratio $(\lambda/L)_i$ calculated with the joint distribution of local wave steepness and lengths. Finally, $C2_{ij}$ is a coefficient depends on ship propulsion and resistance characteristics as follows:

In which $N_\lambda = 80$, $N_a = 100$.

$$w_{ij} = 4 \frac{\sqrt{g} L^{5/2} T_{01}}{\pi v (H_s)^3} s_j^2 r_i^{3/2} \left(\frac{\sqrt{1+v^2}}{1+\sqrt{1+v^2}} \right) \Delta r \Delta s \cdot \exp \left[-2 \left(\frac{L \cdot r_i \cdot s_j}{H_s} \right)^2 \left\{ 1 + \frac{1}{v^2} \left(1 - \sqrt{\frac{g T_{01}^2}{2 \pi r_i L}} \right)^2 \right\} \right] \quad (18)$$

In which, $v = 0.425$; $T_{01} = 1.086 T_z$; $s_j = (H/\lambda)_j$, varying from 0.03 to 0.15 with increment $\Delta s = 0.0012$; $r_i = (\lambda/L)_i$, varying from 1.0 to 3.0 with increment $\Delta r = 0.025$.

The value of $C2_{ij}$ is calculated for each wave, as follows:

$$C2_{ij} = \begin{cases} 1 & \text{if } Fn > Fn_{cr}(r_i, s_j) \\ 0 & \text{if } Fn \leq Fn_{cr}(r_i, s_j) \end{cases} \quad (19)$$

$Fn_{cr} = u_{cr}/\sqrt{Lg}$ is critical Froude number corresponding to the threshold of surf-riding (surf-riding occurring under any initial condition) which should be calculated for the regular wave with steepness s_j and wavelength to ship length ratio r_i .

The critical nominal ship speed, u_{cr} , is determined by solving the following equation with the critical propulsor revolutions, n_{cr} :

$$T_e(u_{cr}; n_{cr}) - R(u_{cr}) = 0 \quad (20)$$

The critical propulsor revolutions, n_{cr} , is determined by solving the following equation in (IMO, Msc.1/Circ.1627,2020)

$$2\pi \frac{T_e(c_i; n_{cr}) - R(c_i)}{f_{ij}} + 8a_0 n_{cr} + 8a_1 - 4\pi a_2 + \frac{64}{3} a_3 - 12\pi a_4 + \frac{1024}{15} a_5 = 0 \quad (21)$$

Dead Ship Condition

Level 1 Vulnerability Criteria of Dead Ship Condition

For each loading condition a ship is considered not to be vulnerable to the dead ship condition failure mode if

$$b \geq a \quad (22)$$

In which, a and b are calculated according to 2008 IS Code Part A-2.3, and MSC.1/Circ.1200 Table 4.5.1.

Level 2 Vulnerability Criteria of Dead Ship Condition

For each loading condition a ship is considered not to be vulnerable to the dead ship condition failure mode if

$$C = \sum_{i=1}^N W_i(H_S, T_Z) \times C_{S,i}(H_S, T_Z) \leq R_{DS0} = 0.06 \quad (23)$$

In which, W_i is weighting factor of short-term sea state as a function of significant wave height H_S and zero crossing wave period T_Z .

$C_{S,i}$ is short-term dead ship stability failure index for the short-term environmental condition under consideration, calculated as specified in (IMO, Msc.1/Circ.1627,2020). N is total number of short-term environmental conditions.

$$C_{S,i} = 1, \text{ if either :} \\ \begin{aligned} &.1, \text{ the mean wind heeling lever } -l_{\text{wind,tot}} \text{ exceeds the righting lever, } GZ, \text{ at each heel to leeward, or} \\ &.2, \text{ the stable angle under the action of steady wind, } \varphi_S, \text{ is greater than the angle of failure to leeward, } \varphi_{\text{fail,+}}, \text{ and} \\ &= 1 - \exp(-r_{EA}T_{\text{exp}}), \text{ otherwise} \end{aligned} \quad (24)$$

Excessive Accelerations

When a ship is rolling, the objects in higher locations travel longer distances. A period of roll motions is the same for all the location onboard the ship. To cover longer distance during the same time, the linear velocity must be larger. As the velocity changes its direction every half a period, larger linear velocity leads to larger linear accelerations. Large linear acceleration means larger inertial force (SDC 3/WP.5/Annex 7, 2016).

Level 1 Vulnerability Criteria of Excessive Acceleration

For each loading condition a ship is considered not to be vulnerable to the excessive acceleration failure mode if

$$\varphi k_L (g + 4\pi^2 h_r / T_r^2) \leq R_{EA1} = 4.64 \text{ (m/s}^2\text{)} \quad (25)$$

In which φ is characteristic roll amplitude (rad) calculated by Eq. (26); k_L is factor taking into account simultaneous action of roll, yaw, and pitch motions; g is gravity acceleration; h_r is height above the assumed roll axis of the location where passengers or crew may be present (m), for

which definition, the roll axis may be assumed to be located at the midpoint between the waterline and the vertical center of gravity; and T_r is natural roll period.

$$\varphi = 4.43rs/\delta_\varphi^{0.5} \quad (26)$$

In which, r is effective wave slope coefficient; s is wave steepness as a function of the natural roll period T_r as determined in (IMO, Msc.1/Circ.1627,2020); and δ_φ is nondimensional logarithmic decrement of roll decay.

Level 2 Vulnerability Criteria of Excessive Acceleration

For each loading condition a ship is considered not to be vulnerable to the excessive acceleration failure mode if

$$C = \sum_{i=1}^N W_i C_{S,i} \leq R_{EA2} = 0.00039 \quad (27)$$

In which, long-term probability index measures the vulnerability of the ship to a stability failure due to excessive acceleration for the loading condition and location under consideration based on the probability of occurrence of short-

term environmental conditions, as specified in (IMO, Msc.1/Circ.1627,2020); W_i is weighting factor for the short-term environmental condition; and $C_{S,I}$ is short-term excessive acceleration failure index for the short-term environmental condition under consideration, calculated by Eq. (28).

$$C_{S,i} = \exp \left[-R_2^2 / (2\sigma_{Lai}^2) \right] \quad (28)$$

In which, $R_2 = 9.81 \text{ (m/s}^2\text{)}$; σ_{Lai} is standard deviation of the lateral acceleration at zero speed and in a beam seaway.

Cross-References

- [Auxiliary Icebreaking Methods](#)
- [Human Occupied Vehicle \(HOV\)](#)
- [Reliability Based Design \(RBD\)](#)
- [Transport Ship](#)
- [Underwater Mining System](#)

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Self-Organizing Network

- [Underwater Acoustic Sensor Network](#)

Self-Propulsion tests

- [Towing Tank Test](#)

Self-Sustaining Profiling Automation Circulation Detector (SPACD)

- [Profiling Float](#)

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Semisubmersible Drilling and Production Rig (SSDP)

- [Polar Offshore Engineering](#)

Semi-submersible Platform

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Synonyms

CUSS I (Continental, Union, Superior and Shell Oil Companies' Project Mohole); DP (dynamic positioning); LFC (low-frequency cycle); VIM (vortex-induced motion)

Definition

A **semi-submersible platform** is a specialized offshore unit used in a number of specific offshore roles such as drilling rigs, safety vessels, oil production platforms, and heavy lift cranes. They are designed with good stability and seakeeping characteristics. Other terms include semi-submersible, semi-sub, or semi for simplification (https://en.wikipedia.org/wiki/Semi-submersible_platform).

Introduction

Offshore drilling in water depth greater than 500 meters normally requires that operations be carried out from a floating vessel, as the cost on fixed structures are normally unaffordable. Initially in the early 1950s, monohull ships such as CUSS I (Keuren and David 2004) were used, but these were found to have significant heave, pitch, and yaw motions in large waves, and the industry needed more stable drilling platforms. Thus, semi-submersible platform concept came into engineering application soon (Fig. 1).

A semi-submersible platform, also known as the column stable drilling platform, which operates mainly in the deepwater area, is a floating production system that enables the drilling and exploitation of offshore oil and gas resource. Most of the floating body is below the water



Semi-submersible Platform, Fig. 1 North Dragon semi-submersible platform

surface, and the general positioning system is mooring lines, anchoring equipment, and dynamic positioning system (DP). Nowadays, a semi-submersible platform generally consists of a platform body, columns, a pontoon, and a number of support rods – a pontoon submerged under water, providing primary buoyancy, and columns connecting the deck body to the pontoon.

A semi-submersible obtains most of its buoyancy from ballasted, watertight pontoons located below the water surface and are away from wave action. Structural columns connect the pontoons and operating deck. The operating deck can be located high above the sea level owing to the good stability and therefore is kept well away from the waves.

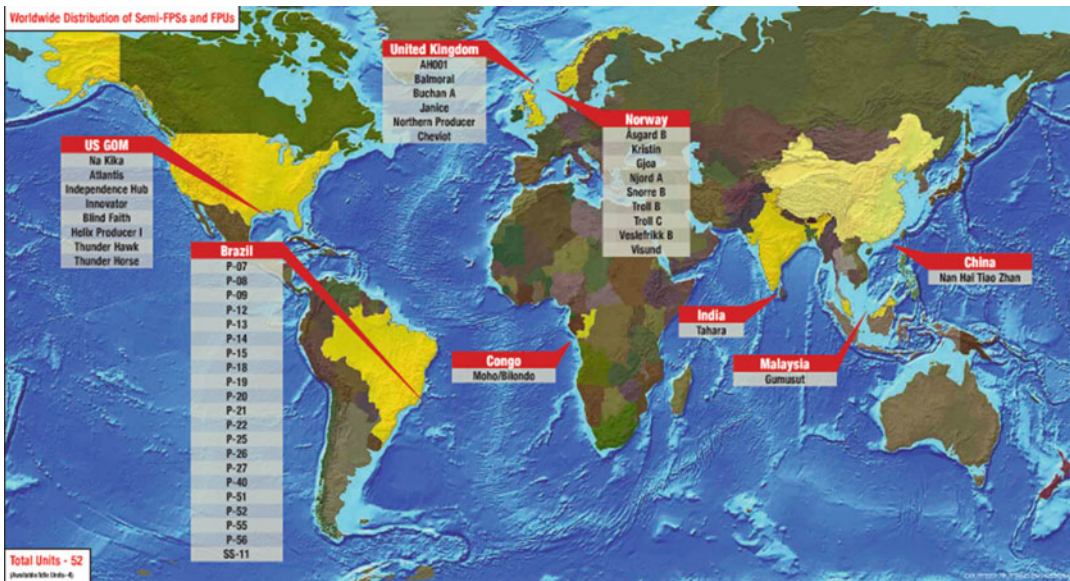
With its hull structure submerged at a deep draft, the semi-submersible is less affected by wave loadings than a normal ship. However, with a small water plane area, however, the semi-submersible is sensitive to load changes and therefore must be carefully trimmed to maintain stability.

Semi-submersible platform can be used as both drilling and production platform. It has a series of advantages such as high safety factor, large deck space (up to 110 m/110 m in length and width), and strong expansion of the topside in the later stage. The deck area and load can be flexibly adjusted according to the actual situation. The platform is also flexible to move, the stability is relatively good, and the reliability is high. In addition, semi-submersible platform can be used for other offshore operation utilizations, including pipelayers, supply vessels, offshore lifting vessels, etc., especially widely used in deepwater oil and gas exploration.

History

The semi-submersible design was first developed for offshore drilling activities. Bruce Collipp (From “Wikipedia [n.d.](#)) of Shell is regarded as the inventor. But Edward Robert Armstrong (From “Wikipedia [n.d.](#)) may have paved the way with his idea of “seadrome” landing strips for airplanes in the late 1920s, since his idea involved the same use of columns on ballast tanks below the surface and anchored to the ocean floor by steel cables. When oil drilling moved into offshore waters, fixed platform rigs and submersible rigs were built, but were limited to shallow waters.

The first semi-submersible arrived by accident in 1961. Blue Water Drilling Company owned and operated the four column submersible drilling rig Blue Water Rig No.1 in the Gulf of Mexico for Shell Oil Company. As the pontoons were not sufficiently buoyant to support the weight of the rig and its consumables, it was towed between locations at a draft midway between the top of the pontoons and the underside of the deck. It was observed that the motions at this draft were very small, and Blue Water Drilling and Shell jointly decided that the rig could be operated in the floating mode. The first purpose-built drilling semi-submersible Ocean Driller was launched in 1963. Since the emergence of a semi-submersible offshore platform in the early 1960s, it has been widely used in the offshore oil exploration and development. In



Semi-submersible Platform, Fig. 2 Distribution of semi-submersible production platform (2008)

1975, the world's first semi-submersible production platform "DeepSea Pioneer" was built. Most of the semi-submersible production platforms were converted from drilling rigs in the initial stages of the development of floating production facilities. Due to the small number of drilling rigs available for modification since the 1990s, it became less attractive than FPSO. However, with the worldwide exploitation of deepwater and marginal oil fields, semi-submersible production platforms were once again favored by oil exploration companies after the 1990s. Its main operating blocks are mainly in the Brazilian waters, the Gulf of Mexico, the South China Sea, and the North Sea. Fig. 2 shows the semi-submersible production platform operational distribution in 2008 (2008 Semi Floating Production Vessel Poster n.d.).

Structural Features

The semi-submersible platform is composed of an upper deck, columns, a lower pontoon or rigs, and anchoring system. The advanced fifth-generation or sixth-generation semi-submersible drilling

platforms are all equipped with DP system. If DP system is not used, the platform is normally connected to the seafloor through the mooring system. Since its appearance, the structure of the semi-submersible platform has undergone many evolutions, and its appearance tends to be simple. Huge pontoon provides the buoyancy required for towing work.

The columns connect the pontoon and the upper deck. During operation, the pontoon and large proportion of columns are sunk into the water, which greatly reduces the water surface area and thus reduce heave motion amplitude of the platform. The larger water plane moment of inertia provides the stability required for the platform operation. In order to consider damage stability and cost-effect, the number of columns is generally 4 to 8.

The upper deck is the main place for equipment storage and personnel residency, providing work site and arranging production and living facilities. The money-maker of the platform, the rig, is installed on the deck. The accommodation facilities are always designed near helicopter platform, in case of emergency evacuation.

Semi-submersible Platform, Fig. 3 Offshore support vessel Toisa Perseus with, in the background, the fifth-generation deepwater drillship Discoverer Enterprise, over the Thunder Horse Oil Field. Both are equipped with DP systems



Dynamic positioning system (DP) is mostly installed on newly built semi-submersible drilling platform, because they are always expected to operate in deepwater area and DP operation can save great time and cost than those of traditional mooring positioning operation. DP system can be composed of 6–8 propellers installed beneath the pontoon, and their directions can be adjusted with 360 degree to counteract the wind load, current load, and second order wave load (Fig. 3).

Construction and Installation Process

The construction and installation process of the semi-submersible platform is very complex, time-consuming, and costly. At present, only a few big companies in the world can afford the construction and operation costs. Some of them can only undertake part of the construction work such as the construction of the hull. Nowadays, some international companies that are famous for their construction of semi-submersible drilling platforms are mainly located in East and Southeast Asia, for instance, Keppel Corporation and Sembcorp Marine Company in Singapore, Samsung Heavy Industries and Daewoo Shipbuilding and Marine Engineering Company in South Korea, and Raffles

Shipyard, Waigaoqiao Shipyard, and Dalian Shipyard in China (Fig. 4).

Vortex-Induced Motion Effect

For the conventional semi-submersible platform, the natural periods of roll, heave, and pitch motions are between 40 and 60s; when using the mooring positioning systems, the natural periods of surge, sway, and yaw motions are between one to several minutes, which are far from the general wave energy range that is normally between 5 and 30s. Therefore, the first-order wave motions including heave, roll, and pitch will be quite small and benefit the drilling operation very well. However, under the action of the second-order low-frequency wave force effect, semi-submersible platform can have obvious slow-drift motion on surge and sway, which has to be borne by mooring system or DP system. In addition wave run-up phenomenon is another important phenomenon often being concerned, especially under severe sea state (Danmeier et al. 2008). Recently, with the application of deep-draft semi-submersible concept, vortex-induced motion (VIM) problem has also arisen to cause much attention.

Semi-submersible Platform,

Fig. 4 Construction and installation process of semi-submersible platform in different shipyards



Vortex-induced motion (VIM) is one of the hydrodynamic problems for deep-draft semi-submersible platform with large column height. Traditional semi-submersible platforms have relatively low drafts, and the height of the underwater part of the column is small, so its vortex-induced motion is not obvious. However, with the development of petroleum exploration toward the deep sea in recent years, the loading requirements for semi-submersible platforms have been continuously improved, and the draft design draft has also gradually increased, that is, it has developed into a deep-draft semi-submersible platform.

The semi-submersible platform is a typical multi-column floating body. The tail of the

column will generate periodic vortex shedding under certain flow conditions. The vortex shedding will generate pulsating pressure. Under certain flow conditions, the vortex will follow the surface of the cylinder and generate alternating falling motion, which leads to the horizontal resonant vibrations that cause vortex-induced motion (VIM). When the natural frequency of the platform system is close to the frequency of vortex shedding, this resonant motion is even more serious, resulting in a large low-frequency cycle (LFC) fatigue damage to the mooring and riser systems. Compared with the high-frequency damage caused by waves, the dynamic tension amplitude caused by the

vortex-induced kinematics is much greater, resulting in more severe fatigue problems. In addition, a larger vortex-induced response will also have a greater impact on the platform's mooring and positioning capabilities. Therefore, in the design stage of mooring and riser systems, accurately predicting the vortex-induced motion characteristics of the platform under the action of the ocean current provides an important reference for checking whether the fatigue lives of mooring and riser systems satisfy the requirements for the whole service life.

The characters of VIM of semi-submersible platform are different with those of other offshore structures, such as SPAR platform. Semi-submersible platforms usually consist of a pontoon, cross braces, and multiple columns, so under the vortex-shedding excitation effect, the vortex-induced motion characteristics of the semi-submersible platform are quite different from that of a standpipe or a slender body. For a deep-draft semi-submersible platform, the maximum vortex amplitude A/D can reach 0.5–0.6 (where A is the amplitude of transverse vortex-induced motion, and D is the dimension of platform for dimensionless), which will affect the fatigue life of anchor chain and the hanging catenary riser. The vortex-induced motion is related to the maximum displacement of the platform, which will increase the drag force acting on the platform, especially when the vortex-induced motion is severe, and even affect normal drilling operations. Therefore, the VIM has become a challenge that restricts the development of the deep-draft semi-submersible platform.

In addition, VIM of semi-submersible platform is special due to the multi-number of columns. Compared with single column platforms, such as the spar platform, the vortex shedding of each column of semi-submersible is superimposed on each other, and the effect of interference makes the VIM characteristics of the deep-draft semi-submersible platform more complicated. Because the importance of predicting VIM effect during the design stage of deep-draft semi-submerged platform, many studies have been conducted based on theoretical analysis and basin experiment methods.

Wave Slamming

With the development of global ocean exploration and development, the semi-submersible platform has become one of the main tools for the development of offshore oil and gas by its advantages. The basic structural design and analysis have become the important technical content in the basic design. DNVGL (DNV RP-C103 Column-stabilized units 2005) classification requirements for the basic structural design of semi-submersible offshore platforms include completion of overall strength analysis, local strength analysis, web frame strength analysis, air gap analysis, wave slamming (on lower deck by wave loading) strength analysis, etc. The strength analysis of the lower deck structure when attacked by the wave slamming load is an important part in the basic structural design report. In severe sea state, the wave will impact the lower deck structure, which may lead to serious structural damage. Therefore, it is necessary to consider the effect of wave slamming load on the lower deck during the design process.

Classification societies provide some analysis procedure for the wave slamming effect check.

Rule-Based Check Method

Rule-based check method is the most easily used method by designers, because they only need to conform to the rules of classification societies and can maintain a high working efficiency. For instance, the DNVGL states that the wave load calculation formula calculates the average slamming pressure (DNV RP-C205-environmental conditions and environmental loads 2007):

$$P_s = 0.5\rho \cdot C_{pa} \cdot v^2$$

In the formula,

P_s – the average attack pressure.

ρ – the liquid mass density, the calculated value of 1025 kg/m³.

C_{pa} – the load coefficient; $C_{pa} = 2\pi$ when the flat bottom surface is impacted; $C_{pa} = 5.15$ when the smooth cylinder is subjected to impact.

v – the relative normal velocity between the water surface and the lower deck surface.

According to this formula, for all areas in the lower deck of the semi-submersible probably subjected to wave load impact calculated in the air gap analysis, the relative speed and slamming force at the impact of the wave slamming load on these areas in all wave directions are calculated.

With the wave slamming load calculation results, the bottom of deck structure of semi-submersible can be checked by classification societies' rules. There are many rules for structural strength check. So, this is a convenient working procedure. Nevertheless, the accuracy of this method relies much on the classification societies' rules, so it is classification societies' duty to guarantee the feasibility and accuracy of those rules.

Numerical Simulation Method

Numerical simulation is one of the most important methods to analyze and assess the structural safety of the lower deck structure of semi-submersible platform. There are many commercial codes available being used to do the numerical simulations of wave slamming, including ABAQUS, FLUENT, and LS_DYNA. One of the advantages of numerical simulation method is the relative accuracy that this method can provide, and the slamming process can be shown in much detail in time series. The other advantages of the numerical simulation method is that the simulation results can find some critical weakness points of deck structure, which will help to strengthen the deck structure accurately. Nevertheless, there are also some weak points of this method, one of the most critical one is the high cost of human labor and computation time. For example, it may be a challenge task for a designer to learn and use sophisticated CFD codes such as FLUENT. Furthermore, the simulation results could be versatile for different users (Qinjing et al. 2011).

It is believable that with the improvement of IT technology, numerical simulation method will become more and more popular and reliable. At that time, the calculation of many nonlinear phenomena will employ numerical simulation methods during the design phase.

Air Gap

Air gap has much relationship with wave slamming effect in previous section. For semi-submersible platforms, the vertical support structure below these platforms, namely, columns, are generally more slender, which is to withstand the wave load, thereby enhancing its ability to work in harsh sea conditions. If the initial air gap of the platform is small, the waves can easily slam the structure of the lower deck and platform directly, which can result in large slamming wave load and cause serious structural damage and endanger the people onboard and even, sometimes, overturning of the platform. So for the platform, air gap must not be too small, so as to avoid damage to the platform.

During the design stage of offshore platform, especially semi-submersible platform, air gap is always one of the most critical issues. It is a highly nonlinear problem and plays a very important role in the two major factors – the response of wave height and the platform vertical motion amplitude.

When considering the wave height calculation, it normally applies both linear and nonlinear wave theories. If only linear wave theory is used, it will be inevitably underestimating the wave climbing value, which is more serious with the increase of wave steepness. Comparatively, the forecast values by applying both linear and nonlinear wave theories are in better agreement with the experimental values. The other calculation issue is the motion amplitude of the platform. If only fixed platform is considered, the air gap problem can be much simplified, but if the motions of floating offshore platform are considered, the forecast of air gap value will become more complicated, because the phases of both motions and that of wave height need to be both considered.

In general, the nonlinear issue of air gap forecast is more accurate than that of the linear forecast value, which is worth exploring and practicing.

Air Gap Forecast of the Semi-Submersible Platform

For semi-submersible platform, it has two types of deck superstructures – open deck structure

Semi-submersible Platform, Fig. 5 Wave slamming on semi-submersible platform



and closed deck structure – which refer to whether the deck superstructure of the platform is open design or closed design. In the case of an open design, small waves on the deck or lashing can damage the safety of lives and property facilities on the platform and so it compulsorily required that the air gap of the platform must be kept positive. As for the closed superstructure design, due to the better protection brought by the enclosed superstructure, slight waves slamming will not have a significant impact on the platform deck, so it can be applied to the harsh marine environment.

Taking into account the cost of the platform construction, the large distance between the bottom of deck structure and the still water surface, namely, the great design value of the initial air gap, often means increasing construction cost. Therefore, for most latest version semi-submersible platforms, in order to avoid the negative air gap completely, it is more economical to perform local strengthening check than to increase the initial air gap (Sweetman et al.) (Sweetman et al. 2002a). Thus, semi-submersible can withstand some wave slamming without the need to increase initial air gap (Fig. 5).

For floating platforms, such as semi-submersible platforms, the interference between platforms motion and wave is serious. Due to the effects of wave diffraction and scattering, the crests of some parts of the platform are often overlapped with each other so that the response of the air gap can be amplified or reduced. Therefore, the forecast of air gap is a high nonlinear problem, and it is quite complicated. At present, CFD and numerical calculation are the most reliable methods for the air gap forecast.

Development of Air Gap Forecast Methods

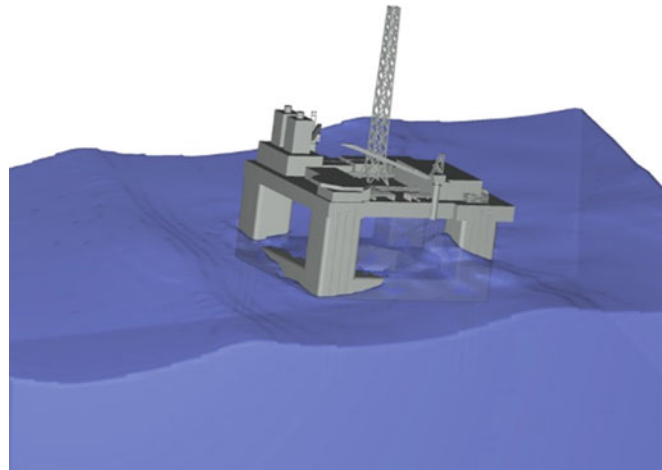
At present, numerical prediction of the air gap mainly includes CFD forecast method and numerical calculation forecast method based on potential flow theory. In addition, basin experiment is also used for air gap prediction.

For CFD method, it can provide accurate forecasting results with good visual effect, if the required technologies can be used in a correct way. There are many successful examples in air gap forecasting by some famous CFD tools, for instance, OpenForm, Fluent et al. But CFD method also has its drawbacks, such as high time-consuming and labor-consumption, and insufficient reliability et al.

Compared with CFD method, numerical calculation forecast method based on potential flow theory costs much less and easily to be mastered, so it is normally used for engineering application. Many popular commercial tools, for instance, WAMIT, SESAM, MOSES, and AQUA, can be used to do the air gap forecast calculation. Nevertheless, it is also needed to point out that sometimes numerical calculation method can underestimate the relative wave height, and thus may not provide a conservative prediction result (Fig. 6).

There are some findings from air gap forecast research. It is found that linear numerical methods often underestimate the extremes of relative wave front rise, so basin model testing is essential for designing a new platform. Sweetman (Sweetman et al. 2002b) believes that it is more economical to allow proper negative air gap phenomena in the extreme sea conditions and to reinforce the local structure, compared to always be without wave slamming. Simos and Fajarra (Simos et al. 2006)

Semi-submersible Platform, Fig. 6 Wave slamming simulation for semi-submersible through CFD technology



from the University of São Paulo concluded that the plateau is affected by waves, and the air gap response of the platform will have significant nonlinear characteristics (especially the response wave height around the platform column) when the incident wave's steepness exceeds 4%. In 2009, Lwanowski et al. used ComFLOW software to conduct a theoretical study of the platform (Bogdan et al. 2009). The software predicts the air gap performance of the platform by capturing the N-S equation combined with the improved VOF method and captures the wave climbing and wave-ups around the platform. The results show that this method can simulate the wave scattering and wave surface distribution around the column accurately. Stansberg et al. used the potential flow software WAMIT to simulate the wave height and air gap around the semi-submersible platform (Stansberg 2007). It was found that the nonlinear method can basically meet the requirements of engineering forecast under some special conditions. However, when the wave steepness is large, there is a big deviation from the test value.

In addition, besides replying on calculation method, basin experiment is another alternative way, and it can give relatively more accurate and direct forecasting air gap data. However, similar to the disadvantage of CFD method, basin experiment has the shortcoming of high cost and human labor, which may not be affordable by most scholars.

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Semisubmersible Vehicle (SSV)

► Semisubmersible Vehicle Autopilot Control System

Semisubmersible Vehicle Autopilot Control System

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Synonyms

Autopilot control system; Semisubmersible vehicle (SSV); Unmanned surface vehicle (USV); Unmanned underwater vehicle (UUV)

Definition

The **semisubmersible vehicle** is a fuel-powered underwater vehicle that most of its structure is below the water surface and its snorkel extends above the water surface. The radio antenna is fixed on the snorkel for real-time data transmission and remote control. Different from traditional unmanned surface vehicles (USVs), the semisubmersible vehicle can use the differential global positioning system (DGPS) mounted on snorkel duct for precise positioning, which has greatly enhanced the opportunity for the use of the semisubmersible vehicle. As the snorkel is the only part of the semisubmersible vehicle to be affected

by sea state, it is much more stable in waves than surface drones or small vessels. The task of the semisubmersible vehicle autopilot control system is to achieve automatic attitude and depth control.

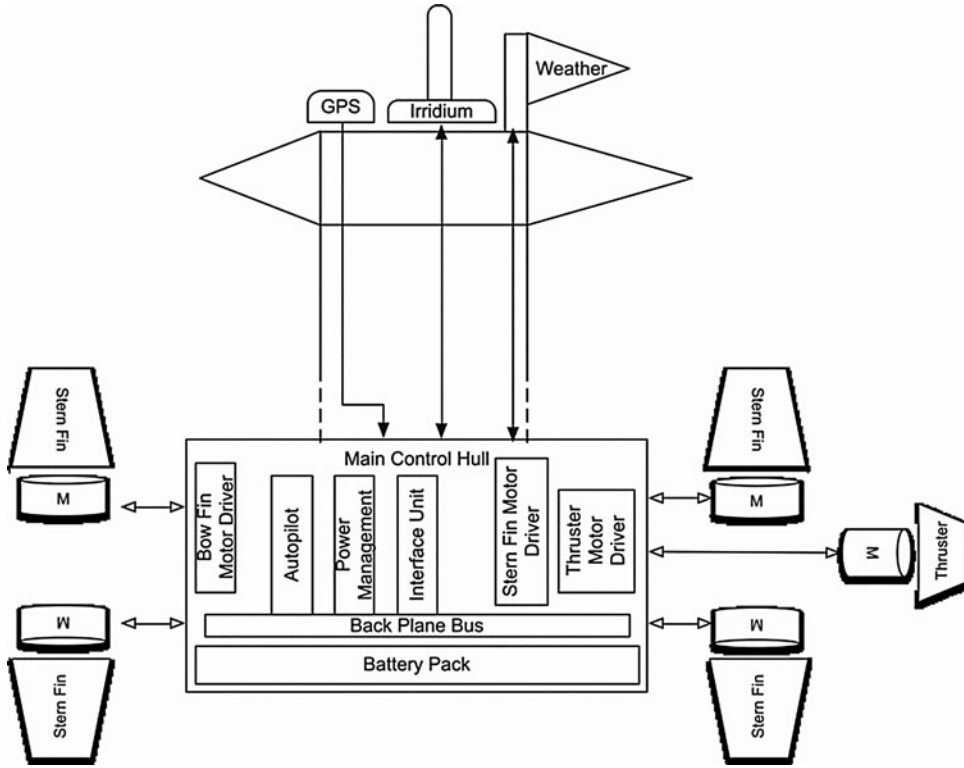
Scientific Fundamentals

The autopilot control system of semisubmersible vehicles faces three major challenges: some hydrodynamic coefficients unknown, disturbances from unpredicted sea environments, and the coupling states' effects. The dynamical performances of the vertical plane control are sensitive to the sharply changing of these coupling states and complex disturbances. Due to the highly nonlinear marine vehicles and the significant environment disturbances, it is a great challenge to control unmanned semisubmersible vehicles (USVs) without knowing the model parameters.

Sensor, Control and Communication, and Power System

As mentioned in the background above, one of the advantages of this semisubmersible is the capability to maintain the communication link to support vessel or shore station and location updated by GPS transceiver. As a categorized unmanned surface vehicle, the topmast of the surface section must have enough space to put some environmental sensors such as temperature, barometric pressure, speed, and wind direction or popular known as weather sensor station. With the most body submerged, it has enough space to put the most common underwater survey equipment such as obstacle avoidance sonar, side-scan sonar, and dual-frequency echo sounder. Figure 1 shows the control architecture of semisubmersible vehicle.

For communication, two types of systems will be provided on this vehicle. As a fact that almost the unmanned and autonomous capability surface vehicles have the mission to deploy on the remote area and very far from support vessel or shore station, the best communication must be based on the satellite system. The other widely used communication method is radio communication.



Semisubmersible Vehicle Autopilot Control System, Fig. 1 Control architecture of semisubmersible vehicle (Kartidjo et al. 2015)

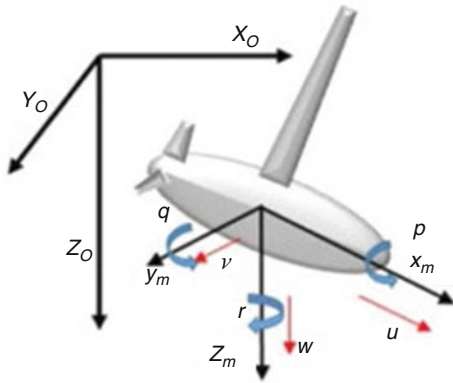
To accommodate the long-endurance capability and operation of many sensors, this vehicle is equipped with a high density of battery as the power source. The main criteria of battery beside the capacity are the longest cycle it has (Kartidjo et al. 2015).

6-DOF Equation of Motion of the Vehicle

We formulate the motion of the semisubmersible vehicle using a 6-DOF rigid body model. To

describe rigid body motion, two coordinate systems need to be defined: the global coordinate system and the body-fixed frame. Figure 2 shows the coordinate systems used for vehicle modeling. Based on Newton's second law, Kirchhoff's equation of motion in the body-fixed frame of the towing vehicle and the towfish can be written as shown in Eq. (1).

$$\begin{aligned}
 m_m [\dot{u} - vr + wq - x_{Gm}(q^2 + r^2) + y_{Gm}(pq - \dot{r}) + z_{Gm}(pr - \dot{q})] &= X_{hyd} + X_{prop} + X_{int} \\
 m_m [\dot{v} - wp + ur - y_{Gm}(q^2 + r^2) + z_{Gm}(qr - \dot{p}) + x_{Gm}(pq - \dot{r})] &= Y_{hyd} + Y_{prop} + Y_{int} \\
 m_m [\dot{w} - uq + vp - z_{Gm}(q^2 + r^2) + x_{Gm}(pq - \dot{r}) + y_{Gm}(pr - \dot{q})] &= Z_{hyd} + Z_{prop} + Z_{int} \\
 I_{xm}\dot{p} + (I_{zm} - I_{ym})qr + m_m[y_{Gm}(\dot{w} - uq + vp) - z_{Gm}(\dot{v} - wp + ur)] &= K_{hyd} + K_{prop} + K_{int} \\
 I_{ym}\dot{q} + (I_{xm} - I_{zm})pr + m_m[z_{Gm}(\dot{u} - vr + wq) - x_{Gm}(\dot{w} - uq + vp)] &= M_{hyd} + M_{prop} + M_{int} \\
 I_{zm}\dot{r} + (I_{ym} - I_{xm})pq + m_m[x_{Gm}(\dot{v} - wp + ur) - z_{Gm}(\dot{u} - vr + wq)] &= N_{hyd} + N_{prop} + N_{int}
 \end{aligned} \tag{1}$$



Semisubmersible Vehicle Autopilot Control System, Fig. 2 Coordinate systems of semisubmersible vehicle (Park and Kim 2015)

Eq. (1) is the equation of motion of the semisubmersible vehicle, where $[u, v, w]^T$ is the surge, sway, and heave directional linear velocity in body-fixed frame, and $[p, q, r]^T$ is roll, pitch, and yaw angular velocity in body-fixed frame. $[x_{Gm}, y_{Gm}, z_{Gm}]$ is the position of the center of gravity of the semisubmersible vehicle. The subscript *int* refers to the interaction force, the subscript *hyd* refers to the hydrodynamic force, and the subscript *prop* refers to the propulsion force. As mentioned earlier, the equation of motion of vehicles is described in the fixed-body frame (Park and Kim 2015).

In the actual control system, we pay more attention to the vertical control of semisubmersible vehicles. So the dynamic model can be simplified properly. The vertical plane states of the semisubmersible, such as the depth, pitch angle velocity, and pitch angles, are usually coupled with surge speeds. And these states are often corrupted by the unexpected surge or the sea wind. Moreover, pitch angles and submerged depths should be confined according to the mechanisms of the studied semisubmersible vehicles system.

PID Controller

As the starting point and to provide a baseline for comparison of designs, a PID control law was employed. In the digital domain, this control law has the form:

$$\delta_z(k) = k_P e_z(k) + k_I \sum_{l=1}^k e_z(l) + k_D \frac{e_z(k) - e_z(k-1)}{T}$$

$$e_z(k) = z(k) - z_d \quad (2)$$

Zhou et al. (2019) set the sampling period to $t = 0.2$ S and set the submergence depth of the vehicle to 1 m in the control system. The control parameters used in the control system are:

$$k_P = 3, k_I = 0.5, k_D = p$$

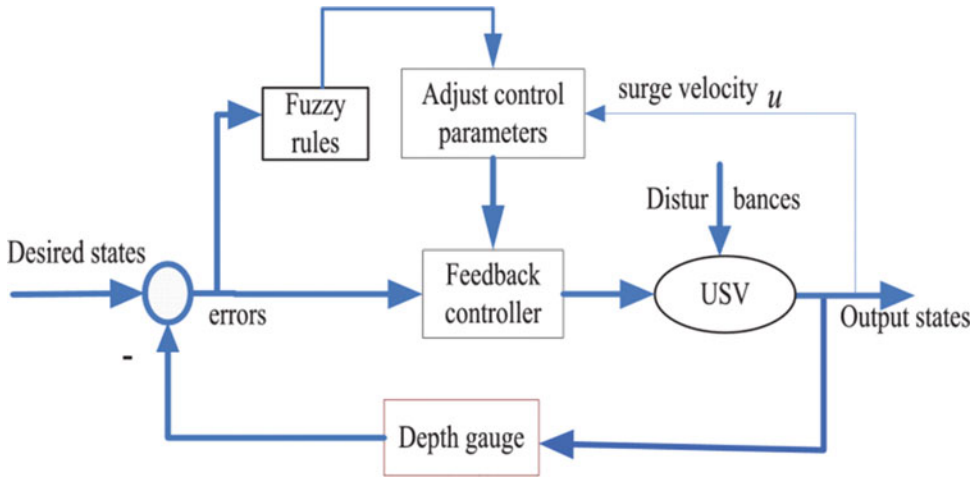
Although the response of the PID controller is stable, the transient and steady-state response is somewhat poor. Moreover, the pitch angle is not well controlled, appears as the semisubmersible vehicle is running through the sea, which indicates that the depth of the vehicle is susceptible to the disturbance from the sea environment.

Fuzzy Dynamic Feedback Controller

The fuzzy dynamic feedback control algorithm can guarantee robustness against external parameter unknown and poor sea environment. Due to the limited submersible depth, motions of the semisubmersible vehicle are directly influenced by wave forces coming from water surfaces. Therefore, the external disturbances on the semisubmersible vehicle are difficult to be described by mathematical expressions. Moreover, the model parameters of the semisubmersible vehicle are unknown, which makes the controller more difficult. Control parameters of the fuzzy rule-based state-feedback controller are auto-adjusted by fuzzified states errors and surge speeds. Figure 3 depicts the block diagram of the depth control of the semisubmersible vehicle.

State Feedback Controller

One alternative to PID control is to assume that the system can be adequately represented by linear time-invariant dynamics for a specified surge speed and design a state feedback law for the vehicle. Simplify the system model and describe its state-space model as:



Semisubmersible Vehicle Autopilot Control System, Fig. 3 Fuzzy controller of the semisubmersible vehicle (Zhou et al. 2019)

$$\begin{aligned} \begin{bmatrix} \dot{e}_z \\ \dot{e}_\theta \\ \dot{e}_q \end{bmatrix} &= Ax + B\delta_s \\ &= \begin{bmatrix} 0 & -u & 0 \\ 0 & 0 & 1 \\ 0 & a_1 & a_2u \end{bmatrix} \begin{bmatrix} e_z \\ e_\theta \\ e_q \end{bmatrix} \\ &\quad + \begin{bmatrix} 0 \\ 0 \\ bu^2 \end{bmatrix} \delta_s \end{aligned} \quad (3)$$

The key idea of the feedback control is to assign the eigenvalues predesigned according to the desired performances of the system. The eigenvalues are equal to the roots of the character equation of the considered system if there is no pole-zero cancellation. The poles of the closed-loop system can be located at any place according to the desired control performances if the plant system is fully controllable (Gruyitch 2013). If a state feedback control law is designed for a given surge velocity, the pole assignment problem has a solution if and only if (3) is controllable. Preset the form of the feedback control law as:

$$\begin{aligned} \delta_s &= k_{e_z}e_z + k_{e_\theta}e_\theta + k_{e_q}e_q \\ &= [k_{e_\theta} \ k_{e_\theta} \ k_{e_q}] \begin{bmatrix} e_z \\ e_\theta \\ e_q \end{bmatrix} = K \begin{bmatrix} e_z \\ e_\theta \\ e_q \end{bmatrix} \end{aligned} \quad (4)$$

where k_{e_z} , k_{e_θ} , and k_{e_q} are feedback control parameters. Suppose also that it is decided to place the controlled system poles at λ_{e_z} , λ_{e_θ} , and λ_{e_q} in the open left-half of the complex plane.

$$\det(sI - (A - BK)) = (s - \lambda_{e_z})(s - \lambda_{e_\theta}) \times (s - \lambda_{e_q})$$

With further analysis, the control parameters can be determined as:

$$\begin{cases} k_{e_z} = \lambda_{e_z}\lambda_{e_\theta}\lambda_{e_q}\tilde{b} \\ k_{e_\theta} = -(\lambda_{e_\theta}\lambda_{e_q} + \lambda_{e_z}\lambda_{e_\theta} + \lambda_{e_z}\lambda_{e_q})\tilde{b} \\ k_{e_q} = (\lambda_{e_z} + \lambda_{e_\theta} + \lambda_{e_q})\tilde{b} \end{cases}$$

where $\tilde{b}$ is decided by the known parameter M_σ (Zhou et al. 2019).

Sliding Mode Controller

The control parameters cannot be provided for the model-based control algorithm, which leads to the

Semisubmersible Vehicle Autopilot Control System, Fig. 4 Dorado vehicle (ISE Ltd. 2017)



undesirable control performances of the system since the depth motion model parameters regarding the unmanned semisubmersible vehicles are unknown. For this purpose, a multi-identification model-based dynamic sliding mode control algorithm is presented for the investigation of system's control problem on the desired depth. In the proposed algorithm, the average-fitting-error method is used to reduce the excessive redundant model parameters, and thus the optimum model parameters are provided for the control algorithm by switching methods. And the state feedback method is used to bring about the chattering exponential decay of sliding mode controller so as to reduce the settling time. The experimental results of lake trials demonstrate that the proposed algorithm could offer the best model parameters for sliding mode control, and the dynamic sliding mode control with multi-identification models is capable of ensuring BQ-01 vehicle to achieve the ideal control performance (Zhou et al. 2017).

Key Applications

Semisubmersible vehicles were first used for military purposes, mainly in anti-mine operations. Only a few countries have mastered the relevant technology and developed equipment, including Canada, France, the United Kingdom, the United

States, and China. In recent years, with the increasing demand for marine surveys, semisubmersible vehicles may expand into the civilian field, for example, ocean seismic survey work.

Dorado Vehicle

The growing threat from new-generation mines that are increasingly more difficult to detect has stimulated the demand for new mine-hunting systems that offer improved safety and operational efficiency. The Dorado vehicle, shown in Fig. 4, in operational service with the Canadian Department of National Defence, was developed to meet this need, which can be considered one of the first examples of truly autonomous USVs (Young et al. 2006). The Dorado vehicle is capable of towing a sonar towfish at speeds up to 12 knots and depths to 200 m. The towfish was developed to follow the seabed at low altitudes and is fitted with a multi-beam side-scan sonar.

SeaKeeper Vehicle

Based on the experience in the design and production of several generations of MCM vessels, the French company DCN had developed the SeaKeeper, shown in Fig. 5, as an answer to face the challenging threat of modern mines. SeaKeeper was presented for the first time in Brest in June 2003 in the complete configuration including the “dry segment.” SeaKeeper was the

**Semisubmersible Vehicle
Autopilot Control
System, Fig. 5** SeaKeeper
vehicle (ISE Ltd. [2018](#))



Semisubmersible Vehicle Autopilot Control System, Fig. 6 AN/WLD-1 vehicle (Luca [2017](#))

latest solution available for sonar reconnaissance against the most modern mines at that time. With a vehicle remotely controlled by radio, it offered the highest level of safety for the crew, together with unmatched performance in terms of area coverage rate. The system, integrated by DCN company, had raised interest in several navies in the world. During 13 days of trials and demonstrations, the

SeaKeeper achieves more than 100 h of sonar operation, showing remarkable reliability for the prototype (Waquet [2003](#)).

AN/WLD-1 Vehicle

Lockheed Martin's AN/WLD-1 RMS, shown in Fig. 6, is a key component of the Littoral Combat Ship's mine countermeasures package (Button

et al. 2009). The system consists of a 7-m semi-submersible autonomous remote multi-mission vehicle (RMMV) which tows the AQS-20A Variable Depth sensor. The RMS has sometimes been referred to as a “snorkeler class” unmanned surface vessel due to the snorkel required above the surface for the system’s diesel engine operation. The RMMV can be preprogrammed to perform autonomously or be manually controlled via line of site or over-the-horizon data link. The Navy successfully concluded the second and final phase of reliability testing of the littoral combat ship (LCS) remote minehunting system (RMS) off the coast of Palm Beach, Florida, enabling the service to progress toward developmental testing, in June 2013. The system completed more than 850 h of testing during 47 missions over a 4-month period.

SASS Vehicle

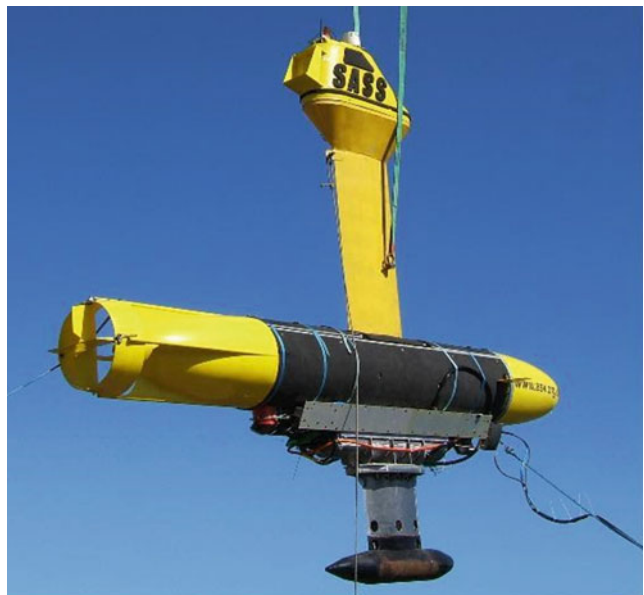
The Survey Autonomous Semi-Submersible (SASS) vehicle, shown in Fig. 7, for Ocean survey was developed by UK company ASV Ltd. (Bertram 2008). The platform which is intended to enhance or even replace conventional survey vessels will provide cost-effective deployment of ocean sensors. SASS involves an unmanned submerged torpedo-like body running below waves with a strong upright strut penetrating the water

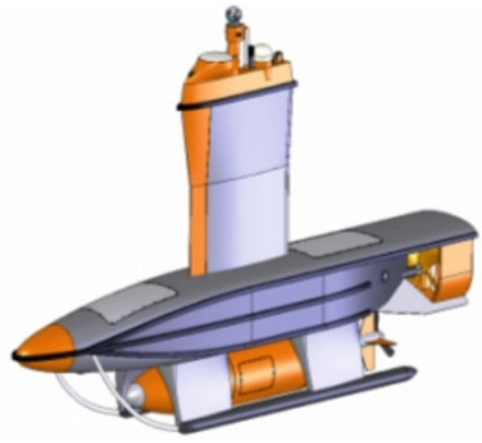
surface. The strut provides buoyancy, air for propulsion, and radiofrequency coverage for communications and positioning. The 6-m vehicle has a maximum speed of 10 knots and a payload capacity of up to 200 kg. It is designed to be launched and recovered from a ship or deployed from the shore. The vehicle permits a wide range of sensor operations for near-surface and deepwater applications. To date, the vehicle has been successfully operated with sonar’s for the underwater survey and with cameras for above water surveillance. Furthermore, the vehicle has been a leading candidate for towed side-scan sonar operations such as deepwater naval mine warfare, and the vehicle has been evaluated by the UK’s Ministry of Defence under their concept capability demonstrator program.

USS Vehicle

USS vehicle, a predecessor to the Remote Minehunting System, was a diesel-powered air-breathing vehicle with a tall narrow mast and the majority of its structure below the water surface (Alleman et al. 2009). The USS vehicle, shown in Fig. 8 (left), is an autonomous semisubmersible vehicle that could be used for shallow-water hydrographic surveys and surface surveillance. This diesel-powered vehicle has an operational speed of

**Semisubmersible Vehicle
Autopilot Control
System, Fig. 7** SASS
vehicle (Bertram 2008)





Semisubmersible Vehicle Autopilot Control System, Fig. 8 USS 6300 vehicle and conceptual model of USS 9500 (Alleman et al. 2009)

Semisubmersible Vehicle Autopilot Control System, Fig. 9 BQ-01 vehicle (Zhou et al. 2019)



8 knots with an associated endurance of 48 h in sea state 4. At a survey speed of 4 knots, the vehicle has an impressive endurance of nearly 100 h. The vehicle has a payload capacity of 300 kg for sonars, computers, sensors, winches, and camera equipment. A recently constructed vehicle, the USS 6300 for C&C Technologies in the United States, is fitted with a Klein 5000 side-scan sonar and Simrad EM3002 multibeam sonar for commercial hydrographic survey. USS 9500 will be designed to offer long-range (1500 miles) survey and

surveillance operations within the “Exclusive Economic Zone” (EEZ); this 9.5-m diesel-powered vehicle has been designed to a rugged specification, with the ability to operate in sea state 5/6 at moderate speeds, with a sprint speed of 20 knots and the ability to remain at sea for extended periods of up to 30 days.

BQ-1 Vehicle

BQ-01 system, shown in Fig. 9, is a semisubmersible vehicle that should not completely submerge

into water, or else some water may flow into it from its chimney and breaks down some important parts equipped inner system. To make sure the USV safety, the pitch angles, and the submerged depth are confined to $[-15, 15]$ degrees and $[0, 3]$ m, respectively. The features of BQ-01 are as the followings: its power source is a diesel engine, its body can submerge a certain depth, its breather pipe should not be awash, otherwise some water will enter its control cabinet. The length of BQ-01 is 5.9 m, and the width is 2 m. Its performance basic indices are the maximum surge speed 15 knots, the maximum submerged depth is 3.5 m, and the strongest drag force is 10,000 N@10kn (Zhou et al. 2017, 2019).

Cross-References

- [Autonomous Underwater Vehicle \(AUV\)](#)

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Sequential Engineering

- [Design Spiral](#)
- [Empirical Design](#)

Service Ships

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Definition

Service ships are vessels designed to provide support for commercial ships and industrial vessels; some of them are also used for professional work.

Scientific Fundamentals

Composition and Basic Type

There are many types of work ships depending on the function, such as tugs, icebreakers, fire boats, etc.

A tugboat is a type of vessel that maneuvers other vessels or floating structures. The vessel itself does not have the ability of loading and unloading. And it is mainly used to tow barges loaded with cargo and other different kinds of engineering ships. Towing facilities include towing hook,



Service Ships, Fig. 1 Inland river tug (<http://www.quanjing.com/>)



Service Ships, Fig. 2 The icebreaker "Bolheim"

towing column, mooring winch, etc., and at the same time, the aft deck has a large working area for operating easily (Wu 2017) (Fig. 1).

Icebreakers are special vessels designed to open waterways in ice waters and to rescue trapped vessels. As early as 1857, Sweden built the world's first real icebreaker "Bolheim". The hull structure of an icebreaker is quite different from that of a general ship. The ship type, bow, propulsion, and propeller are the main distinguishing areas.

Icebreakers generally use two methods (continuous icebreaking and collisional icebreaking) for icebreaking operations. In addition to the use of

their own weight, traditional icebreakers often need to use the cooperation between the front and rear pressure tanks to complete the icebreaking work (Fig. 2).

Fireboats are professional ships for firefighting and extinguishing in ships or buildings near the shore. Fireboats look like tugboats, so there are also fire boats that double as tugboats. The ship is equipped with high-power water pump system, high-pressure spray gun, and fire-extinguishing agent, as well as rescue workers and medical equipment. Fire hoses are located on fire towers high above the water or on the top of a thickened mast with a range of more than 40 m. The ship



Service Ships, Fig. 3 Fire of ocean-going freighter (<http://www.sina.com.cn/>)

also has a water curtain in order to get deeper into the fire area. When entering the fire area, the whole ship will be covered by a water curtain. Fireboats have a high speed and good seakeeping and require good maneuverability, so they can perform firefighting missions in narrow waterways or crowded ports (Fig. 3).

Development History

In the seventeenth century, ships began to use steam engines. The first truly tugboat was the *Charlotte Dundas*, which was powered by a steam engine and paddle wheels, mainly along the coast of Forth and Clyde near Glasgow. In order to improve the efficiency of propulsion, the propellers began to appear on tugboats in the mid-nineteenth century, and the size and towing capacity of the tugboats also increased significantly. The surge in demand for tugs in the first and second world wars drove the development of tugs, and the pace of tugboat modernization was accelerated by the development and applications of diesel engines, propulsion technologies, navigation technology, etc. Today, tugboats still play an important role in port operations, inland water transport, and marine development (Chen 2010).

Since 1900, Russia has been the first to build icebreakers on a large scale. Most of these icebreakers are used for overseas sailing. Among the early construction of icebreakers, the first vessel to sail in the Arctic was the “*Yelmaf*”. In 1957, the world’s first icebreaker driven by nuclear power was successfully built. It was the “*Lenin*” built by the former Soviet Union. Nuclear power technology is a major breakthrough in the field of icebreaking.

In 1969, the construction process of modern icebreakers was rapidly developed. The USA used these new technologies to transform the transport ship “*Manhattan*” into a 115,000 t commercial icebreaker. Since then, a large number of high-quality icebreakers have emerged. In 1980, Canada built a new type of icebreaker, the “*Kigoglik*”, it mainly uses a spoon-shaped head similar to a reamer, which can greatly reduce the friction between the broken ice crystal and the hull, thus effectively improving the ship’s swing performance. In the 1980s, scientists invented a “bubble drag reduction system” that would prevent the icebreaker from being caught by the ice when it was icebreaking, allowing the ship to continue in the ice.

In 1762, the first fireboat appeared in Paris. At that time, it was mainly to deal with the fires that occurred on the wooden bridge over the Seine and the fires that occurred on the boats. By 1764, there were a total of eight fireboats in Paris which were equipped with handcuff pumps. They did not quit service until the nineteenth century. The rise of the industrial revolution in the nineteenth century brought the busyness of American ports, which increased the risk of fire, making Boston introduce lifeboats for firefighting. In 1888, the Paris Fire Brigade Commander asked the municipality for a steam fireboat, but his request was not met. Until 1931, due to the soaring fuel prices, in order to reduce costs, fireboats began to use diesel drives instead of steam drives (Shi 1990).

Key Technology

The tugboat is characterized by its small hull, high power, high tractive force, solid structure, strong fenders, and anti-collision equipment, which make the vessels have good stability and maneuverability. Meanwhile, excellent barking performance is very necessary for the tugboat. During towing operation, the tugboat will have a large heeling because of transverse urgent traction from towing cable. Therefore, in order to ensure the transverse stability, the beam will be relatively large and the length-breadth ratio will be smaller

than that of the ordinary transport ship (Yang et al. 2010).

In order to keep the good speediness and reliability of tugboats, double engine-paddle propulsion system is basically applied in modern towboats and sometimes especially for four engine-paddle propulsion system. Meanwhile, adjustable pitch paddles have been utilized in most of boats. For keeping excellent maneuverability, lateral thruster and sometimes full-swing devices will be installed in bow and stern of most towboats, respectively (Fig. 4).

Icebreakers generally use two methods: continuous icebreaking and collision icebreaking. In addition to using their own weight, traditional icebreakers generally need to use the cooperation between the front and back water tanks. For example, when heading to break the ice, the back water tank will be filled with water, so that the ship will be upward-headed, so that the front part of the ship will be pressed on the ice, and then the forward water tank will be filled with water, and the back water tank will be drained, so as the ship's center of gravity moves forward, the purpose of crushing the ice will be achieved. When breaking the ice from left to right, the ballast tanks of the ship will be flooded alternately from left to right, which will make the ship shake from left to right, thus breaking the ice (Bostrom 2018) (Fig. 5).



Service Ships, Fig. 4 Operations of tug (<http://www.quanjing.com/>)



Service Ships, Fig. 5 The icebreaker (<http://www.baidu.com>)

When the thickness of ice does not reach the designed thickness of icebreaker, the continuous icebreaking method is adopted. Propelled by propellers, icebreakers, which slowly and undulately break through the ice, can sail about 9.2 km/h through the ice. If the ice is thicker than the maximum ice break the icebreaker is designed to be, or if it is in more complex ice accumulation, thick ice zones, and more severe ice ridges, then the impact icebreaking method is used. First, the icebreaker needs to move back a distance of one or two captains and then speed up the collision. The tilting bow of the icebreaker can easily run onto the ice surface, in the thick ice area or encountered ice ridge, through repeated impact of the method of breaking the ice (Wei et al. 2017).

The advanced external fire protection system is the core part of the firefighting vessels. External fire protection system consists of external fire-extinguishing system, external foam fire-extinguishing system, self-spraying water curtain protection system, hydraulic lifting device, and external fire monitoring system. The advanced system design can transfer the ship from the navigational state to the fire state in the shortest time. The arrangement of the external firefighting system should have a certain height from the surface of the water due to the consideration of the scope

of action, and it should be arranged more in the top area of the firefighting vessels, so it is often arranged in the top area of the fireboat. Taking account of the personal safety of the firefighters, remote control of the cab should be used as much as possible, unless manual operation is necessary (Fig. 6).

Main deck and left and right sides of the upper deck of firefighting vessels shall be provided with fixed water-spray system devices, protecting the hull, superstructure, and deckhouses with water-spray, including the vertical surface of the base and parts of the fire monitor. Water curtain protection can absorb part of the convection and radiation heat, which can protect firefighters from being burned by large fires. Moreover, it can also prevent glass windows from being cracked by fire. At the same time, fireboat shells can be cooled (Shi 2011).

The hull and superstructure of firefighting ships and the peripheral walls of deckhouses must be steel. Doors and window frames on all peripheral walls must also be steel (aluminum). Firefighting vessel's glass windows (especially the big glass window in the cab) should be equipped with tempered glass that is resistant to air blasts to avoid accidental casualties of crew members and firefighters.



Service Ships, Fig. 6 Firefighting vessels (<http://www.pic.sogou.com/>)

Key Applications

With the advancement of technology and changes in people's needs, service ships are constantly evolving, and the overall trend is modernization, specialization, and multifunctionality.

Along with the drafting object increasing and improved requirement for drafting efficiency, the function for tugboat has been enhanced continuously. More advanced equipment has been utilized, and the tugboats have already entered the scope of high value-added ships. The number of newly built tugboat has already reached an unprecedented level, and the development of tugboats has directed to a functional and intelligent orientation (Tang et al. 2016).

The main shipyards for tugboats manufacturing are mainly located in European and American nations and China. The total number of tugboats under construction is currently around 400, and it is the USA which possesses most of the tugboats. There are several decades of history for tugboats manufacturing in our country, and enriching experience has been gathered. The design and manufacturing technology for tugboats has achieved great progress and reached the international advanced level mainly including harbor

tugs, offshore and inshore towboats, and multi-purpose stand by ship serving offshore oil platform (Fu 2010).

Versatility

Due to the restrictive factors in economics, there are drafting functions as well as fire control, marine salvage, supply, and diving work functions in towboats which present the multifunctionality trend.

For instance, the “De Hong” tugboat which was designed by Shanghai Merchant Ship Design and Research Institute in China is mainly used in marine salvage including marine long-distance drafting for salvage boats, wrecked offshore drilling platform, stranded reef ships, and boats lacking of maneuverability. Meanwhile, there are also plugging, drainage, and firefighting ability to wrecked ships, setting sail and anchoring ability for marine platform, drafting ability for buoy and undulant wrecked boat, boat hauling and transportation, and marine residuum collection functions. Moreover, bigger cabin has been arranged during preliminary design phase to load the deck cargo and lance pipe so that this ship can also be used as supply boat.

Intelligentization

Along with the increasing tasks for towboats and burden reducing requirement for the crew, the various systems on ships are needed to keep pace with the times and become more intelligentized and automated.

For instance, the VS468 multifunctional tugboat has already met the DNV's unmanned cabin on duty requirement. Meanwhile, one respective 10 and 3 cm radars, electronic chart, automatic navigator, echo detector, doppler log meter, electronic gyrocompass, and clinometer have been devised as well. Monitoring and remote control can be set to all pump valves including freshwater, drilling water/ballast water, fuel, brine, and methanol. Moreover, alarm and control system can be offered to safe operation of master and auxiliary equipment, and the engine room "death alarm" function is also configured. And skills in unmanned cabin, automatic steering, computer monitoring, and a series of modern communication navigation equipment have been applied in the towboats which are manufactured by shipyards in Zhenjiang and exported to Egypt (Giron-Sierra et al. 2015).

The patents and literatures of various new icebreakers are summarized and analyzed, and some possible design conjectures of new icebreakers are put forward. The icebreaker innovatively designed the bow of a ship with a narrow top and wide bottom, which allows the bow to be easily extended below the ice. When the icebreaking operation is carried out, the ship tilts forward at a small angle, and the conical structure of the ship's bow underwater reaches below the ice sheet. Through the action of the ship's ballast water system, the distribution of ballast water is changed to make the ship's center of gravity move back. The forward hull generates forces that push the conical structure forward, which impacts the ice in both the forward and upward directions, thus realizing the "skid-type" icebreaking process.

The idea of a semisubmersible icebreaker was first proposed by Vadim Pikeur, a former Soviet engineer, who proposed an icebreaker that could sail on the surface as well as semisubmersible through specially designed reinforced hulls and "comb" icebreakers. The specific icebreaking method is the lower hull reaches the

predetermined depth and then the ballast water is quickly discharged to make it float and break through the ice with the "comb" type icebreaking mechanism. The main structure of the icebreaker is a lower hull equipped with buoyancy module, ballast tank and main and auxiliary equipment, and a tower of about 20 m. The tower is equipped with a bridge, control system, etc. The icebreaking capacity of semisubmersible icebreakers will be greatly improved compared with current icebreakers, and the icebreaking operation of 5-m-thick ice will be carried out. At the same time, because the propeller is under the ice, it can improve the performance of the propeller.

The mission of the fireboat is clear, and it is inevitable that the city port must be equipped with the supporting fireboat. Meanwhile, the focus of the design of the fireboat is mainly to select a reasonable host propulsion, a fire pump drive mode, an excellent performance, advanced and reasonable firefighting equipment, and an excellent hull line with good resistance performance. The future development of fireboats also needs to reflect the particularity of the image of the city government. People's modern aesthetic requirements are constantly improving, and the appearance of fireboats will be very important. In the new situation, modern fireboats have been given a new content and meaning in addition to the necessary fire-extinguishing technology, which means they are used to protect the water environment from pollution.

Cross-References

- [Ship Overall Design](#)
- [Ship Structural Design](#)
- [Special Marine Vehicle](#)
- [Transport Ship](#)

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SeSAm

- [Jacket Platform](#)

SGPC - Salinity Gradient Power Conversion

- [Salinity Gradient Power Conversion](#)

Shallow Foundations

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Synonyms

[Combined vertical, horizontal and moment loading](#); [Concrete foundation template](#); [Gravity-based structures](#); [International Organization for Standardization](#); [Offshore wind turbine](#); [Pipeline end manifold](#); [Pipeline end termination](#); [Simple shear](#);

[Tension leg platform](#); [Triaxial compression](#); [Triaxial tension](#); [Unlimited tension interface](#); [Water depth](#); [Zero-tension interface](#)

Definition

Shallow foundations applied offshore are considered as having an embedment depth to foundation diameter (width) ratio less than unity.

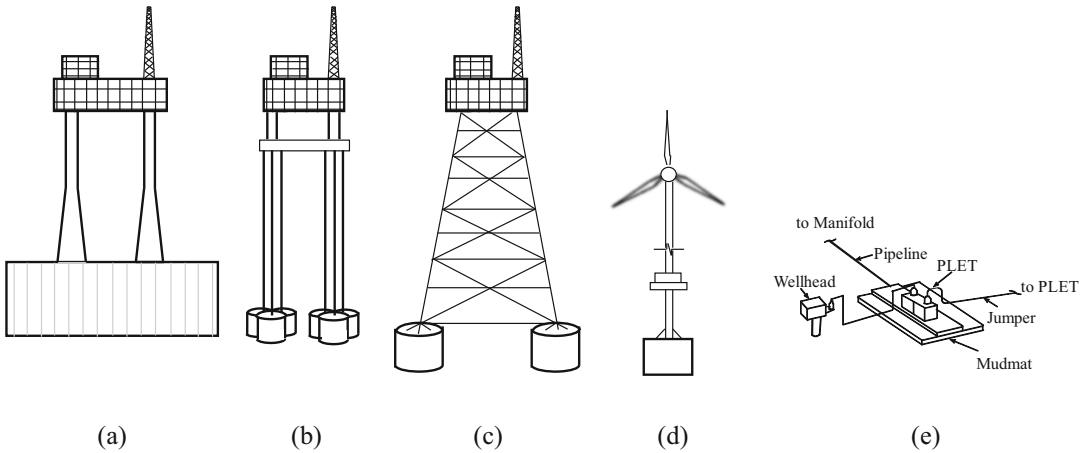
Design Approach of Offshore Shallow Foundations

Historically, offshore shallow foundations can either be large concrete gravity bases to fix the giant superstructures, or the steel mudmats used as temporary support for jacket platforms prior to piling. As the offshore oil and gas developments move into deep water, shallow foundations have become more diverse, including bucket foundations used as anchors mooring floating platforms and skirted mudmats as permanent supports for subsea infrastructures such as pipeline end terminations and manifolds (PLETs and PLEMs). Recent upsurge in exploiting offshore wind as a clean and renewable energy resource around the world makes the skirted foundations a more attractive option for the foundations of offshore wind turbines in deep water, either as a single foundation or in a multipod arrangement. A number of different applications of offshore shallow foundation systems are illustrated in Fig. 1. The partial safety factor approach, referred as to load and resistance factor design approach (LRFD), is generally preferred in routine design. To allow the overall reliability of a design to be quantified better, the applied loads to the foundations are generally scaled up by the load factors, and the shear strength of the soil is reduced by the material factors.

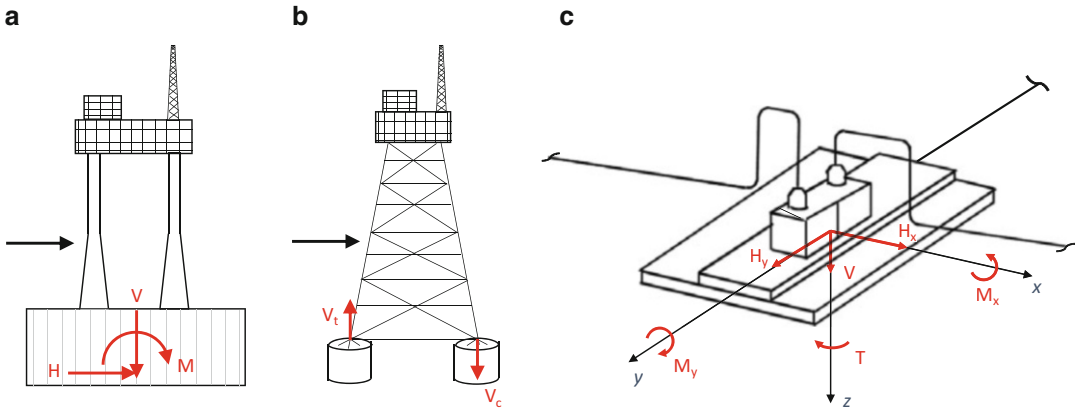
General Loads and Soil Parameters

General Loads

The general loads applied to a gravity-based foundation under working conditions consist of



Shallow Foundations, Fig. 1 Application of offshore shallow foundations: (a) GBS (tank-type structure); (b) TLP; (c) jacket; (d) OWT; and (e) subsea mudmats



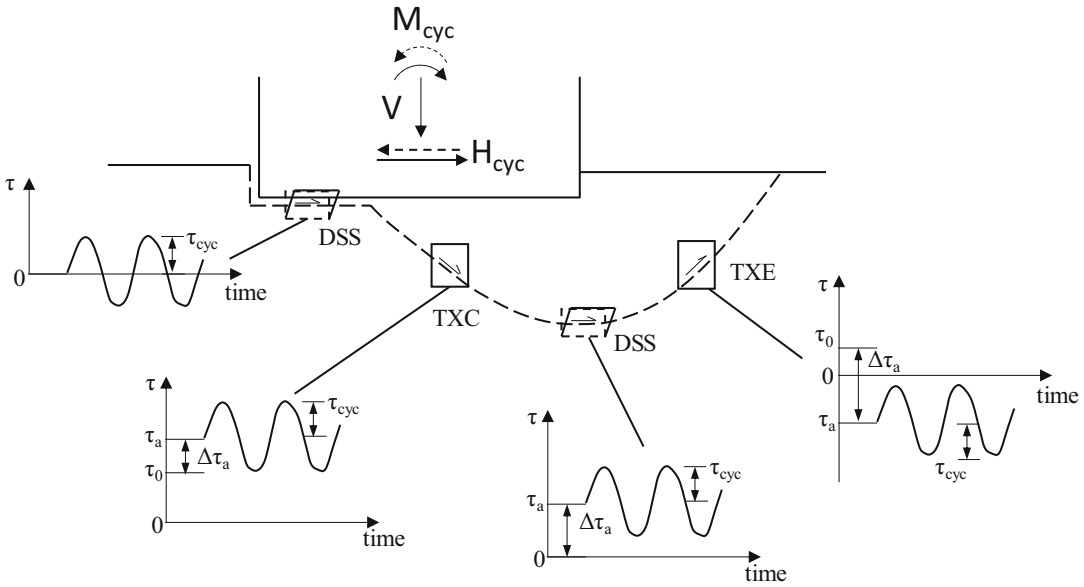
Shallow Foundations, Fig. 2 General loads applied to offshore shallow foundations

vertical load V , horizontal load H , and moment M (Fig. 2a). The vertical component mainly comprises the weight of the platform and the foundations, whereas the horizontal and moment components are typically due to the environmental forces from wind, waves, and currents. The ratio of M/H indicates whether a structure is prone to sliding or overturning failure. Concrete bucket foundations used for TLPs or the steel bucket foundations in a multipod arrangement are generally subject to tensile loads due to the “push-pull” mechanism (Fig. 2b). For the subsea mudmats supporting PLETs and PLEMs, the loads acting on the foundations typically include the vertical load V , the biaxial horizontal load H_x

and H_y , the biaxial moments M_y and M_x , and the torsion T (Fig. 2c). The biaxial horizontal loads are induced by the thermal expansion and contraction of the attaching pipelines and jumpers, whereas the moments and torsion are generated by the vertical and horizontal eccentricities. The vertical load is relatively small to the ultimate bearing capacity of the foundation, while the torsional and horizontal sliding or the overturning failure generally governs the design.

Soil Shear Strength Used for Design

The relevance of stability calculations depends on the choice of appropriate shear strength parameters. The stress paths in the soil under a foundation



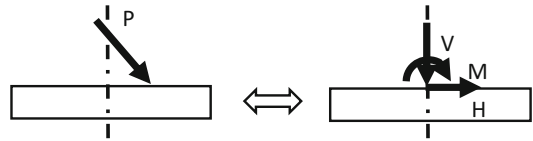
Shallow Foundations, Fig. 3 Simplified stress conditions along a potential failure surface beneath a shallow foundation

are complex. Figure 3 shows that the relevant shear strengths at different locations along a potential failure surface can be approximated to those measured in triaxial compression (TXC), simple shear (SS), or triaxial extension (TXE), which may differ by a factor of two. In engineering practice, most designs adopt an average shear strength in TXC, SS, and TXE conditions to use in standard design approaches. The environmental loading applied to an offshore foundation is cyclic in nature. Therefore, the effects of cyclic loading on the available soil shear strength have to be considered (Andersen and Lauritzen 1988).

Bearing Capacity

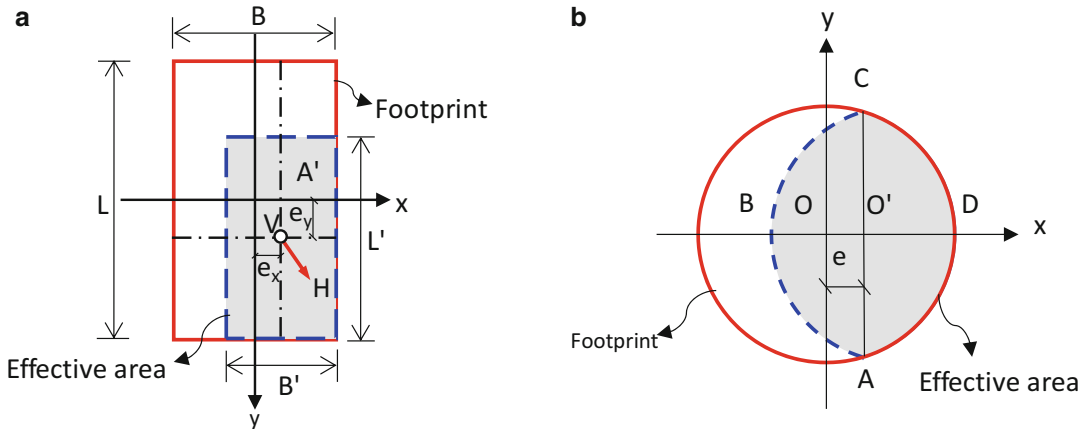
Classical Bearing Capacity Theory

Figure 4 depicts a shallow foundation subjected to a point load P . The eccentric-inclined load P is decomposed into a vertical component V normal to the base and a horizontal component H , intersecting the center of an equivalent fictitious rectangle for which the bearing capacity is calculated.



Shallow Foundations, Fig. 4 Load-equivalent transformation

The “effective width” hypothesis was proposed by Meyerhof (1953) for strip foundations to consider the detachment of the foundation due to an overturning moment resulting from eccentricity of vertical load. The concept of an “effective area” has since been developed for other foundation geometries and more complex loading conditions – including biaxial moment (Hansen 1961; Taiebat and Carter 2002), as shown in Fig. 5. The traditional bearing capacity theory calculates the load-carrying capacity of shallow foundations under combined loading by adopting the concept of effective foundation area in conjunction with modification factors for load inclination, and other boundary conditions including foundation shape and embedment, and soil strength heterogeneity (ISO 2016).



Shallow Foundations, Fig. 5 Definition of effective area for various foundation geometry

Undrained Bearing Capacity

The classical approach for predicting the design unit-bearing capacity is conventionally expressed as Eq. (1) for undrained conditions with uniform isotropic undrained shear strength increasing approximately linearly with depth below mudline (ISO 2016).

$$q_d = F \left(N_c s_{u0} + \frac{kB'}{4} \right) \frac{K_c}{\gamma_m} \quad (1)$$

where q_d is the design vertical bearing resistance ($Q_d = q_d A$), F is a correction factor given as function of kB'/s_{u0} , N_c is the undrained plane strain-bearing capacity factor for uniform shear strength ($\pi + 2$), s_{u0} is the undrained shear strength of soil at the foundation base level, and k is gradient of the increase of the undrained shear strength with depth. The superposing modification factor K_c accounts for load inclination, foundation shape, and foundation embedment.

Drained Bearing Capacity

Equation (2) is a general formula for determining design vertical bearing capacity for drained conditions

$$q_d = \frac{1}{2} \gamma' B' N_\gamma K_\gamma + \sigma'_{v0} (N_q - 1) K_q \quad (2)$$

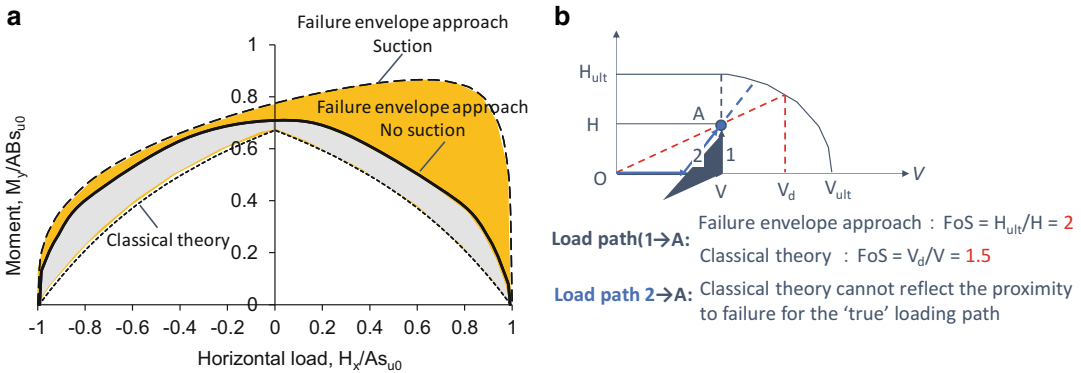
where N_γ and N_q are the drained bearing capacity factors; γ' is the submerged unit weight of soil;

σ'_{v0} is the in situ effective overburden stress at base level; and K_γ and K_q are the correction factors that account for load inclination, foundation shape, and foundation embedment.

Shortcomings of the Classical Bearing Capacity Theory

Shortcomings of the classical bearing capacity approach for application to offshore shallow foundations are illustrated in Fig. 6 and include that:

- The approach uses linear superposition of separate modification factors to account for load inclination and eccentricity. Therefore, the true coupling of horizontal and moment interaction leads to underprediction of load carrying capacity (Fig. 6a).
- The approach uses the effective area method to account for moment which does not allow for transient tensile resistance mobilized by the generation of negative excess pore pressure (Fig. 6a).
- The approach defines the load-carrying capacity by the single equivalent allowable vertical load, V_d , adjusted to account for horizontal load and moment (Eqs. (1) and (2)). The factor of safety determined by this approach cannot reflect the true load path for a foundation (Fig. 6b).



Shallow Foundations, Fig. 6 Shortcomings of the classical theory

Advanced Approach

Failure Envelopes

As an alternative, the failure envelope approach has been recommended by the ISO guidelines (ISO 2016) for calculating the load-carrying capacity of shallow foundations under multidirectional loading conditions. The failure envelope describes a surface in multidirectional load space which defines the combinations of loads causing failure. A load state that falls within a failure envelope is permissible while a load state that falls outside a failure envelope is non-permissible. A load state that falls on the failure envelope indicates a factor of safety of unity, i.e., all available capacity is mobilized.

Benefits of the failure envelope approach over classical bearing capacity theory include overcoming limitations (a) – (c) listed in the last section, specifically that:

- The approach accounts for inclination and eccentricity of loading by defining capacities under pure moment and horizontal loading (which form apex points of the envelope), rather than by adjustment of the vertical capacity.
- The approach allows visual evaluation of the proximity to failure and the factor of safety against any combination of loads, illustrating visually whether a given load is favorable or unfavorable from the direction of the load path relative to the failure envelope.

- The approach does not require the effective area principle since moment capacity is defined directly, and solutions exist for both full tension and zero tension assumptions at the foundation-soil interface (Shen et al. 2016, 2017). The effectiveness of the effective area method has been investigated in Feng et al. (2019).

Additional advantages of the failure envelope methods include that:

- No adjustment is made to the classical bearing capacity theory to consider the effect of torsion. However, the effect of torsion can be determined explicitly by the failure envelope approach (Feng et al. 2014).
- The approach provides an indication of the displacement path at failure through the principle of normality (Bransby and Randolph 1998).

Determination of Failure Envelopes

There are three main methods for determining a failure envelope in three-dimensional coplanar VHM load space or under the more general six degree-of-freedom-loading conditions. In their initial development, the most common approach was experimental (Gottardi et al. 1999). Plasticity limit solutions can be developed for the combined loading problems using a delicate kinematically admissible mechanism within the soil body (Bransby and Randolph 1998; Randolph and

Puzrin 2003). Numerical finite element method has become a more common method to determine the failure envelopes because the pursue of the plasticity limit solutions is generally not straightforward under more complex conditions, including the foundation geometry, the soil conditions and the loading regime, e.g. the six degree-of-freedom loading (Feng et al. 2014).

General Procedures for Applying Failure Envelopes to Design

In-house program can be developed to assist routine design using the published algebraic expressions of the failure envelopes. A procedure for applying the failure envelopes to design of subsea shallow foundations was provided in Feng et al. (2014), as summarized in Table 1.

Undrained Uniaxial Capacity: Fine-Grained Soil
Exact solutions exist for the pure undrained ultimate vertical capacity for a surface strip or circular foundation. For the surface strip foundations, the rigorous value of the vertical capacity factor of N_{cv} ($= V_{ult}/As_{u0}$) is 5.14, whereas $N_{cv} = 5.91$ and 6.05 for smooth or rough surface circular foundations, respectively. The vertical capacity factor has been determined for other boundary

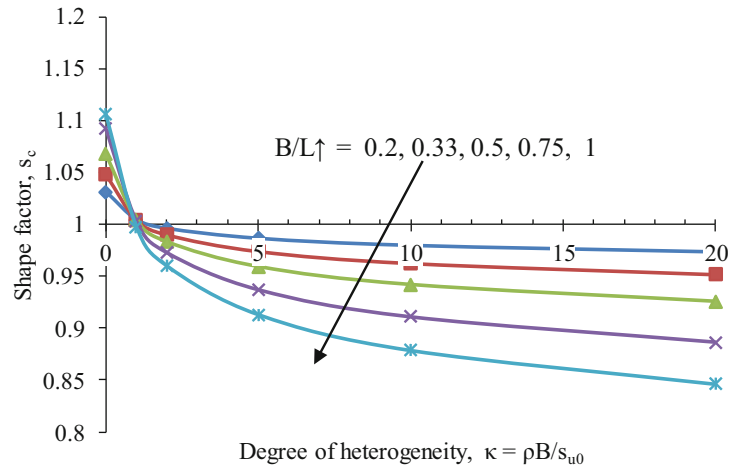
conditions, including the foundation geometry, embedment depth, and the degree of soil strength heterogeneity. The shape and depth factors were derived through finite element limit analyses and finite element analyses (Edwards et al. 2005; Gourvenec et al. 2006; Salgado et al. 2004), where only the uniform soil profile was examined. Analytical solutions of the effect of soil strength heterogeneity were first presented for strip foundations in Davis and Booker (1973). Gourvenec and Randolph (2003) examined the effect of soil strength heterogeneity for strip and circular foundations using finite element analysis, but the embedment was not considered. The coupling effect of the foundation shape and soil strength heterogeneity on the vertical capacity factor has been investigated recently for surface-rectangular foundations by Feng et al. (2017a). Figure 7 shows that as the degree of soil heterogeneity increases the shape factor reduces, becoming less than unity for values of kB/s_{u0} less than about 2. The reduction in the shape effect becomes less significant as the foundation varies from a square to a longer rectangle, since the plain strain condition prevails in the soil beneath a longer foundation.

The undrained ultimate moment capacity of the foundations with zero-tension interface (ZTI) is generally determined by the effective area method. The pure undrained ultimate moment capacity for surface foundations with unlimited tension interface (UTI) can be determined using the upper-bound limit analysis based on the “scoop” mechanism, giving $N_{cm} = 0.69$ and 0.67 for a strip and circular foundation on uniform soil, respectively (Randolph and Puzrin 2003). The effect of soil strength heterogeneity on the moment capacity factor can be described by the quadratic expressions for strip and circular foundations (Gourvenec and Randolph 2003). The coupling effect of foundation shape and soil strength heterogeneity on the moment capacity factor was determined by Feng et al. (2017a).

Pure undrained horizontal and torsional sliding capacity for a surface rough foundation is straightforward. The horizontal capacity factor for embedded strip foundations was derived from the upper-bound limit analyses by Bransby and

Shallow Foundations, Table 1 Design steps for offshore shallow foundations

Step	Details
1	For given foundation geometry, evaluate s_{u0} and nondimensional quantities B/L , d/B , and κ
2	Evaluate uniaxial capacities for vertical, horizontal, moment, and torsional loading
3	Reduce ultimate horizontal, moment, and torsional capacities to maximum values available, according to mobilized (design) vertical capacity, $v = V/V_{ult}$
4	For given angle of resultant horizontal load, evaluate corresponding ultimate horizontal capacity, and similarly for ultimate moment capacity
5	If required, evaluate reduced ultimate horizontal and moment capacities due to normalized torsional loading
6	Evaluate extent to which applied (design) loading falls within H-M failure envelope, and thus safety factors on self-weight V , live loading H , M , T , or material strength s_{u0}

Shallow Foundations,**Fig. 7** Shape factor of vertical capacity as function of aspect ratio and degree of soil heterogeneity

Randolph (1999). An extension of the solution was made for rectangular foundations by Feng et al. (2017b). Torsion is more relevant to the design of subsea shallow foundations. Finnie and Morgan (2004) investigated the torsional capacity of a surface-circular foundation. The torsional capacity factor for surface-rectangular foundations was derived in Murff et al. (2010). The effect of the foundation embedment on the torsional capacity has been systematically investigated for rectangular foundations in Feng et al. (2017b).

Failure Envelopes for Undrained Conditions

Unlimited Tension Interface The pioneer development of the VHM failure envelopes for skirted foundations under undrained soil conditions ignored the physical embedment and often assumed a surface foundation with an unlimited tension interface (UTI) to represent the tensile capacity provided by the foundation skirts. The idealization is appropriated when the soil resistance due to the work done by shearing in the soil above skirt tip level (e.g., due to the disturbance during installation or scour) is negligible. A close-form expression is available to describe the failure envelopes for a circular foundation with UTI on homogenous soil (Taiebat and Carter 2000) (Fig. 8a).

$$f = \left(\frac{V}{V_{ult}} \right)^2 + \left[\left(\frac{M}{M_{ult}} \right) \left(1 - 0.3 \frac{HM}{H_{ult}|M|} \right)^2 \right] + \left(\frac{H}{H_{ult}} \right)^3 - 1 \quad (3)$$

A closed-form expression is also available to describe the failure envelope for a strip foundation with UTI resting on a deposit with a linearly increasing undrained shear strength with $kB/s_{u0} = 6$ (Bransby and Randolph 1998) (Fig. 8b).

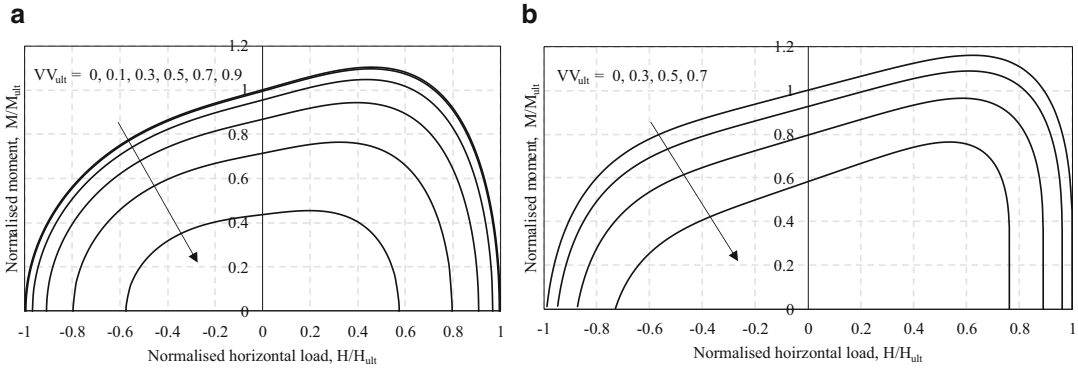
$$f = \left(\frac{V}{V_{ult}} \right)^2 - 1 + \left[\left(\frac{M^*}{M_{ult}} \right)^{2.5} + \left(\frac{H}{H_{ult}} \right)^5 \right]^{\frac{1}{2}} \quad (4)$$

where M^* is modified moment parameter given by the expression

$$M^* = M - h_s H \quad (5)$$

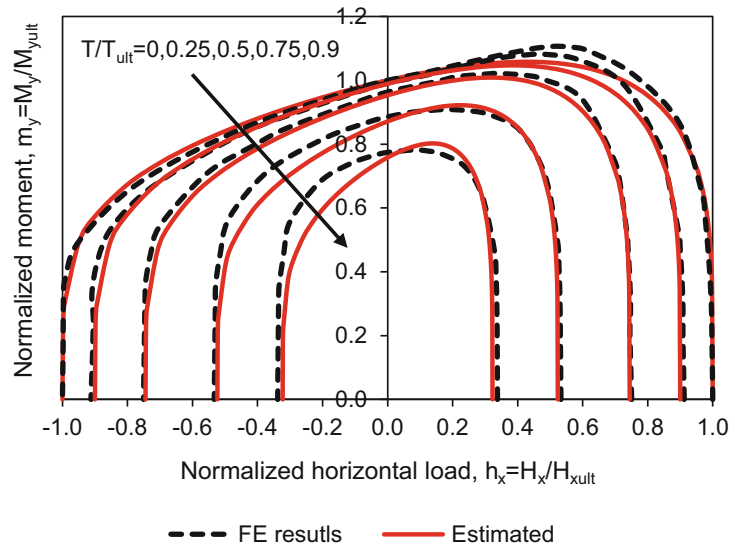
where h_s is the height of the rotation of the scoop mechanism above the foundation level.

The size of the normalized failure envelope reduces with the increasing level of vertical load, as indicated by Eqs. (3) and (4). The normalized



Shallow Foundations, Fig. 8 Example failure envelopes for circular and strip foundations under planar VHM loading

Shallow Foundations, Fig. 9 Example failure envelopes for rectangular foundation under six degree-of-freedom loading



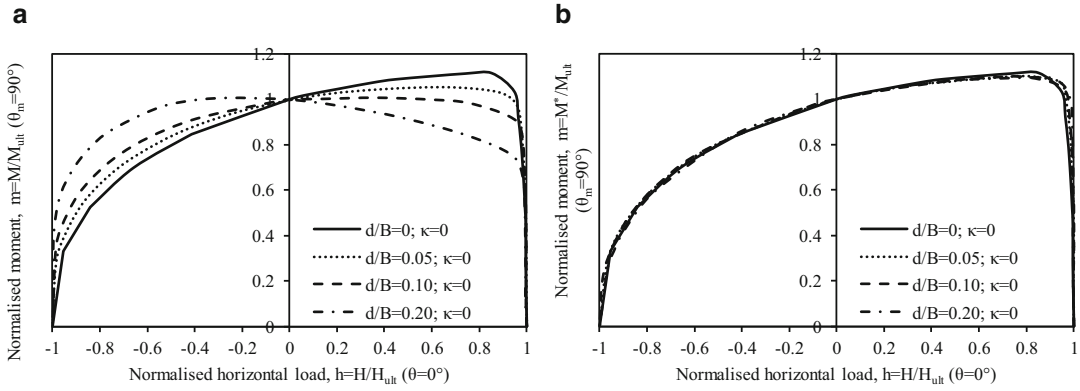
size of the failure envelope also depends on the degree of heterogeneity and generally increases with the decreasing degree of heterogeneity of the most common combination of horizontal load and moment acting in the same direction (Gourvenec and Randolph 2003).

The normalized failure envelope for rectangular foundations was proposed by Feng et al. (2017a) for varying soil strength heterogeneity, and under six degree-of-freedom loading (Fig. 9). A general algebraic expression was given to capture coupling effects of foundation and soil strength heterogeneity on the normalized failure envelopes for rectangular foundations

$$f = \left(\frac{M}{M_{\max}} \frac{M}{|M|} \right)^q \left[1 - \alpha \frac{H}{H_{\max}} \frac{M}{|M|} + \beta \left(\frac{H}{H_{\max}} \right)^2 \right] + \left(\frac{H}{H_{\text{ult}}} \right)^2 - 1 \quad (6)$$

where $H_{\max}$ and $M_{\max}$ are respectively the maximum available horizontal and moment capacity by considering the vertical load and torsion mobilization, and

$$q = 3(1 + 0.1\kappa) \quad (7a)$$



Shallow Foundations, Fig. 10 Effect of failure foundation embedment on the shape of VMH failure envelopes

$$\alpha = \text{Min}(0.9, 1.25 - 0.17\chi) \cdot \text{Max}\{0, 1 + \kappa \cdot [0.11 - 0.08\text{Max}(1, \chi)]\} \quad (7b)$$

$$\beta = 0.03\kappa \cdot \text{Min}(3, 5 - \chi) \quad (7c)$$

where $\kappa = kB/s_{u0}$ $\chi = (B/L)^{1 - 4\theta/\pi}$.

When a pure horizontal load is applied at the base of an embedded foundation, the resulting displacement will involve rotation as well as translation. Likewise, if a pure moment is applied at the base of an embedded foundation, the resulting displacement will involve translation as well as rotation. The coupling of horizontal load and moment leads to an oblique failure envelope, and this effect becomes more evident with increasing embedment ratio. The shape of the normalized VHM failure envelopes for skirted strip and circular foundations was proposed in Vulpe et al. (2014). Feng et al. (2014) presented the failure envelopes for the subsea-skirted rectangular mudmat foundation and found that the effect of foundation embedment on the shape of the normalized failure envelopes can be diminished by translating the load reference point to the foundation level (Fig. 10). For skirted rectangular foundations, it was also revealed that the shape of the normalized failure envelopes can be represented by that of surface foundation if $d/B < 0.3$.

Zero-Tension Interface The effective area method adopted by the offshore geotechnical design guidelines for shallow foundations is

reasonable for predicting the loading-carrying capacity for strip and circular foundations with ZTI under eccentric vertical load, i.e., the V-M loading (Taiebat and Carter 2002). The effective area method is also effective for rectangular foundations under V-M loading, with one-way eccentricity, but significantly underestimated the biaxial moment capacity since this method does not consider the potential of the foundation to rotate about the axis of least moment resistance, and the equivalent rectangles derived from the effective area method are rather different from the contact regions (Feng et al. 2019). For combined VHM loading, the effective area method is conservative by ignoring the coupling effect between the horizontal load and moment.

Failure envelopes for surface foundations with ZTI under combined V-M-H loading have been investigated using plastic limit analysis (Houlsby and Puzrin 1999) and finite element analysis (Gourvenec 2007; Taiebat and Carter 2002) for soil with homogeneous undrained shear strength profile.

The failure envelopes for ZTI were compared with the traditional method for strip and circular foundations under coplanar loading conditions (Shen et al. 2016). The failure envelopes for rectangular foundation with ZTI were investigated in Shen et al. (2017) and given by

$$f = \left(\frac{H}{H_{max}}\right)^2 + \left(\frac{M}{M_{max}}\right)^{2-\nu} - 1 \quad (8)$$

Yield Envelope for Drained Conditions

The term yield envelope rather than failure envelop is commonly adopted for the drained bearing capacity. Yield and failure are synonymous for the undrained limit state of a rigid plastic material while the stress dependency of shear strength for a frictional material leads to gradual yield and hardening with increased displacement. The yield envelope for shallow circular foundations under six degree-of-freedom loading is given by Bienen et al. (2006)

$$f = \left(\frac{h_2}{h_0}\right)^2 + \left(\frac{h_3}{h_0}\right)^2 + \left(\frac{m_2}{m_0}\right)^2 + \left(\frac{m_3}{m_0}\right)^2 - 2a\left(\frac{h_3m_2 - h_2m_3}{h_0m_0}\right)^2 + \left(\frac{q}{q_0}\right)^2 - \beta_{12}v^{2\beta_1}(1-v)^{2\beta_2} \quad (9)$$

where $h_2 = H_2/V_0$, $h_3 = H_3/V_0$, $m_2 = M_2/DV_0$, $m_3 = M_3/DV_0$, $q = T/DV_0$, and

$$\beta_{12} = \left[\frac{(\beta_1 + \beta_2)^{(\beta_1 + \beta_2)}}{\beta_1^{\beta_1} + \beta_2^{\beta_2}} \right]^2 \quad (10)$$

where β_1 and β_2 are shaping parameters for the yield surface in the vertical load plane, varying with foundation types and different soils.

Serviceability

Serviceability assessment requires prediction of foundation settlement over design life. The settlement of an offshore shallow foundation comprises an immediate, undrained component, a component due to primary consolidation, and a component of creep.

Immediate Undrained Deformation

The immediate undrained deformation of a shallow foundation is generally estimated based on the elastic theory. Elastic solutions have been derived for a variety of load conditions, boundary value problems, and layered, heterogeneous, and anisotropic conditions. A comprehensive

selection of elastic solutions is provided in Mayne and Poulos (1999). For a homogeneous elastic soil layer, the settlement w of a circular foundation due to a vertical load V is given by

$$w_c = \frac{q d I_F}{E} \quad (11)$$

where I_F is the rigidity correction factor.

For a rigid foundation, the magnitudes of settlement are equal at the center, corner, and edges, whereas, for perfectly flexible circular foundations, the edge settlements are two-thirds of the magnitude of the center-point settlement. For a rigid rectangular or square foundation, a circular foundation with equivalent area is used to determine the settlement. For a flexible foundation, the corner settlement is approximately one-half of those at the center point.

The reliability of the elastic calculations can be guaranteed by selecting appropriate stiffness parameters. Immediate deformation should be computed using the undrained Young's modulus E_u and Poisson's ratio ν_u .

Primary Consolidation Settlement

An estimate of the vertical time-dependent consolidation settlement of a fine-grained layer under an imposed vertical stress can be determined by the following formula:

$$w = \left(\frac{hC}{1 + e_0} \right) \log_{10} \frac{\sigma'_{V0,z} + \Delta\sigma'_{V,z}}{\sigma'_{V0,z}} \quad (12)$$

where h is the layer thickness, e_0 is the initial void ratio, C is the characteristic value of the compression index of the soil over the loading range considered, $\sigma'_{V0,z}$ is the effective overburden stress at the level of a given soil layer, and $\Delta\sigma'_{V,z}$ is the increment of effective vertical stress in a given soil layer.

The compression index, C_c , should be used in the calculation of consolidation settlements of normally consolidated clays. The swelling index, C_s , should be used in the calculation of consolidation settlements for highly overconsolidated clays where the relevant stress range falls on an unload-reload line. The calculation should be

divided into two parts for stress ranges that span the unload-reload and normal compression lines.

Creep

Deformation due to the reduction in void ratio at constant effective stress condition is referred to as creep. Normally consolidated, particularly highly plastic, clays, organic soils, and calcareous soils are generally most susceptible to creep. Creep effects may be largely ignored in most sands, silts, and heavily overconsolidated clays.

Key Applications

Gravity-Based Structures

Gravity-based structures (GBS) rely mostly on their weight and size of their footprint on the seabed to withstand the environmental lateral and moment loading. The first gravity-based platform, Ekofisk I, was installed in the North Sea at water depth (WD) of 70 m in 1973. Skirts are typically fabricated to the periphery of the foundations when the gravity-based structures are founded on seabeds where softer surficial deposits exist. The provision of deep skirts transmits the foundation load to the deeper, stronger soil. Gullfaks C, built in 1989 at a soft clay site in the North Sea (WD = 220 m), was the first deep-skirted gravity base installed under the assist of active suction. Gravity-based platforms were mainly installed in the Norwegian section of the North Sea with the water depth up to 300 m, and offshore Australia where the water depth does not exceed 80 m.

Concrete Bucket Foundations for Tension-Leg Platforms

The experience from the suction-installed deep-skirted foundations led to applying the technology to individual or clusters of small concrete bucket foundations. The first application of the bucket foundations was for the Sonre A tension-leg platform (TLP), followed by the employment for the Heidrun TLP. The water depth for both installations is above 300 m. The bucket foundation systems were designed such that the dead weight

of the concreted foundation templates (CFT) and ballast counteracts the tension in the tethers under calm weather, working conditions. Only during storm conditions is skirt friction and suction underneath the top cap of the buckets relied on to provide resistance to the moment incurred as the platform offsets over the CFT.

Steel Bucket Foundations for Jacket Platforms

Steel-braced jacket platforms are prevalent worldwide, and steel bucket foundations have been used as an alternative to the traditional pile foundations to support the structures. The Draupner E platform was the first occasion that the steel bucket foundations were provided on a jacket structure. A similar foundation solution was employed on the nearby Sleipner Vest SLT jacket. A unique aspect of this foundation system is the reliance on mobilizing tensile capacity in sands through passive suction under the top cap when the foundations are subject to extreme environmental loads.

With the installation of offshore wind turbines (OWTs) heading into deep water (WD > 60 m), it may prove more economical to use bucket foundations, either as a single foundation or in a tripod or quadruped arrangement. The engineering challenge for suction bucket foundation is to ensure the OWTs can resist the wind- and wave-induced forces without unacceptable deflections, and determine the stiffness foundation system for assessing the fatigue life of the OWTs.

Skirted Mudmats for Subsea Infrastructures

Skirted mudmats have been widely deployed onto the seabed to support subsea infrastructures, including pipeline end terminations and manifolds (PLETs and PLEMs), for oil and gas developments in deep and ultradeep water (WD > 1500 m). The mudmat foundations are generally rectangular in plan, consisting of a top plate fitted with peripheral and internal skirts. Deepwater oil and gas developments continue to push the envelope with regard to the increasing weight and size of subsea mudmats and installation water depth. The Gorgon Project in Western Australia installed 20 subsea structures, some close to 1000Te in weight, and in water depths of up to 1350 m.

Cross-References

► Subsea Production System

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Shear Strength

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Definition

The shear strength is a fundamental property of sea ice. It defines the maximum shear stress that

the ice can bear prior to failure. The shear strength is an important mechanical property since it is involved in the complex stress condition, which is mainly associated with ice-structure interaction.

Scientific Fundamentals

In many cases, ice–structure interaction and ice–ice interaction are subjected to complex stress condition, where shear stress is involved. The knowledge on ice shear strength is required to analysis ice load and ice failure modes. Since ice is a brittle material, there are difficulties to generate pure shear stress without normal stresses.

Testing Methods

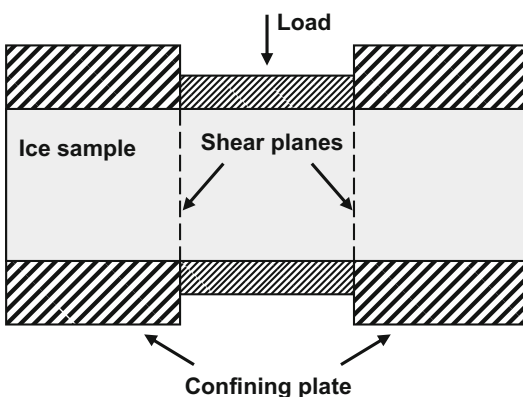
To generate shear stresses, three cylindrical rings are applied as loading apparatus. As shown in Fig. 1, two rings work as confinement while the central part moves downward to provide load (Butkovich 1956; Zhang et al. 1995). The ice core sample, therefore, experiences shear force during the loading. In this setup, two shear planes are generated as marked with dashed lines.

Single shear tests have been performed by Paige and Lee (1967) and Dykins (1971) with loading symmetrically over a semicircular surface of an ice core specimen, as shown in Fig. 2. The failure plane was along the length of the core, since the loading direction was parallel to the long axes of the confinement cylinder. To reduce the confinement effect and expansion, relief holes

are designed on the central of the cylinder. Similar apparatus was applied by Ji et al. (2013) with rectangle confinement and specimen.

The main problem of these tests is that both normal stress and shear stress are generated on the shear plane. Undoubtedly, the normal stress influences the determination of the shear stress and shear strength. Especially for the setup without the relief holes, as we shall introduce in the later section, the shear strength from Ji et al. (2013) is higher than the measurement from others. When the ice sample is under shearing, the deformation leads to volume dilation. The volume change is confined by the confining plate and it results in strong normal stresses on the sample as well as the shear plane.

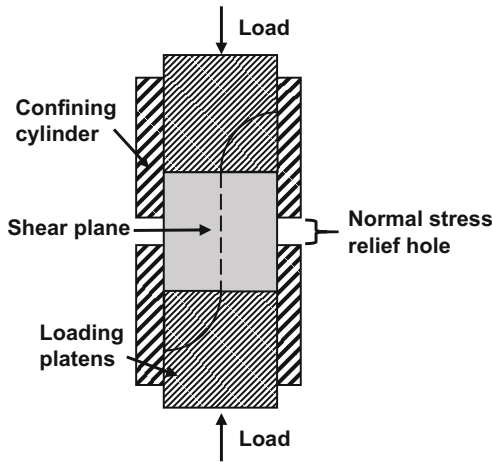
To avoid the influences of normal stresses, an asymmetric four-point bending rig was designed by Frederking and Timco (1984). In this setup, load is applied at four different points on the ice sample with beam shape, as shown in Fig. 3. The shift of top and bottom loading position results in a high shear stress condition in the central of the beam. And the shear failure is expected to appear in this surface. Although the loading arrangement avoids the influences of normal stresses, it has to deal with the bending stresses. Depending on the slenderness (ratio between height and length), the sample may fail in bending instead of shearing. From this point of view, the specimen needs to be sufficient thick. If so, the specimen only fails after reaching significant load and the local crashing at loading point would be another issue (Bailey et al. 2012). Besides, for the combination of bending and shearing, the central of the beam is no longer under pure shear stress neither, compared with previous setups.



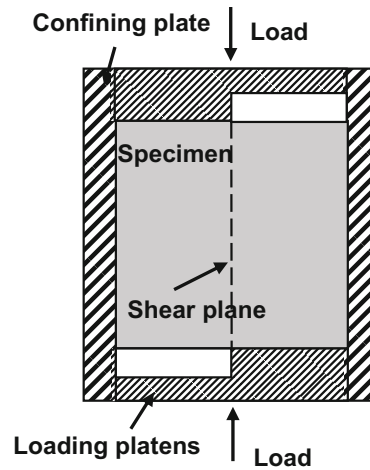
Shear Strength, Fig. 1 Double shear apparatus redrawn from Butkovich (1956) and Zhang et al. (1995)

Main Database of Shear Strength

The shear strength of sea ice has not been measured as much as other mechanical properties such as compressive strength, bending strength, tensile strength. The relative few database and the parameter matrix are given in Table 1. The overall strength is between 0.1 and 2 MPa. Although some parameters are missing, most of them are collected from these references. It shows that the variation is significant depending on the testing condition. It is reasonable that the experiments are

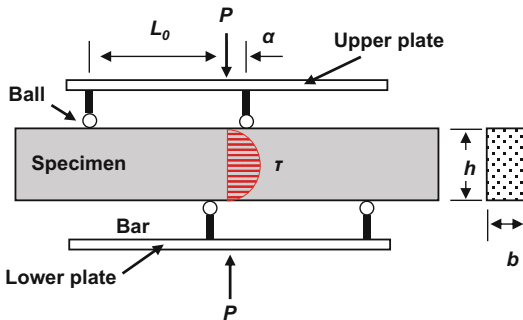


(a) Apparatus redrawn from Paige and Lee (1967) and Dykins (1971)



(b) Apparatus redrawn from Ji et al. (2011)

Shear Strength, Fig. 2 Scheme of single shear test. (a) Apparatus redrawn from Paige and Lee (1967) and Dykins (1971). (b) Apparatus redrawn from Ji et al. (2013)



Shear Strength, Fig. 3 Asymmetric four-point loading apparatus redrawn from Frederking and Timco (1984), Frederking et al. (1986), and Bailey et al. (2012)

performed in different location and research group. Thus, efforts are made in order to that the results can be comparable. Aside of the parameters given in the table, the experiment setup may also be worthy to be considered in the evaluation of the shear strength. As we mentioned in the previous section, the sample is in a complicated stress condition, in most experiments, instead of pure shearing. The normal stress from the confinement may influence the final results.

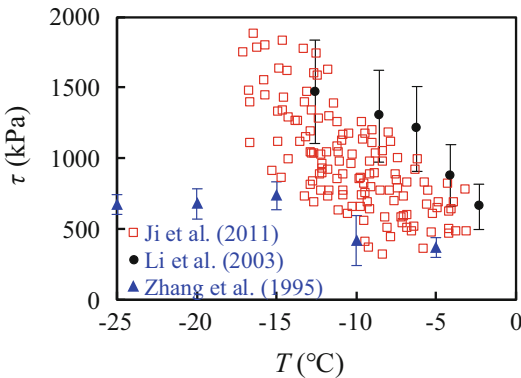
Natural sea ice is consisting of brine, pure ice, air bubble, and solid salt. Since the ice temperature

is close to its melting point, the ice mechanical properties are sensitive to temperature. Besides, when temperature rises up, the brine is diluted by melting pure ice so that the chemical balance is kept. Therefore, the temperature is an important factor of the shear strength. The relation between the shear strength of sea ice and temperature is given in Fig. 4 based on different resources. In general, the shear strength is decreased with increasing the ice temperature. It shows that the shear strength from Ji and Li (Ji et al. 2013; Li et al. 2002) is higher than that from Zhang et al. (1995). It should be due to the former tests that were performed with lateral confinement.

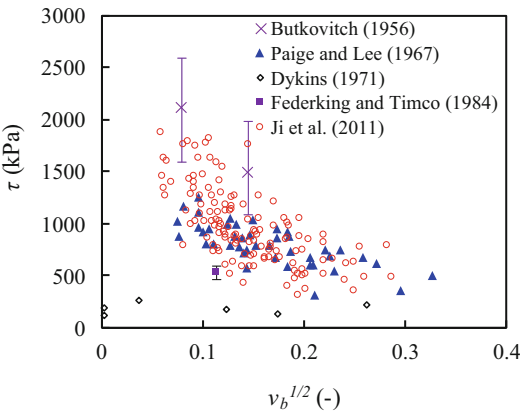
For sea ice, only the solid part carries the shear forces so that brine volume becomes a key parameter to evaluate the shear strength. In Fig. 5, it gives the relation between brine volume and shear strength based on different literatures. Overall, there is significant scatter in the measurements of the shear strength. The physical property of ice itself is relative random and the grain structure varies depending on the weather condition during ice growth. The loading condition as well can influence the measured shear strength due to the unknown normal stresses.

Shear Strength, Table 1 Matrix of key experimental parameters

References	Ice structure	Loading direction	$T(^{\circ}\text{C})$	$S(\text{ppt})$	$\tau_f(\text{kPa})$	Sample dimension (mm)	Loading condition
Butkovich (1956)	Columnar	Cross to columnar	-6 ± 2 -11.5 ± 1.5	6	1600 2300	76 mm-diameter core	–
Paige and Lee (1967)	Columnar	Parallel to columnar	–	–	500–1000	76 mm-diameter core	–
Dykins (1971)	Columnar	Parallel to columnar	–	–	100–250	76 mm-diameter core	–
Frederking and Timco (1984)	Granular	Both	13 ± 2	4.2	550 ± 120	400*100*50	Constant stress rate
Zhang et al. (1995)	–	–	-5 -10 -15 -20 -25	4	369 ± 69 417 ± 175 735 ± 97 677 ± 108 670 ± 71	70*70*300	Constant strain rate
Li et al. (2002)	Columnar	Parallel to columnar	-2.3 -4.1 -6.2 -8.6 -12.6	4.7 ± 0.87 5 ± 0.98 4.6 ± 1.35 4.3 ± 0.4 4.63 ± 0.93	608 ± 301 756 ± 306 716 ± 395 1080 ± 447 922 ± 309	105*80*60	Constant strain rate
Ji et al. (2013)	Columnar	Parallel to columnar	-3 to -17	1–7	350–1860	70*70*50	Constant stress rate



Shear Strength, Fig. 4 The shear strength as a function of the temperature



Shear Strength, Fig. 5 The shear strength as a function of the square root of brine volume

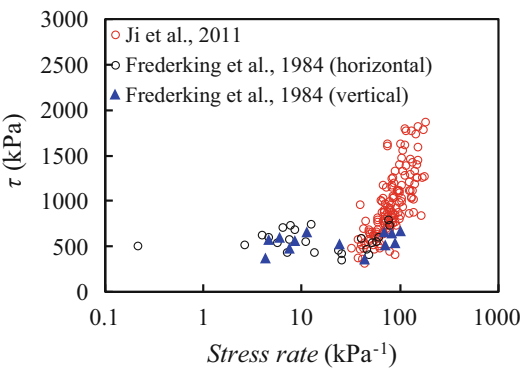
Generally, the shear strength linearly decreases with decreasing the square root of the brine volume. It is reasonable since the higher brine volume the less solid ice to damage. The results from Butkovich (1956) can reach

2.6 MPa and are higher than the others. Although, in his tests, there was almost not confinement at all based on the experiment setup, the loading direction is crossing to the

grain columns. It might be the reason behind the high shear strength since breaking the grains causes significant loads.

Based on previous results, the brine volume, undoubtedly, plays a major role in the shear strength of the sea ice. It is reasonable since only the solid part resists the shear forces. However, as the discussion above, it is very difficult to generate pure shear forces on the shear plane. Normally, the shear plane is accompanied with normal stress depending on the setup design. The nominal elastic modulus is related to loading rate and stress rate. Therefore, the shear strength is also influenced by the stress rate.

In Fig. 6, it gives the relation between the shear strength and stress rate. It shows that the stress rate has very little influence on the shear strength of the tests by Frederking et al. (1986). In their experiment design, the shear forces were generated by a smart design of four-point bending. In this way, the compressive forces have almost no influences on the loading procedure. The shearing may be generated together with bending, but the bending is not sensitive to the stress rate neither. Thus, it seems that the shear strength from Frederking et al. (1986) is independent on the stress rate. In the results from Ji et al. (2011), on the other hand, the shear strength is strongly influenced by the stress rate. If we move back to the experiment setup (Fig. 2b), there was strong lateral confine during the shearing tests. Thus, the influences from compression are taken into account in the shear strength.



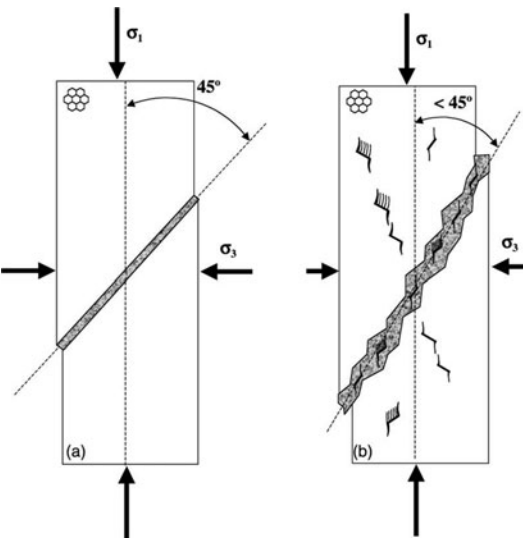
Shear Strength, Fig. 6 The shear strength as a function of the stress rate

Key Applications

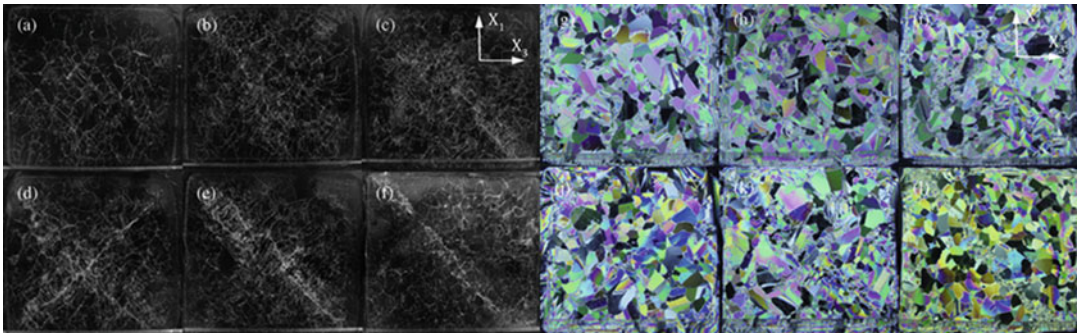
The shear strength of sea ice draws attention since its potential application in both scientific and engineering circumstances. For the former one, the shear strength is helpful to understand the failure and deformation mechanisms of ice under complex stress states while for the latter one, the shear strength can be an input parameter for structure design.

Compressive Shear Faults

When sea ice moves against with vertical structure, the compression process may dominate the ice–structure interaction. Even under compression, the ice usually fails with an inclined shear crack $45 \pm 5^\circ$ to compressive stress direction, depending on the stress condition (Schulson 2002). In some scenarios, the ice is usually under confinements, especially for the thick ice. For the ice under biaxial and triaxial compression, it may fail with a narrow shear band orientated on a plane of maximum shear. As shown in Fig. 7, it gives the sketches of different shear faults. In the Fig. 7a, the terminal failure is assumed to be caused by plastic flow and the compressive strength is represented by the difference between



Shear Strength, Fig. 7 Schematic sketches: (a) plastic shear fault with a shear crack; (b) Coulombic shear fault with a shear band (Schulson 2002)



Shear Strength, Fig. 8 Structural evolution of P-faulting in granular ice (Golding et al. 2012)

the greatest and the least principle stress. In the Fig. 7b, the failure can be explained in terms of comb-crack mechanism and Coulombic frictional criterion. The shear faults are in terms of P-fault and C-fault.

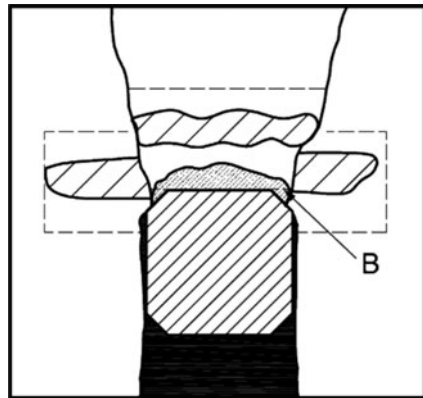
Based on the main purpose of this section, we focus only on the P-fault, which is dependent on the shear strength. During the P-fault, it can be observed that a shear band appeared on the ice sample as shown in Fig. 8. Golding and others suspect that the process involves local heating and recrystallization (Golding et al. 2012). Since the grain size in the shear band is smaller than the rest, it is reasonable to assume that part of the ice is melted and refreeze in a very short period.

Shear Ridge Formation

Ice ridges can be categorized by the formation process shearing and compression. The shear strength may not be directly applied on ocean engineering, but it is helpful for understanding the formation of ice ridges and makes contribution to the force balance model (Lipscomb et al. 2007). One potential application is the ice forecasting model. In this model, the shear strength takes part in the local stress calculation as a parameter. Although the model usually focuses on a geophysical scale, it could provide environmental parameters for ship navigation and ocean structure operations.

ISO19906 Standard

Shear failure can be a relevant limiting mechanism for which an example is shown in Fig. 9.



Shear Strength, Fig. 9 Ridge shear failure (dashed line shows the displacement of failed ridge) (ISO19906 2010)

When a contact appears between an ice ridge and a structure, the shear failure is one of the potential failure processes. Ice is seldom in a state of pure shear while acting on a structure, as shown in Fig. 9 (ISO19906 2010). Therefore, the shear strength is not usually considered as an independent material parameter. In the design of Arctic offshore structures, the shear strength may be used as a parameter to interpreting full-scale data for the calculation of ice actions.

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Sheath

- [Umbilical Cable](#)

Sheerleg (SL)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Ship

- [Ship-Iceberg Interactions](#)

Ship Construction and Operation Impact on Energy Efficiency

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Synonyms

[Alternative fuel](#); [Energy efficiency](#); [Renewable energy propulsions](#); [Ship hull design](#); [Ship machinery maintenance](#); [Ship optimization](#)

Definition

MEPC adopted a series of important guidelines at its 63rd session, held from February 2 to March 2, to promote the improvement of energy efficiency of international shipping vessels. The pivotal content consists of two aspects: focusing on the adoption of energy-saving technologies from the construction stage of ships based on technical measures, of which the concrete index is EEDI, and focusing on improving fuel economy and energy efficiency through equipment management and navigation optimization in the operation stage, of which the concrete index is EEOI. In this section, the specific calculation methods of EEDI and EEOI are not discussed in detail, and relevant measures are expounded only around the goal of lowering these two indicators.

Measures to Improve Energy Efficiency at the Design Stage

The application of one or more energy-saving measures in the design and construction of ships can effectively improve the energy-saving level of ships and reduce pollutant emission. A full workflow of various energy transmission processes in ship as a whole system is thoroughly studied by Tillig et al. (2017). In the following, some aspects to increase a ship's energy efficiency from design perspectives are firstly described.

Ship Optimization Design

The key of energy-saving technology of ship is the optimum design of energy-saving ship. It satisfies the ship exploitation conditions, optimizing hull form design and ship form to minimize the ship's resistance and selecting the main engine with low oil consumption to make the overall coordination match, to achieve the optimal configuration of the ship's engines, propellers, and rudders, and thus to improve the ship's propulsion efficiency and reduce the operation cost. Through the hull form optimization, the ship has better fluid lines to reduce its resistance, thus may achieve the higher speed at the same fuel consumption or reduce 8%~15% the ship energy consumption at the same speed.

Bulb Stern Ships

The ship type is characterized by a bulb having a length of 1%LPP~2%LPP and a very narrow width at the stern of the ship at the full load draft. According to the principle of fluid mechanics, the bulb stern is used to generate an equal amplitude wave opposite to the stern wave to reduce the wave height. There are two main functions: one is to absorb wave to make the wave-originated point of the stern wave back. As a result, the diffusion surface of the stern wave is greatly reduced, which reduces energy loss to reduce the ship resistance. And another is to improve the wake flow, as a rectifier, to improve ship propulsion efficiency. At the same time, it can also reduce the excitation force of the propeller, which is conducive to vibration reduction. The main parameters of the bulb stern design are its length, size and shape of its cross section and spherical center, etc. Generally, it is advisable to be located on the center line of the propeller shaft, and its size and shape can be selected according to the ship model test. This ship shape saves main engine power about 7% to 9%.

Bulbous Bow Ships

In $F_n = 0.25 \sim 0.35$ for the designed ship, the adoption of bulbous bow can reduce wave resistance, wave breaking resistance, and bilge-eddy resistance. For this, a bulbous bow is provided on large ships and also on some small ships. Bulbous

bows are shaped in water drops, melon seeds, ellipses, and small noses. The main parameters of bulbous bows, such as length, width, sizes and shapes of cross section, and bulb immersion depth, can be determined according to designed ship types, Froude numbers, drafts etc. Generally, the shape and size of the bulb are optimized by ship model experiment. The wave caused by the bulbous bow and the bow wave can counteract with each other, so as to improve the flow at the forward entrance to achieve the purpose of reducing wave resistance. A bulbous bow can also increase the ship buoyancy, acting as a ballast tank to adjust the ship's trim. Bulbous bows can reduce the total resistance by 8% to 11% when used on large cargo ships and oil tankers and about 6% when used on medium-speed ships and high-speed ships.

Shallow Draft Large Full Ships

Shallow draft large full ships are a kind of ship restricted by the channel and port berth drafts. Their main features: firstly, the deadweight is increased by 20%~30% compared with the conventional ships, the higher the design tonnage, the better the load effect. In general, the deadweight ratio of carriers of 2500 gross tonnage and upward is above 0.67. Secondly, they have high economic benefit, because the ship type has the characteristics of large cargo capacity and shallow draft and can serve as ore carriers, coal carriers, oil tankers, chemical liquid carriers, and replenishing ships for fleet in sea. This ship type also has the advantages of low cost, less investment cost, more operating routes, and more navigable routes, which are widely used at home and abroad.

Construction Optimization and Application of High-Strength Steel

It is mainly to improve the local external characteristics to make the ship structure lightweight. The methods are as follows: selection of appropriate frameworks and frame spacings; application of corrugated bulkheads; selection of appropriate connections between members; determination of the total longitudinal strength of the ship by optimization design; reasonable increasing of the proportion from 30% to 55% of high-strength steel applied to longitudinal members,

decks and longitudinals, top wing tank sloping plating and longitudinals, shells and inner bottom plating and longitudinals, side shell plating, inner hulls, and bottom stringers transverses; reasonable structure arrangement to improve the utilization rate of materials, such as joint type design and reasonable frame spacing; and reasonable structure design, such as the optimized configuration of transverse corrugated bulkheads. On the target ship, the rectangle cargo hold transverse bulkheads are replaced by trapezoidal bulkheads, which can reduce the overall weight of each transverse bulkhead, reducing the number of bottom stiffeners for anti-buckling reinforcement.

Main Engine Type Optimization

Main engines are selected after increasing the diameter of propeller, mainly to downshift the engine to realize the purpose of keeping ship speed unchanged and reducing the fuel consumption rate of main engine, thus to reduce EEDI value of ship and improve energy efficiency (Bailou 2013).

Optimal Configuration of Ship, Engine, Propeller, and Rudder

The optimal configuration of ship, engine, propeller, and rudder is a very complicated technical subject, which involves many technical problems such as ship type, ship's main parameters, power units, selection of propeller, rudder design, etc. To improve the ship's speed, effective measures can be taken to reduce ship resistance, improve propulsive efficiency, and increase engine power. The measures to improve the efficiency of propulsion are as follows.

Design of Large Slow-Turning Propellers

The propeller diameter is usually determined according to the shape of the stern and ship's draft. Increasing the propeller diameter as well as decreasing the rotating speed can improve the open propeller efficiency and increase the thrust deduction factor and wake fraction to improve the hull efficiency. Therefore, to design a large slow-

turning propeller, matching the optimum stern shape is one of the important measures to save energy. On high-speed ships, installation of a large slow-turning propeller will obtain the best propulsion efficiency.

Application of Inboard-Rotating Screws

Domestic studies on an inboard-rotating screw show that the hull efficiency using inboard-rotating screws can be increased about 20% above that using an outward turning propeller. For propellers of MAU type and B type, the hull efficiency can be increased about 2.4%, and the rotation efficiency is improved by 3% to 5%. The mechanism is mainly to produce a kind of reverse oar effect, because the inward turning propeller is opposite to the tangential speed of the incoming flow into the oar disc, then the rotating speed of the propeller is equal to the increase of the rotating speed of the flow, and the increase of the thrust of the child propeller, thus improving the ship propulsion efficiency. If the optimal stern shape is designed skillfully, the wake field can be improved, the wake fraction can be increased, the thrust deduction fraction can be reduced, and the propeller can be operated in a stable and high wake region, then the ship propulsion efficiency can be improved significantly.

Application of High-Performance Airscrews (PBCF Devices)

A PBCF device (also known as the Propeller Boss Cap Fins) has been developed abroad, which has fins on the propeller shaft sleeve cover having the same number as that of the propeller blades and having different shapes depending on the shape of the stern. The main function of PBCF devices is to reduce the vortex generated by propeller rotation to reduce the eddy resistance. This device has the advantages of simple structure, lightweight, and low cost. It is suitable for all kinds of ships, especially for those with a larger propeller pitch.

Application of Boost Wheel Devices

A ship boost wheel device is a device recycling the propeller races to convert some wake energy into additional thrust for the ship propulsion. The

device adopts a roller bearing to support the end of the propeller shaft, which can freely rotate on the propeller shaft to generate additional thrust under the action of propeller wake, thus improving the ship propulsion efficiency. The device is simple in structure and easy to install, disassemble, and maintain.

Positioning Optimization of Propeller Tip Relative to the Hull

The gap between propeller tip and hull surface can be optimized to reduce eddy current effect. For vessels sailing in deep water, it is generally advisable to design propeller tip beyond the hull baseline and make propeller shaft center line and hull horizontal line form proper stern shaft inclination angle (1.5° to 2.3°), which can improve the incoming flow in front of propeller disc and provide sufficient water supply to improve the efficiency.

Application of Rudders with Additional Thrust Fins

A rudder with additional thrust fins is a boost installation for ship energy saving, of which the pushing fins are set at the appropriate positions on both sides of the rudder blade to adjust the angle of incidence of the rudder sides in order to be suitable for the propeller wake flow. In this way, on the one hand, the rotational energy in the propeller wake can be recycled, and on the other hand, part of the lift generated by the thrust fins can be converted into additional thrust. The main parameters for additional thrust fins include length, cross-sectional types, locations, and mounting angles of fins. Thrust fins are installed on some large full-formed single-screw ships abroad, which can save main engine power about 4% to 5%. The thrust fins studied in China have also been installed on more than 20 inland waterway boats, and the energy-saving effect is considerable.

Application of Auto-steerings

After selecting the ship type and power unit as well as the appropriate propellers, auto-steerings may be applied to get the best result.

Optimizing Arrangement of Equipment and Systems

It is mainly to optimize the layout and setting of ballast water operation, piping, and valves from the perspective of economic and convenient use, such as less use of pipelines and valves, and discharge of ballast water by combining pump and gravity.

Speed Design

Traditional ship design is usually optimized for design draft service speed or structure draft maximum speed. However, in the actual operation of the ship, the actual operation state and design point of the ship are generally quite different according to the different types of the ship, different types of goods, different environmental meteorological conditions of the route, and different operation strategies of the shipping company. For example, with most oil tankers, the full load condition will not exceed 50%, and the main engine power is below 50% MCR on average when operating. For container ships, goods are lighter in many cases, and the average operating condition is about 60% to 70% of the full load. According to the operation strategy of the shipping company, the main engine power is only 50% of MCR (MCR is the maximum continuous power of the main engine). Hydrological and meteorological conditions, channel water depth and other restrictions on the vessel equipment and hull surface state, and fuel quality have different degrees of impact on the operating conditions.

The vessel operating condition combination (draft/load, main engine power/speed) and its occurrence frequency are called operation profile. Each shipping company has its own operation profile. Airspeed design according to operation profiles is a comprehensive optimization method with multiple operating conditions, which gives greater weight to the operating conditions with high-operating frequency, so that the energy efficiency of ship operation can be optimized. This airspeed design method is widely studied by the industry and will be applied more and more.

Measures to Improve Energy Efficiency at the Operation Stage

According to the calculation formula of EEOI and relevant literature, the main factor affecting the size of EEOI is the fuel consumption rate of the berthing time during the voyage speed loading rate (Faber et al. 2009).

Speed is positively correlated with EEOI and is most sensitive to its changes. Reducing speed is therefore the most effective way to increase energy efficiency. IMO's priorities include reducing speed. Because of the high oil price at present and the global trade is relatively low, the liner companies are highly competitive, and each ship company has reduced the speed to save fuel consumption, which is objectively beneficial to emission reduction. In addition, the speed increases or decreases in the same proportion, EEOI changes in different proportions, and the influence of growth rate on EEOI is greater. Although reducing speed is most effective for improving energy efficiency, it will also lead to a decrease in transportation turnover per unit of time. In order to ensure the normal transportation of goods, it may increase the shipping capacity, resulting in an increase in total CO₂ emissions. Therefore, the ship's speed should be optimized.

The sensitivity of EEOI to full capacity is second only to speed, and the two are negatively correlated. Therefore, increasing the load factor will play a greater role in reducing EEOI. It is also in the economic interest of the shipping company. The full load rate increases or decreases in the same proportion, while EEOI changes in different proportions.

EEOI is negatively correlated with voyage. EEOI has a low sensitivity to changes in voyage relative to speed and load factor. That is to say, if other factors remain unchanged, the increase of single voyage helps to reduce EEOI, but the change will be more limited. It is worth mentioning that in the calculation of the deadweight, six items, such as personnel and luggage (WPB), total reserves (WR), fuel oil (WF), lubricating oil (WL), furnace water (WBW), spares and supplies (WRS), account for a small proportion of the total displacement. The empirical formulas for the total

amount of fuel oil and lubricating oil furnace water in a ship are all related to the voyage. Therefore, if these small quantities are not neglected, the sensitivity of the voyage factor may vary slightly.

Berthing time is positively correlated with EEOI. This is because ships do not increase turnover when berthed, but CO₂ emissions remain. Therefore, the longer the berthing time, the higher the EEOI and the lower the energy efficiency. The berthing time is mainly the loading and unloading and waiting time at the port, improving the efficiency of loading and unloading at the port can accelerate the turnover, which is helpful for reducing the emission and improving the efficiency. Relative to speed and load factor, EEOI is not very sensitive to berthing time. However, in actual operation, the ship will be attached to several ports, and the use of the ship crane will increase the fuel consumption of auxiliary engines.

The ship's emissions come from the fuel consumption of the auxiliary engine boiler, including maximum continuous power of host (PMCR), engine fuel consumption relation (FM), auxiliary engine average fuel consumption (FA), and boiler average fuel consumption (FB). Therefore, in the future, if manufacturers of marine auxiliary boilers can design relevant marine machinery and equipment with lower fuel consumption and higher efficiency, it will be very beneficial to the improvement of ship energy efficiency. In the daily operation, the shipping company pays attention to protect the machinery and equipment, so that they can achieve the best operating environment, and also save fuel consumption and improve energy efficiency.

In conclusion, measures to improve energy efficiency in the operation process should closely revolve around the above key factors.

From market perspectives, different ways related to increase a ship's energy efficiency and reduce air emissions to comply with IMO regulations have been surveyed by some major classification societies, such as in ABS (2015) and DNVGL (2014). In the following, a few typical measures are presented to show how they can help to reduce the IMO EEOI during a ship's operation/service period.

Routing Planning

For example, multi-port direct transport scheme, the combined operation of trunk and branch lines. The energy consumption and energy efficiency level of the ship fleet should be compared and analyzed. The energy efficiency target evaluation system should be established as the basic basis for evaluating the strengths and weaknesses of each decision-making scheme, and the optimal scheme should be selected.

The main factors, such as supply factors and port factors, which are related to ship energy efficiency and ship utilization efficiency, should be considered. For example, the distance between the basic port and the route and the distance between the voyage and the ship's energy efficiency, the basic port is selected in the port with a relatively concentrated source of goods, which can reduce the turnover of goods and reduce energy consumption. Natural conditions of the port and handling efficiency are the operational efficiency factors that should be considered by owners or ship operators during a ship's voyage planning. Various optimization methods used in the routing planning markets can be referred to Wang et al. (2017).

Many factors, including seasonal factors, should be taken into account in the formulation of schedule and schedule, and economic speed should be adopted from the perspective of energy efficiency and economy of ships. In the operation process of liner ships, the flow of goods between ports is changing at any time. The original confirmed order of docking port and docking should also be adjusted in real time according to the situation, so as to make the ship full load and improve the load rate. According to the loading capacity of the ship, the time schedule or the frequency of ship dispatch should be adjusted in time. The spare time of the ship arriving at the port should be designed according to the specific conditions of a port and carefully planned and arranged by the company so that the time schedule drawn up can be flexible and adapt to the changes of external conditions. The ship should slow down and use the economic speed according to the shipping date, so as to avoid the fuel consumption caused by the

accelerated voyage caused by the delay and shift caused by the overtight design. The reduction of freight volume due to deceleration shall be coordinated by increasing vessels, and the corresponding measures shall be communicated and coordinated with the port side and cargo owner.

Timely Communication

Good early communication with the next port to maximize availability of traffic or berths is conducive to reducing ship lock or berthing waiting time, thus using the best speed in advance and reducing fuel consumption.

Reasonable Scheduling

We should strengthen the management of shipping routes, optimize the resources of shipping capacity and goods supply, reasonably arrange the ships, optimize the ship types, and reduce the operations at intermediate ports. According to the voyage mission plan, the ship should take the factors such as the check of tidal weather channels into full consideration and choose the best time to ship, so as to reduce the frequency of berthing.

Speed Optimization

The main engine power is directly proportional to the three times of the speed, so the ship should choose a reasonable speed according to the importance and urgency of the task. If time permits, the engine power and fuel consumption will be reduced to the third power, which can significantly reduce fuel consumption. Considering the actual operation condition of the marine diesel engine, when the power and speed change, the fuel consumption rate is affected by the quality speed of the air change of the fuel injection quantity and is not a constant value. Generally, the fuel consumption rate is the minimum when the load is 75% to 90%. Therefore, the principle of ship speed optimization is to select the right evaluation approach, taking into account of the navigation route and operation requirements, to analyze and determine the reasonable economic speed for various ship types and ages and find the best

equilibrium point between fuel consumption and speed according to the technical data, such as ship's speed and fuel consumption of main and auxiliary engines.

Application of Economic Routes

Under the premise of ensuring no violation of navigation rules and safety, the actual voyage can be shortened to the maximum extent, but the shortest route may not be the optimal one; especially when sailing across the ocean, the optimal route should be the one that takes into account various factors such as wind flow, weather, and sea conditions. Make full use of the equipment such as the depth finder, and make the best use of ocean currents and tidal currents while avoiding adverse weather effects and strong counter current when external conditions such as climate regulations of the navigation channel allow. Although the use of tidal navigation is an effective method to save energy in operation, this method requires the company to consider comprehensively from the efficiency production arrangement of ships in the freight cycle.

Ship and Equipment Management and Maintenance

Underwater Hull Maintenance

The growth of marine attachments to the hull increases ship resistance. Regular hull cleaning reduces drag and reduces overall fuel consumption. The energy efficiency measures for the hull include hull coating, reducing marine biological adhesion and ship resistance; when the ship has fouling bottom and seriously affects the speed of the ship, it should report to the company in time, and the company should arrange to clean up the ship. Regular maintenance of the hull, regular docking and evaluation of the ship's performance and application of coating to ensure the hull smooth. Check and clean the hull regularly to ensure the cleanliness of the hull and to timely complete removal and replacement of underwater paint to avoid repeated point sandblasting and multiple dock repairs resulting in increased hull roughness.

Host Performance Monitoring and Optimization

The operation and maintenance of main engine may be optimized by monitoring and analyzing its performance data including fuel consumption, operating loads and cylinder wear, e.g. properly adjusting oil supply quantity and oil supply advance angle, checking the cylinder and replacing the piston ring, cylinder liner and gasket by measuring the cylinder explosion pressure and compression pressure and referring to other operating parameters. Oxygen monitoring: The combustion performance of diesel engine can be improved by monitoring and optimizing oxygen content in intake air.

For modification of diesel injection systems: Some disadvantages of traditional fuel injectors are gradually revealed with a great number of ships slowing down. For example, slow running results in poor fuel atomization and even leakage, which causes a certain amount of fuel to enter the cylinder combustion chamber at lower temperatures where it is not fully burned, resulting in oil and carbon fouling and smoke, increased emissions of hydrocarbons and particulate matter, even shorting the life of main engine. The fuel saving and greenhouse gas emission can be reduced by using the spool valve injector or high-pressure common rail electronic injection.

Turbosupercharger matching: Match the supercharger reasonably according to the working condition and load of the diesel engine.

Application of fuel additives: Adding fuel additives can improve fuel quality, improve combustion performance, reduce harmful material emissions, and reduce carbon deposition in cylinder. In addition, there is no need to add device or change engine structure to use fuel additive, so using fuel additive is regarded as a convenient and effective energy-saving and emission reduction measure.

Propeller Maintenance or Update

The surface finish of propeller blade should be maintained, and the damage of edge defect, crack, and wear to the thickness outside the specification should be handled in time. The choice of propeller is usually determined at the design and construction stage of the ship, but the redesign and

selection of propeller should be considered when the conditions of the ship navigation area change. When the propeller is redesigned and selected, the whole of propeller should be considered.

Optimization of Fuel Management and Use

Fuel Oil Addition and Recovery

As far as possible, reasonable fuel loading frequency should be adopted, measures should be taken to prevent leakage from running, and quality and quantity monitoring of the added fuel should be strengthened. Oil, such as diesel oil for cleaning parts, bilge recovery and smear oil in cargo tank, should be effectively recovered, cleaned and utilized reasonably.

Oil Quality Maintaining

To prevent the heavy oil mixed with light oil and reduce the grade of light oil

To avoid or reduce mixing of different batches of oil to prevent chemical reactions from forming large amounts of oil sludge

To shorten storage time of mixed oil to prevent delamination

To closely track the change of fuel quality in use and adjust the heat separation amount according to the oil property

Application of Heavy Oil Technology

On the premise of meeting the requirements of regulations and relevant technical documents concerning the sulfur content of fuel oil for ships, the use of heavy oil technology for ships can be considered to achieve the purpose of energy saving in the following aspects:

- Through the oil supply unit and its heating, heat, insulation, separation, filtration, and hydrophobic system to improve the flow of low-quality fuel and improve the oil quality to achieve the goal of full combustion.
- In the process of using heavy oil, the constant vehicle speed should be controlled reasonably, because the prominent problem of burning heavy oil ships is that the constant vehicle

speed is too low, causing the host engine to operate under low load for a long time and resulting in the combustion incomplete; carbon deposition situation is relatively serious, such as cylinder sleeve wear faster, supercharger pollution, and high-unit fuel consumption rate.

- To manage and make good use of the main facilities and equipment such as the oil distributing unit boiler, clean the boiler regularly, and clean the oil tank regularly.
- The period of oil for the monitoring and testing, the establishment of the ship lube oil use files, to gradually find the rule, to achieve reasonable oil change.

Energy-Saving Awareness, Energy-Saving-Related Training, and Incentive Mechanism

Energy-Saving Consciousness

The cultivation and continuous improvement of energy conservation awareness is an important guarantee for effective implementation of relevant energy conservation measures, as well as an important content of human resource training. Through the cultivation of consciousness of energy conservation, we should be able to achieve the goal of improving the initiative of all members. The cultivation of energy conservation awareness is a long-term process, which cannot be accomplished overnight. First of all, it can start from the small steps that are easy to realize, such as turning off the lights, turning off the air conditioning, turning off the hot water kitchen stove, etc., to form the institutionalized requirements, so as to gradually improve the energy conservation awareness of the whole staff and gradually infiltrate this awareness into the daily work.

Shipshore Energy Efficiency Training

As ship manager, every crew member plays a very important role in higher efficiency and energy saving. Everyone should be familiar with the operation of the ship's equipment and clear about the potential of certain specific equipment in waste or energy saving, so as to manage and operate the equipment well. At the same time, the

crew should also develop a good habit of energy conservation. For example, turning off the pressure fan of floodlight TV has huge energy-saving potential. Each ship has an energy-saving awareness training program for new and returning crew, to create a list of best energy practices, including the main crew on board, and what should be done to save energy.

Incentive Mechanism

The company shall establish a sound energy efficiency assessment and incentive mechanism on relevant internal functions and levels (including ships) for the purpose of better implementation of ship energy efficiency measures.

Cross-References

- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)
- [Ship Propulsion System](#)

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Ship Design Phases

- [Ship Design Process](#)

Ship Design Process

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Synonyms

[Ship design phases](#); [Ship design stages](#)

Definition

The ship design process is an iterative process that needs to meet various technoeconomic requirements, which are partly contradictory to each other. According to the mission characteristics and the completion, the design process can usually be divided into four steps: concept design, preliminary design, contract design, and detail design, where the first two steps can also be named as basic design.

Scientific Fundamentals

Concept Design

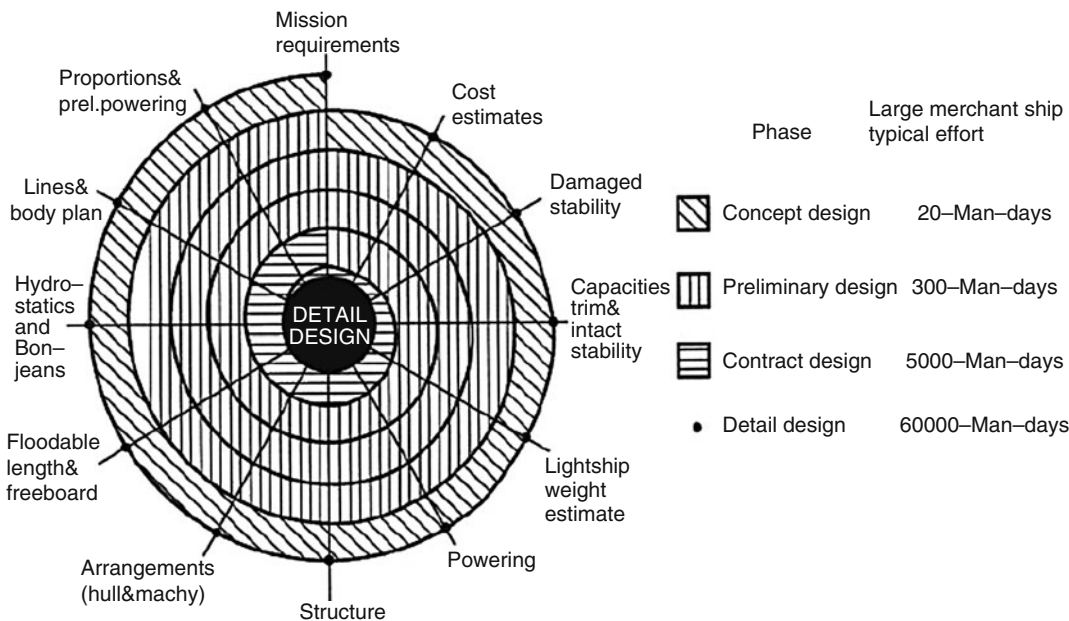
It is generally agreed that this is the most important design stage as it translates the mission or ship owner's requirements into naval architectural and engineering characteristics. This stage of ship design is also called feasibility verification, because it further demonstrates whether the design assignment is reasonable, especially for those design assignments without sufficient analysis and demonstration. This part of work is innovative, because the ship design must seek the most economical solution, in which an owner's requirements can be met or not must be considered (Papanikolaou 2002). It requires some experienced

naval architects, who can put forward several feasible schemes of the hull, turbine, and electronics, through drawing and calculation. Preliminary estimations of the fundamental elements of the proposed ship, such as length L , beam B , side depth D , draft T , block coefficient C_B , powering P_B , or alternative sets of characteristics, should meet the required speed range, cargo cubic, deadweight, etc. It includes preliminary light ship weight estimates usually derived from curves, formulas, or experience.

It is worth mentioning that the fundamental elements of a ship have a great influence on the technical and economic performance of the ship, which is the most important work to be solved in ship design (Schneekluth 1998). Whether the fundamental elements are suitable or not is an important factor affecting the overall design and the success of the new ship. Ship's main elements determine the whole aspects of the ship's technical performances and requirements, and various professional technical configurations, in turn, affect the major fundamental elements. Thus, it is a process that needs to iterate and approximate gradually to make sure the ship's fundamental elements are reliable and feasible. The ship design

process can be illustrated by the well-known design spiral, which can better describe the ship design process, originally introduced by J. H. Evans (Taggart 1980; see Fig. 1 below).

The very first effort, concept design, spent about 20 man-days in the 1950s. However, due to the development of computers and software, it takes less time to complete this stage now. Besides, the computer has probably had the greatest influence on the concept design stage (Nowacki 2010). It enables more design variants to be considered quickly, and each variant to be studied in more depth with fewer approximations, so provides more reliable estimates of cost and technical performance. In concept exploration, the naval architect starts with a very broad look at a wide range of options, studies the more promising, and ends up with a single preferred option (Tupper 2013). If the ship owner needs a conventional ship without special requirements, the design process will be much easier for the shipyard, as there exist ships whose information are similar to the new designed ship in size and type, which are usually named as “mother ships.” As a result, the cost will be lower relatively. On the contrary, if the new designed ship does not have information of mother ships to be referenced,



Ship Design Process, Fig. 1 Design spiral, J.H. Evans 1959. (From Taggart (1980))

the shipyard needs to do a lot of calculation. Essentially, the owner's requirements, operating environments, and national and international regulations and conventions must be taken into consideration. Finally, the selected concept design is to be used as a talking paper for obtaining approximate construction costs, which often determine whether to initiate the next level of development, the preliminary design (Taggart 1980). By the end of the concept design, several things should be settled: (1) feasibility to produce a ship to meet a ship owner's requirements within cost, (2) the overall characteristics agreed by ship owner, and (3) good estimate of the cost of acquisition and the through-life cost of the ship.

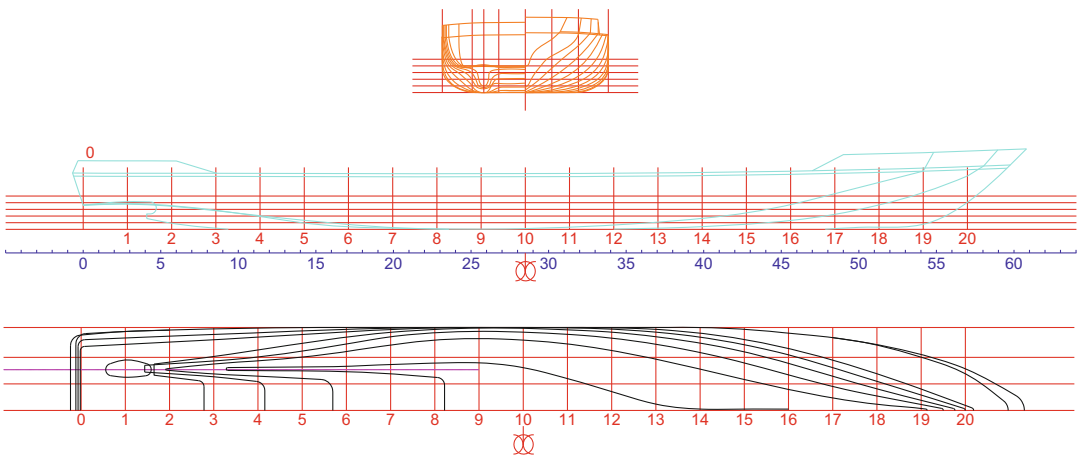
Preliminary Design

A ship's preliminary design further refines the major ship characteristics affecting cost and performance, which encompass objectives such as length, beam, draft, horsepower, and deadweight, so as to satisfy the owner's requirements and to achieve an optimal solution with respect to a set of economic criteria. This stage requires some major technical documentation of the new ship, including lines plan that is unique to ship design (shown in Fig. 2), general arrangement draft, drawing of midship section, calculation file of structural strength, demonstration files of estimated vessel speed and cargo cubic, and specification list of main equipment and materials. Figure 3 shows a

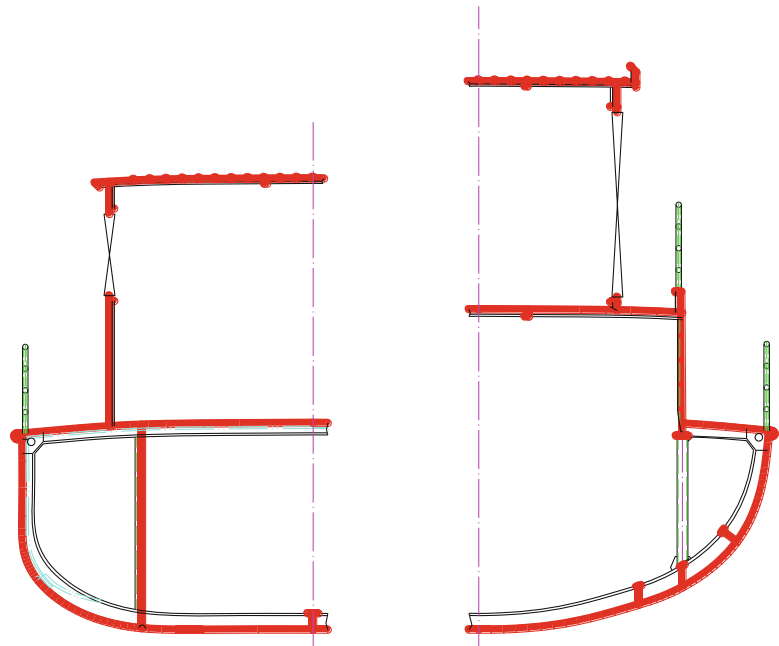
typical profile of a vessel. Its completion provides a precise definition of a vessel that will meet the mission requirements. Therefore, the outcome of the preliminary design forms the basis for compilation of the shipbuilding contract between the owner and the appointed shipyard.

Any vehicle must comply with relevant standards to ensure its safe operation, and ships are no exception. It is essential that the determination of all technical factors of ship design is subject to compliance with the specifications of various national and international maritime rules and regulations, which are enforced by national and international authorities (flag and port states, IMO) or by an internationally recognized classification society. In case of a lack of regulatory specifications, it is understood that the designed and built ship corresponds to the modern state of the art in shipbuilding science and technology (Papanikolaou 2014).

The combination of concept design stage and preliminary design stage is also known as basic design. Based on the results of the basic design, both the main technical features and the construction cost of an economically efficient vessel can be reliably estimated. Thus, the shipyards may proceed with the preparation of a tender to the interested ship owner. In case the tender is accepted, the more detailed and demanding third and fourth design stage are to be completed (Papanikolaou 2014).



Ship Design Process, Fig. 2 Lines plan of a vessel. (From Huang (2017))

Ship Design Process,**Fig. 3** Typical profile of a cargo ship. (From Huang (2017))

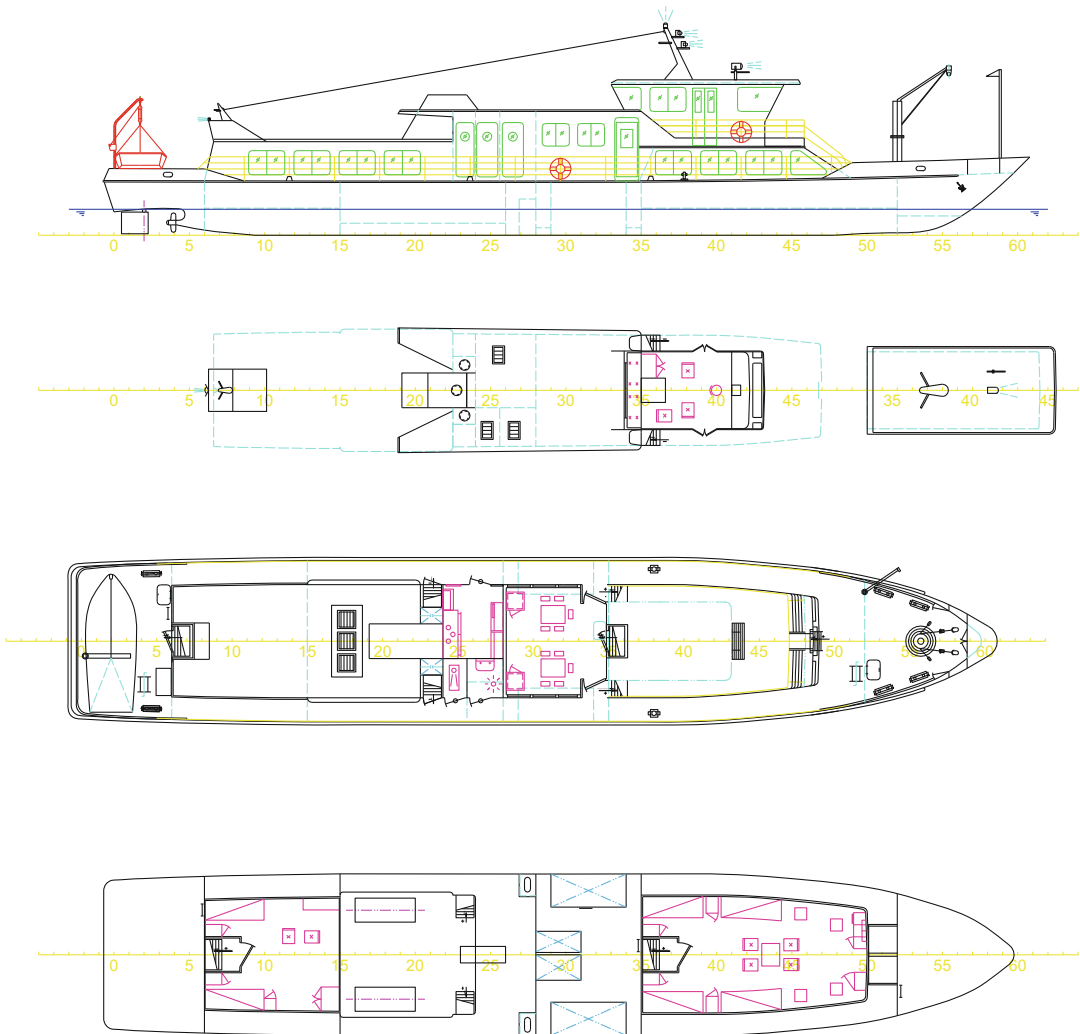
The difficulty of basic design will vary with the degree of departure from past practice. Some ship-operating companies are closely tied to successful previous designs, and they will permit little variation from these baselines in the development of replacement vessel design. If the prospective mission appears to parallel existing operations, this may be a sound approach. Consequently, in such situations, basic design may be limited to examination of minor modifications to dimensions, powering, and arrangements (Papanikolaou 2009). At the other extreme, totally new seagoing missions, such as the ocean transportation of liquefied natural gas (LNG), when first introduced, cause the designer to begin with a blank piece of paper and proceed through rational design engineering with crude assumptions subject to frequent and painstaking revision and development (Huang 2013; Taggart 1980).

Contract Design

The objective of this stage is the completion of the necessary calculations and naval architectural drawings, as well as the drawing up of the technical specifications of the ship's building, which all form indispensable parts of the formal shipbuilding contract between the ship owner and the

appointed shipyard (Taggart 1980). This design stage involves a more precise description of ship's hull form, which is based on faired set of lines, the exact estimation of the powering for achieving the specified speed based on model tests in a towing tank, the theoretical or experimental analysis of the behavior of the designed ship in waves, the analysis of the ship's maneuverability properties, the effect of number of propellers on hull form, details of the ship's structural design, use of different types of steel, spacing and type of frames, design of the ship's auxiliary/supply networks, and so on. Paramount, among the contract design features, is a weight and center of gravity estimate, considering the location and weight of each major item in the ship. The final general arrangement is also developed during this stage. Figure 4 shows the general arrangement of a vessel. This fixes the overall volumes and areas of cargo, machinery, stores, fuel oil, fresh water, living and utility spaces, and their interrelationship, as well as their relationship to other features such as cargo handling equipment and machinery components.

The accompanying specifications delineate quality standards of hull and outfit and the anticipated performance of each item of machinery and equipment. They describe the tests and trials that



Ship Design Process, Fig. 4 General arrangement of a vessel. (From Huang (2017))

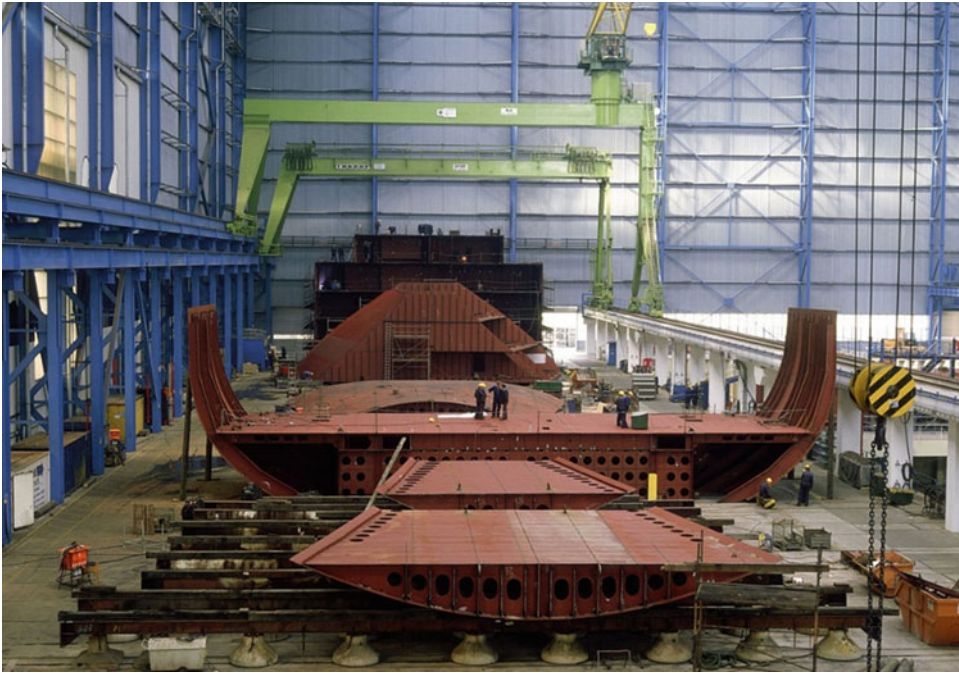
shall be performed successfully in order that the vessel will be considered acceptable. A set of drawing, numerical studies, and technical specifications should be developed during contract design, and for more details, please refer to Taggart 1980. These technical documentations shall be examined and approved by the ship owner and classification society as the basis for the construction design or contract signing.

Not everything envisaged in the concept design and preliminary design will work out as planned, and discussions between the ship designer and owner will be an ongoing process. A record must be kept on who made the decisions

and when safeguards were taken to ensure that changes are made only by those with the necessary authority person. The contract design may be carried out by the same group as produced the concept design and preliminary design, or it may be contracted out, perhaps to a potential builder or to some third party. Ideally the same CAD system will be used, and this must include the safeguards controlling design changes (Tupper 2013).

Detail Design

The final stage of ship design is the development of detailed working plans. These plans are the installation and construction instructions to the



Ship Design Process, Fig. 5 Ship construction (<http://www.baidu.com/>)

ship fitters, welders, outfitters, metal workers, machinery vendors, pipefitters, etc. Figure 5 shows the ship construction scenario. One unique element to consider in this stage of design is that up to this point, each phase of the design is passed from one engineering group to another. At this stage the interchange is from engineer to artisan, that is, the engineer's product at this point is no longer to be interpreted, adjusted, or corrected by any other engineer. This engineering product must unequivocally define the desired result and be producible and operable (Huang 2013).

A characteristic of this phase is that, while the generated drawings and specifications are the outcome of studies and work of expert engineers, the subsequent implementation of the designs into practice depends solely on the capabilities of the shipyard's production units, in terms of both hardware infrastructure and human resources (Papanikolaou 2014). When a new ship is built, the actual situation will inevitably be different from the original design. Although the weight and center of gravity had been calculated in detail, it is impossible to be completely accurate. It is also common to change the material or

equipment specifications without affecting the basic performance of the new ship. In this way, the weight, center of gravity, general arrangement, and structure of the ship will be changed. Therefore, after the completion of the ship's construction, the completed drawings should be updated according to the modification during the construction period. Besides, based on the real ship tilting test's results, the designer needs to modify the original relevant calculation results (such as stability, anti-sinking and floating state), complete other tests, and write the report, which are valuable data for ship design and research (Cheng and Pan 1988).

Accordingly, from mission requirements to the design work, although the design of the ship can be divided into several different stages, but each stage is not independent, they are intertwined in a spiral way forward. Certain calculations, drawings, and instructions are required at each stage. The forward stage is the basis of the later stage design, and the latter stage is also the deepening and development of the previous stage work (Watson 1998). However, the four stage of the ship design and the specific content

of each stage are not immutable, which often be different according to the specific circumstances (such as the urgency of the task, the complexity of the products, complete situation of the references, etc.). In some case, there is no clear boundary between each stage. For example, in the ship-type demonstration and protocol specification of some important products, most of the complete technical documentation should be supply in the design stage. All in all, ship design is a complex and hard process; there will be a lot of documentations produced to define the ship for the users and maintainers during the whole design process.

Key Application

The ship design process is a standardized design process that has been accumulated over hundreds of years of experience. Any ship that is to be built must go through this stage, from the most common bulk carrier to high-performance and high value-added LNG ship, such as bulk cargo vessel (shown in Fig. 6), container vessel (shown in Fig. 7), and LNG ship (shown in Fig. 8). This design process is inseparable to the construction of ships. Thus, it applies to all regulated and legally operated ships, including civilian ships and warships.

Ship Design Process,
Fig. 6 Bulk cargo vessel
(<http://www.baidu.com/>)



Ship Design Process,
Fig. 7 Container vessel
(<http://www.baidu.com/>)



Ship Design Process,
Fig. 8 LNG ship ([http://](http://www.baidu.com/)
www.baidu.com/)



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Ship Design Stages

- [Ship Design Process](#)

Ship Dismantling

- [Complete and Partial Dismantling](#)

Ship Electrical Design

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Synonyms

[Shipboard electrical design](#); [Shipboard electrical system design](#)

Definition

Ship electrical design is one of the major parts of ship design. It handles all design issues in ship electrical system to meet the requirements of ship-owner and guarantee ship quality and safety according to specific ship electrical design standards and specifications.

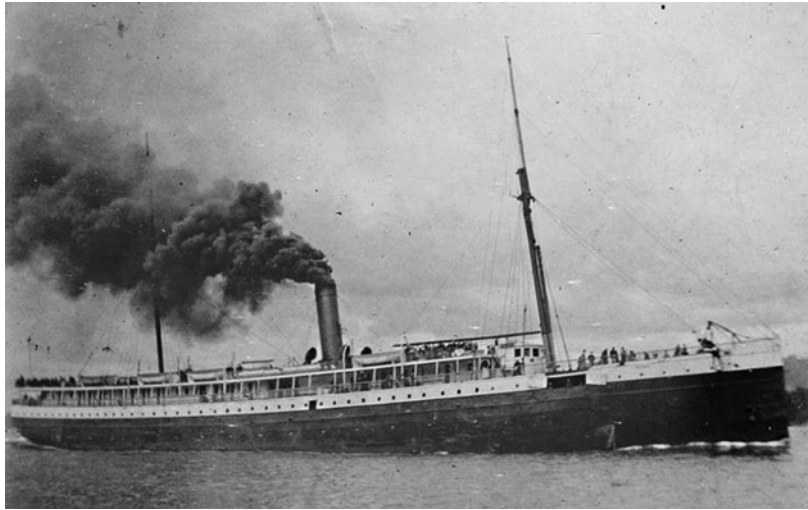
Scientific Fundaments

Background and Brief Introduction

A ship or vessel can be considered as a floating city, which, as a typical town or city, also needs the underlying conveniences to support the life aboard. One of the ship's necessary amenities is the electrical system because most of the marine equipment and other instruments on ships are electrically driven. Thus, there is a considerable demand for an electrical power, which makes a

Ship Electrical Design,**Fig. 1** The SS Columbia

(<https://en.wikipedia.org/wiki/>)



ship electrical system very significant. However, compared with the electrical system on the land, a ship electrical system has the following prominent characteristics (Ma 2009).

1. There are special requirements for the marine electrical equipment due to the harsh environmental conditions on a ship. Namely, the marine electrical equipment is directly exposed to the complex natural environment facing with the high heat, high humidity, low temperature, salt spray, mold, solar radiation, and electromagnetic interference.
2. The ship power system is a typical independent power system with limited capacity. Therefore, it is difficult to ensure an excellent power quality of ship power system with complex load conditions, which make it susceptible to load fluctuations.
3. The ship electrical system has high power density, and a large number of ship electrical equipment are compactly installed in the ship's cabin within a limited space. Therefore, high reliability is required to reduce the number of equipment repairs and replacements.
4. There are interactions and interferences between various electrical equipment due to the narrow space and EMC/EMI issues.

Starting with the earliest records of a commercially available shipboard electrical system which

dates back to the 1880, the electric light system was first installed and operated on the SS Columbia (shown in Fig. 1). The SS Columbia was equipped with 120 incandescent lights, which were distributed on several circuits and powered by four belt-driven 6 kW dynamos (Skjong et al. 2015). The electrical power plants in modern ship have grown in size and complexity with the growing demands on ship sizes, speed, economy, safety, comfort, and convenience. From the original electrical power of a few kilowatts at 120 V, some of today's navy and cruise ships require an electrical load of over 100 MW at a voltage of 11KV or higher. In the meantime, shipboard electrical system design has changed from traditional low-voltage to medium-voltage generation and distribution due to higher power requirements (Patel 2012).

Design of a ship electrical system includes parameters calculation, equipment selection, and electrical schematic design for ship power system, electric motor drive system, lighting system, electric propulsion system, ship interior communication system, radio communication and navigation system, and ship automation system, which should be in accordance with the requirements of shipowner and relevant standards, rules, and specifications to ensure ship quality and navigation safety.

Traditionally, shipboard electrical system design is supported by numerous electrical system

network analyses in accordance with applicable regulatory body requirements. These electrical system analyses are, namely, electrical system load analysis, voltage drop calculations, electrical load flow calculations, electrical short-circuit analysis, and circuit breaker coordination and selectivity study.

The advent of all-electric ships and advances in components, such as the development of high-efficiency, power-dense, lightweight motors, transformers, drives, and other devices, require careful and, in many cases, entirely unique integration into the ship system design and development. These imperatives motivate the use of tools for the system-wide analysis of shipboard electric power systems. In order to perform these analyses, digital simulation of electrical systems is essential. For example, in order to establish the physical laws, governing power system components and their interactions data are collected in large numbers and analyzed to predict the behavior of a system. Digital modeling and simulation are fast, accurate, and convenient approaches for predicting power system behavior. The new technology of electric-driven propulsion power systems requires additional system analyses such as power system dynamic analysis, harmonic analysis, network transient analysis, electric motor drive performance analysis, and extensive modeling simulation of power generation and distribution. These analyses are usually done by the use of computer-aided engineering and design tools (Islam 2004).

As an example, it is necessary to calculate the load factor in ship electrical load analysis. Not all loads are turned on at all times, and many come one after another, each for a short duration of time in a set sequence. Therefore, the required generator capacity in kW is always less than sum total of kW rating of all connected loads. This time diversity in their demand is taken into account by assigning a load factor to each load equipment based on the heritage data on a similar ship built and operated earlier. The *load factor* (*LF*) – also known by other names such as demand factor, diversity factor, utility factor, or duty factor – is defined as the average power over a period of time as a fraction of peak (or rated) power of the

equipment. Thus, the load factor indicates to what extent a specific load contributes to the total load on the generator, which aggregately powers all the connected loads. If the time-varying load power $p(t)$ is expressed in kW over any specific operating period T (e.g., at sea, in-port loading, etc.), then the load factor during that operating period is given by

$$\begin{aligned}\text{Load factor} &= \frac{\text{Average kW load}}{\text{Peak kW rating}} \\ &= \frac{1}{\text{Peak kW rating}} \frac{1}{T} \int_0^T p(t) dt \\ &= \frac{\text{Actual kWh used during } T}{\text{Peak rated kW} \cdot T \text{ in hours}}\end{aligned}$$

Obviously, $LF = 1.0$ for continuously on load, $LF < 1.0$ for intermittent load, and $LF = 0$ for standby equipment. For example, a 10 kW motor used at full load for 4 h and at one-half load for 2 h during a 12-h period on loading days at the port has a load factor of $(10 \times 4 + 5 \times 3) \div (10 \times 12) = 0.42$ during the daytime loading period. During other periods, it may have a different load factor such as zero at night (Patel 2012).

Design Standards

The standards for a ship electrical system design are formal documents, established by a consensus that provides the engineering and technical specifications, rules, guidelines, and characteristics for design, construction, and testing of ship electrical system and equipment. These standards address a range of issues, including various electrical protocols with the aim to meet the design performance requirements, maximize functionality and compatibility, facilitate interoperability, and support people's safety and health on a ship.

There are many groups of standards established and published by different organizations around the world, such as international standards, national standards, association standards, classification society standards, and military standards.

The international standards are developed by the international organizations for standardization, such as the International Electrotechnical

Ship Electrical Design, Table 1 The members of IACS (<http://www.iacs.org.uk/>)

Abbreviation	Name
ABS	American Bureau of Shipping
BV	Bureau Veritas
CCS	China Classification Society
CRS	Croatian Register of Shipping
DNV GL	Det Norske Veritas/Germanischer Lloyd
IRS	Indian Register of Shipping
KR	Korean Register of Shipping
LR	Lloyd's Register
NK	Nippon Kaiji Kyokai
PRS	Polish Register of Shipping
RINA	Registro Italiano Navale
RS	Russian Maritime Register of Shipping

Commission (IEC), the International Organization for Standardization (ISO), the International Maritime Organization (IMO), and the International Association of Classification Societies (IACS) as listed in Table 1. The international standards are usually a reflection of the experiences of industry experts, researchers, consumers, and regulatory authorities, and they contain the common requirements of many countries; one of such standards is the IEC60092-501 standard (Electrical installations in ships-Part 501: Special features-Electric propulsion plant) which was published by the IEC/TC18 (Technical Committee No. 18 of IEC).

The national standards are approved and published by the national departments for standardization. The example of the national standard is the ANS standard published by the American National Standards Institute. Further, the GB standards are the national standards of China issued by the Standardization Administration of China (SAC), and “GB” stands for “Guobiao” which represents Chinese national standard. The mandatory standards are prefixed with “GB,” and the recommended standards are prefixed with “GB/T,” such as “GB/T 29484-2013 Electrical Installations in Ships-Part 503: Special Features-AC Supply Systems with Voltage in The Range of Above 1kV up to And Including 15kV.”

The association standards are developed by special institutes, societies, and association. The

example of association standards is the IEEE standards published by the Institute of Electrical and Electronics Engineers. The IEEE is a historical and influential academic community in the world. Namely, standard formulation and publishing are major activities of the institutes. The IEEE Std-45 (IEEE Recommended Practice for Electrical Installations on Shipboards) is a recommended standard for electrical installations on shipboard which reflects the technological progress of the US ship electrical engineering and has an extensive reputation in the world. In fact, it is usually taken as a reference for the development of other relevant standards.

The classification society standards represent the standards developed and published by the classification societies, which are the nongovernmental organizations that establish and maintain technical standards for the construction and operation of ships and offshore structures. For instance, the “Rules and Regulations for the Classification of Ships” is a standard published by the Lloyd’s Register, which includes the topics about ship electrical system design, such as “main source of electrical power,” “supply and distribution,” and “electric propulsion,” in chapter Electrical Engineering, Part 6.

The military standards are used to achieve the standardization objectives of the military equipment. They are beneficial in achieving the interoperability ensuring the products meet specific requirements, commonality, reliability, cost of ownership, compatibility with logistics systems, etc. For instance, the US military standard is used to achieve the standardization objectives of the US Department of Defense, and it includes the defense specifications (MIL-SPEC), handbooks (MIL-HDBK), and standards (MIL-STD). These documents serve different purposes; namely, military specifications are used to describe the physical and/or operational characteristics of a product, military standards are used to detail the processes and materials to be used to make a product, and military handbooks are the primary sources of the compiled information and/or guidance (<http://www.hansandcassady.org/>). Among those standards, the MIL-STD-1399 is a notable one used for the US ship electrical system design.

The international standards relate to the development direction of advanced technologies and successful experiences in the ship design field. Furthermore, it is necessary to expand the international trade, so employing the international standards can be beneficial for the commonality and interoperability of ship design. Therefore, many countries adopt the international standards, and some countries directly use the international standards as their national standards. The comparison of the requirements for an alternating current distribution system in the IEC60092-101 standard (the IEC standard, “Definitions and general requirements”) and the GB/T 6994 standard (the national standard of China, “Electrical installation in ships-Definitions and general requirements”) is given in Table 2. It can be seen that the data listed in the second column are identical to the data listed in the third column, which shows that the GB/T 6994 standard adopts the same design requirements as the IEC60092-101 standard.

General Design Procedure

The design of a ship electrical system is an important part of the overall ship design, and it can be divided into four stages: concept design,

preliminary design, contract design, and detail design (Huang 2013).

In the concept design, technical designs and scheme demonstration are developed, and main electrical parameters (such as voltage level and frequency), distribution system, main power supply type and configuration of the main communication, and navigation equipment are determined.

In the preliminary design, the optimal technical solution is developed according to the design requirements. The key tasks include (1) evaluation of a whole load power of a ship to determine the capacity and number of generator sets; (2) drawing of the distribution system diagram for electric power, lighting, electrical drive, communication, and navigation system; (3) estimation of a short-circuit current to support the distribution equipment selection; (4) proposing of a preliminary order list of the main electrical equipment (e.g., the equipment for power supply, distribution, lighting, and navigation); and (5) estimation of a major equipment size (e.g., the main switchboard).

The contract design is an important stage in the entire design process, and it is based on the approved preliminary design and technical design

Ship Electrical Design, Table 2 Characteristics comparison of alternating current distribution system (IEC60092-101 at 2002, GB/T 6994 at 2006)

Characteristics	IEC60092-101	GB/T 6994
Voltage tolerance (continuous)	+6 to −10%	+6 to −10%
Voltage unbalance tolerance	7%	7%
Phase to phase voltage unbalance (continuous)	3%	3%
Voltage cyclic variation deviation (continuous)	2%	2%
Voltage transients		
Transients (slow), e.g., due to load variations tolerance (deviation from nominal voltage)	±20%	±20%
Voltage transient recovery time	Maximum 1.5 s	Maximum 1.5 s
Fast transients, e.g., spikes, caused by switching, peak impulse voltage amplitude	5.5U _{nom}	5.5U _{nom}
Rise time/delay time	1.2 μs/50 μs	1.2 μs/50 μs
Harmonic distortion (voltage waveform)		
Total harmonic distortion	Not to exceed 5%	Not to exceed 5%
Single harmonic	Not to exceed 3%	Not to exceed 3%
Frequency characteristics		
Frequency tolerance (continuous)	+5% to −5%	+5% to −5%
Frequency cyclic variation tolerance (continuous)	0.5%	0.5%
Frequency transient tolerance	+10% to −10%	+10% to −10%
Frequency transient recovery time	Maximum 5 s	Maximum 5 s

assignment. The main tasks in this stage are as follows: (1) a load power are further evaluated; (2) a system diagram and a control diagram of each subsystem on a ship are drawn; (3) an arrangement of electrical equipment is realized; (4) a short-circuit current is calculated, and it represents the basis for relevant equipment selection and protection design; (5) selective protection design is realized and the corresponding analysis is conducted; (6) the routes of main power cables are determined; (7) layout diagrams of electrical equipment installed in the cabins are drawn (e.g., wheelhouse and chart room); (8) the ordering list of ship electrical equipment is finished; (9) the outlines of commissioning and trial trip are proposed; and (10) other necessary designs and calculations, such as transient voltage drop and illumination calculations, are completed.

The last stage of the ship electrical system design is a detail design or production design. This stage is based on the technical specifications and the approved results obtained in the contract design stage. In this stage, the complete set of special drawings and technical documents on the manufacture, installation, commissioning, and inspection are compiled combining with the production requirements of a contract manufacturer.

Key Applications

The design of a ship with reasonable technical performance and good economic performance has to be performed in accordance with the ship's purpose, the requirements of shipowners, and relevant rules and regulations. As one of the major parts of ship design, the design of a ship electrical system has become more and more important with the increase in the large-scale, automation, and energy-saving requirements of ships, which have enhanced the interconnections between diverse subsystems on a ship and brought great challenges to the ship electrical system design. Though there are differences in the design of different ship electrical systems, the key tasks of ship electrical design are similar, and they are introduced in the following sections (Huang 2013).

Ship Electrical Design for Ship Power System

The ship power system integrates the power supply devices, power distribution devices, and loads by a certain manner. The ship power system denotes a generic term for all devices and networks on a ship which perform generation, transmission, distribution, and consumption of the electricity. The design of ship power system mainly includes the determination of system voltage, frequency and distribution, selection of power supply devices, design of distribution equipment, power network, and protection.

The ship power supply equipment mainly includes the main generators and main switchboard (shown in Fig. 2). The control and monitoring tasks of the main generators are all implemented by the main switchboard, and the main functions of power distribution are also completed by the main switchboard. Therefore, the design of the main switchboard is an important part of the power system design and a key to guarantee the quality of a ship power system.

The power distribution equipment is in a very close relationship with the ship power grid. The main task of a power distribution equipment design is to reasonably determine the power mode according to the characteristics and capacity of different loads and select appropriate switches, cables, and other needed components.

The ship power system protection design includes the selection of the proper protection devices and strategies to protect the power consumption equipment (loads) that will guarantee the continuity of power supply and improve the survivability of ship power system.

Ship Electrical Design for Electric Motor Drive System

The ship electric motor drive system (e.g., the motor drive system for ship auxiliary machinery, anchor cable, crane, steering gear, etc.) is an important part of ship electrical system which consumes from 70% to 90% capacity of a ship power system. Besides, it reflects the technical level of a ship electrical equipment.

The design of electric motor drive system includes selection of a control equipment of a motor power circuit (including equipment used

Ship Electrical Design,**Fig. 2** Shipboard main switchboard (<https://www.km.kongsberg.com/>)

for motor start, power converter, and control equipment), an auxiliary equipment of a control circuit (telemetry equipment, signal equipment, and alarm equipment) and cables for power and control circuits (including cable type, number of cable cores, and cable cross section), drawing of the electrical wiring diagram of a motor drive system, and determination of the installation and arrangement of control equipment with consideration of safe operation and convenient manipulation.

Ship Electrical Design for Lighting System

Ship lighting system is directly related to the safety of navigation and daily life of the crew. When the ships sail in the sea and go through some ports, the work conditions often change. Therefore, the marine lighting must meet the requirements in different conditions. The marine lighting system can be divided into the main lighting, emergency lighting, temporary emergency lighting, navigational lamps, and signal lamps.

Lighting system should be designed according to the requirements for lighting and navigation defined in the classification rules, such as that of a steel seagoing vessel and the relevant clauses in the SOLAS and “International Regulation for Preventing Collisions at Sea.” For different countries

and sailing areas, the requirement norms and rules can be different, and all of them have to be satisfied.

The design of the lighting system involves the design of a power supply of lighting system and its control. The design mainly includes (1) selection of the optimal number, path, and location of distribution boxes and lighting fixture type, (2) illumination calculation according to the type and arrangement of the selected lighting fixtures, and (3) design of a single line diagram and layout of the lighting system (Zhao 2013).

Ship Electrical Design for Electric Propulsion System

The ship electric propulsion systems have experienced the impressive growth over the last decade. The growth of an offshore exploration is one of the drivers for such a development, but the advances in power electronics and increased sensitivity of fuel costs have also contributed significantly to the technical development and applications of ship electric propulsion system. Compared with the conventional mechanical propulsion system, an electric propulsion system has advantages of simpler design (e.g., the complicated reduction gear which was used to run the ship is omitted), reduction of the onboard noise, fuel efficiency improvement (due to the ability to

constantly maintain an optimal rotational speed that provides good fuel efficiency in the generators used for production of a propulsive electric power), and lower life-cycle costs. The electric propulsion systems are often used on icebreakers and oceanographic research vessels that take advantage of the aforementioned operational benefits and large passenger cruise ships and other ships that emphasize the cost and silent operation.

The typical electric propulsion system (shown in Fig. 3) consists of the propulsion generator, switchboard, propulsion transformer, propulsion motor and its power converter, and relevant control system.

The design procedure of electric propulsion system can be divided into a few steps: (1) determination of electric propulsion scheme, (2) main circuit parameters selection (selection of power, speed, and voltage of a generator and propulsion motor according to the general scheme and propeller parameters), and (3) drafting of the main circuit and control circuit, listing the order information of a major equipment. In general, design tasks mainly include the technical documents and schematic diagram development of the electric propulsion system. Typical technical documentation on a ship consists of the specification of ship electric propulsion system, schematic diagram of the main circuit and control circuit (including various manipulation and automatic control

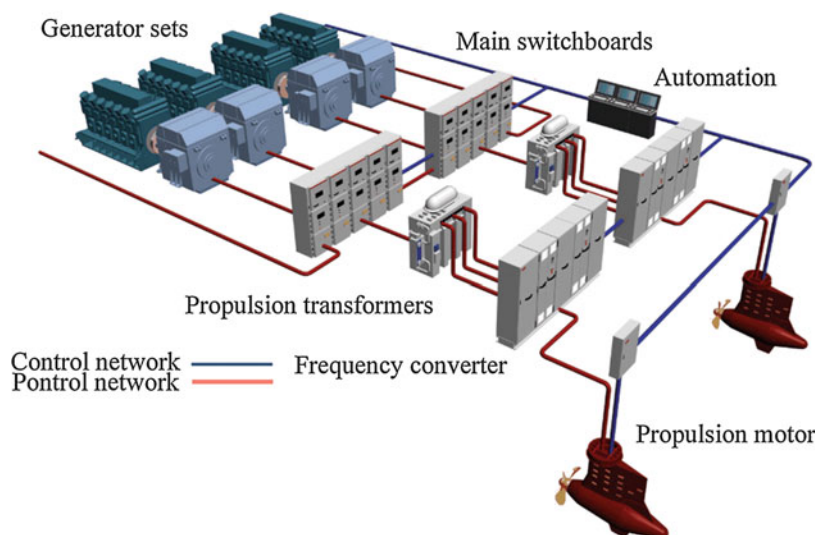
systems, protection, and signal indications), layout and structure diagram of a main switchboard, electrical wiring diagrams and structure diagrams of a console, control board, and other equipment in electric propulsion system.

Ship Electrical Design for Ship Interior Communication and Signal Devices

The ships interior communication and signal devices perform like the human nervous system, which are distributed throughout the ship cabins, making the entire ship an organic whole. Ship interior communication systems can generally be divided into ship telephone communication systems and ship broadcasting systems. The ship telephone communication systems mainly include three parts, wherein the first part is the navigational command telephone that enables the wheelhouse to communicate with the navigation-related departments, and it is essential to ensure the navigation security. The second part includes the special telephones for several important work sites, guaranteeing the rapidity of communication on a ship. The third part consists of automatic telephones installed for the communication in daily work and life.

On the other hand, to get the information on the operation of various kind navigation mechanisms, the telegraph system, indicators, foghorns, and signal control devices should be installed in a

Ship Electrical Design,
Fig. 3 Simplified single line diagram of the power plant with a propulsion system
(<https://proyectosnavales.com/>)



ship control equipment. Besides, the general alarms, fire alarms, fire detection alarms, radio intercoms, and shared antenna systems should be designed and installed such that to transmit the acoustic-optic signals and instructions.

Ship Electrical Design for Radio Communication and Navigation System

The radio communication can be a conventional radio communication or satellite communication, which is used for distress alert, search, rescue work, daily public communication, and business contact. The design of a ship radio communication system mainly includes layout diagram of the radio communication equipment, layout diagram of antennas, layout diagram of radio communication equipment in ship cabins, and the detailed list of the spare parts according to the specifications and shipowner's requirements.

The mission of a ship navigation system is to ensure the navigation safety of maritime transport and marine engineering ships and guide the ships to reach their destinations on a scheduled route accurately. Marine navigation equipment mainly consists of a compass, an adaptive autopilot, echo sounder and speed log, a rudder angle indicator, GPS receiver, foghorns, anemometer, and automatic identification system (AIS). The design of a

ship navigation system includes the creation of a navigation equipment scheme (according to the ship navigation area, specifications, shipowner's requirements, etc.), selection of a ship navigation equipment type, drawing of the equipment system diagram, and equipment layout.

Ship Electrical Design for Ship Automation System

The ship automation system refers to the automation control system and monitoring system which are composed of various instruments, elements, and computers. In this kind of system, the operation of a machine is supervised and controlled by a self-regulating mechanism which senses the machine performance and provides a rapid and accurate analysis and record on the related parameters. For instance, a machine can be regulated to the desired state or give an alarm to a malfunction without human interference by its automation control system. An automation system contributes to the reduction of the amount of improper operation of the engine and improves the safety and reliability of navigation, which becomes more and more important in the ship electrical systems.

The design of a ship automation system includes an engine room automation (shown in Fig. 4) design, navigation automation design,

Ship Electrical Design,

Fig. 4 Engine control room console (<http://ships.fish.hokudai.ac.jp/>)



and design of automation for loading and unloading. Engine room automation design includes the design of the engine room monitoring and early warning, main engine remote control, and automatic power station.

For instance, in the design of an engine room monitoring alarm system, it is necessary to take into account the automation mark of ship classification, as well as electromagnetic interference and cable type and clearance. The design of all marine automation systems should obey the rules, standards defined by special organizations, such as IMO (International maritime organization), local authorities, and IACS (International Association of Classification Societies) and the classification societies.

Cross-References

- [Concept Design](#)
- [Contract Design](#)
- [Preliminary Design](#)
- [Ship Design Process](#)
- [Special Marine Vehicle](#)
- [Surface Warships](#)

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Ship Energy Efficiency

- [New Technologies in Auxiliary Propulsions](#)

Ship General Design

- [Ship Overall Design](#)

Ship Hull Design

- [Ship Construction and Operation Impact on Energy Efficiency](#)

Ship Hull Lofting and Marking-Off

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Synonyms

[Numerical control \(NC\)](#); [Three-dimensional \(3D\)](#)

Definition

Lofting

Is a drafting technique to amplify the design drawing drawn with a certain ratio into a 1:1 full-scale drawing (or a 1:10 or 1:5 ratio drawing), and to use it as the basis for steel blanking and processing of the hull members (Liu et al. 2011).

Manual lofting

Includes full-scale lofting based on scale drawing and ratio lofting based on 1:10 (or 1:5) scale drawing.

Mathematical lofting	Is the technique to use mathematical equations to define the hull lines or the molded base, to establish a mathematical model, and to complete the hull lofting with a computer.
The content of hull lofting	Includes theoretical model line of hull lofting, frame line lofting, lofting of structural line, hull member development, and provision of information for the subsequent procedures.
Hull member marking-off	Is a technique to draw (print) cross lines and processing lines of a member on the flat steel plate and profile according to the member template, sketch, sample rod, and data provided by lofting. It is also necessary to arrange patterns of various member drawings reasonably on the materials in order to save materials and labor, and this procedure is called nesting (Liu et al. 2011).

smooth and fair. The line drawing could be used as the basis in all procedures in hull construction.

Theoretical model line lofting is a process to conduct the 1:1 full-scale lofting according to the station line of 1/10 or 1/20 of the ship length (the length between perpendiculars) with the lines drawing provided by the design department, so as to achieve the smoothness and correction of the three-sided projection of the hull lines drawing (Xu 2011).

1. Drawing of the lofting graticule: The graticules in the lines drawing are the projection lines of each group of hull lines on the corresponding projection surfaces and are the base for measuring the size in line lofting. Methods of drawing graticules include plumb line method, square gauge method, theodolite method, laser theodolite method, etc. (Fig. 1)
2. Drawing of the camber curve: The deck camber at the midship cross section is the largest, and its length gradually decreases toward the bow or stern. The curvatures of the camber curves at each cross section of the deck are the same as the curvature at the midship cross section, so only the camber curve at the midship cross section needs to be given and be used as the criterion to determine the shapes of deck camber at each cross section (Fig. 2).
3. Drawing of the deck center line: Generally, the offset of the deck center line is not given, in the lines drawing, but is instead obtained according to the offset of the deck side line and the height of the round of beam at the widest (breadth molded) point of the deck. When the camber

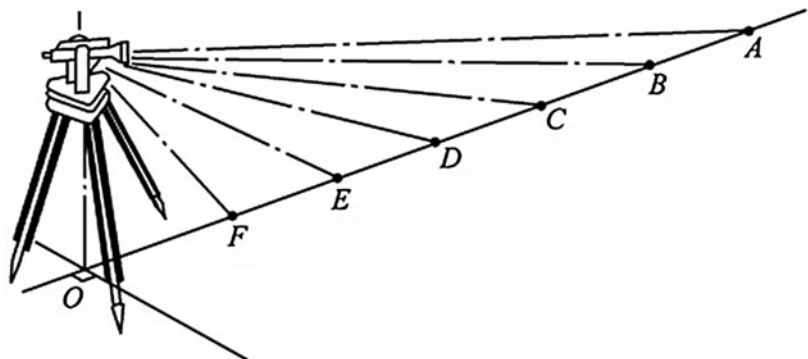
Scientific Fundamentals

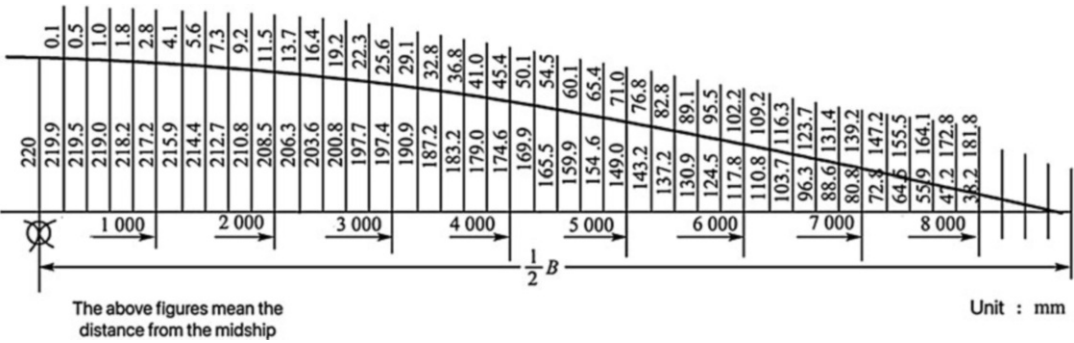
Hull Line Lofting

Line lofting is conducted in the initial stage of the hull construction, so that the design could be reflected and some unreasonable offsets could be corrected and perfected to make the hull lines

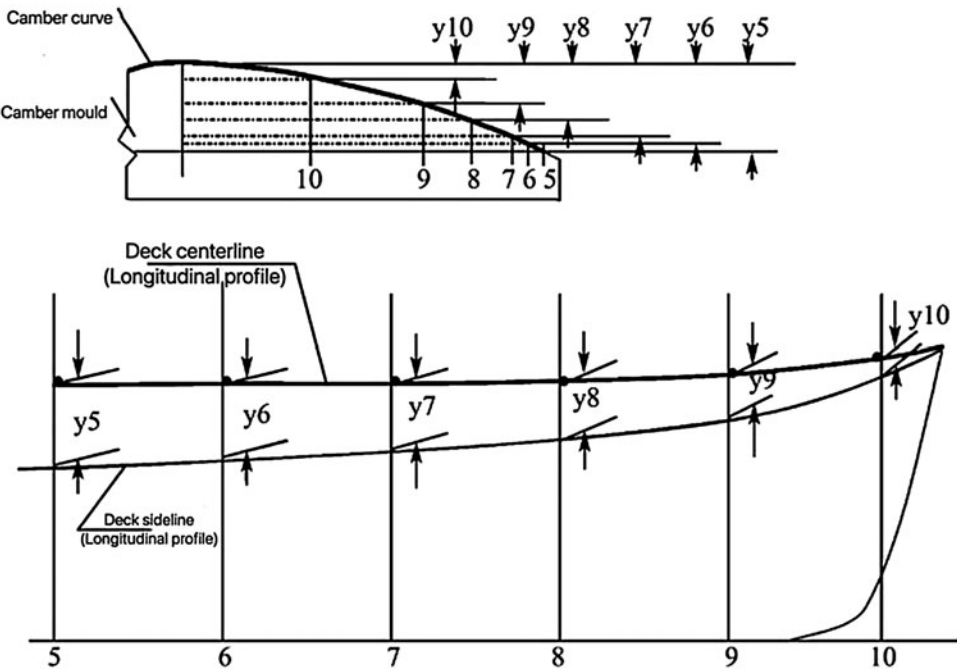
Ship Hull Lofting and Marking-Off,

Fig. 1 Graticules drawn with the laser theodolite





Ship Hull Lofting and Marking-Off, Fig. 2 The offset table of the deck camber curves



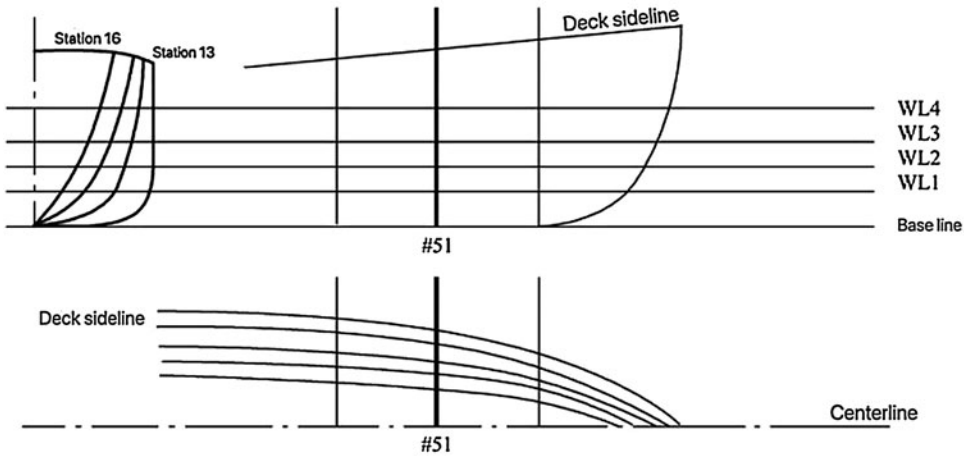
Ship Hull Lofting and Marking-Off, Fig. 3 Drawing of the deck center line

- curve and the deck sideline are drawn, the height of the deck center line can be obtained (Fig. 3) (Liu et al. 2011).
4. Theoretical model line of hull lofting also includes the lofting of waterlines, buttocks, body lines, bulwark top lines, etc., and the hull line drawing could be considered in theoretical wrap lofting.

Frame lines lofting is a technique to insert different frame lines in the sheer plans and half-

breadth plans according to the actual frame positions based on the theoretical model line of hull lofting, with the consideration of the given frame spaces in the drawings, and then to draw the frame body plan on the midship cross section according to the mutual projection relationship of the three basic projection surfaces (Fig. 4).

Lofting of frame lines at the screwshaft outlet is a technique to not only ensure the smoothness of the axle casing itself, but also ensure the smooth connection between the axle casing and



Ship Hull Lofting and Marking-Off, Fig. 4 Lofting of frame lines

the curved hull surface at both longitudinal and transverse levels. Usually, the smoothness could be achieved and examined through the creation of longitudinal oblique section, which is also an important means to obtain the radius of a circular arc (Fig. 5).

Lofting of Structural Line

Structure line lofting refers to the drawing of the theoretical projection lines of the longitudinal and transverse hull members and shell plating cross junction lines in the frame body plan according to the basic structure drawing, midlength section plan, shell-plating expansion plan, structure plan of bow and stern, block division plan, etc. provided by the design department (Ji 2005).

Lofting of the longitudinal members is a technique to draw the projection of the intersecting lines of longitudinal members with the molded hull surface and frame cross sections in the frame body plan. In the frame body plan, the perpendicular line segments between upper edge lines and lower edge lines, and between outer edge lines and inner edge lines, are the frame section lines of the longitudinal members. The shape enclosed by the two edge lines and the frame section lines at both bow and stern is the projection shape of the certain longitudinal member in the frame body plan (Fig. 6).

Shell plating cross junction line lofting is the technique where the shell plating cross junction

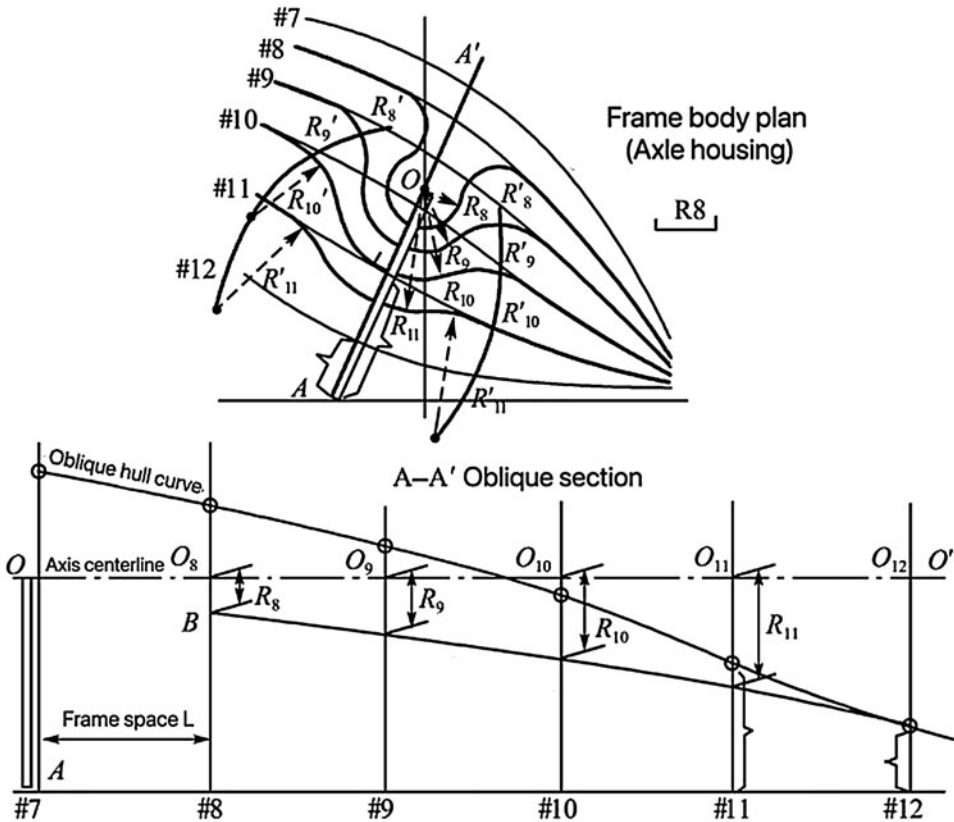
lines are finally arranged after the structural line lofting is completed in the frame body plan. Each plate may have a special name associated with its position on the hull. The plates are named after codes, usually arranged in alphabetical order from the centerline to the sides of the ship. The reasonable arrangement of plate junctions is closely related to the appearance of the ship, the utilization rate of steel, the processing of parts, the erection of sections, the welding technology, and the quality of hull construction.

Hull Member Development

The process of finding the true shape of a curved surface which cannot be shown in the projection drawing, spreading it out on a flat surface, as well as finding the true shape of the plane member which only has a projection in the frame body plan is referred to as development (Li et al. 2006).

The purpose of hull member development is to calculate the number and specification of the steel plates for the whole ship for drawing up the marking-off from sketches and the template to facilitate the marking-off on the flat steel plates.

When using the frame body plan to develop hull members, the three elements of development should be mastered, namely, to find the actual length of projection line, to find the curvature of frame (commonly known as the “brunt”), and to determine the datum line.



Ship Hull Lofting and Marking-Off, Fig. 5 Lofting of frame lines at the screwshaft outlet

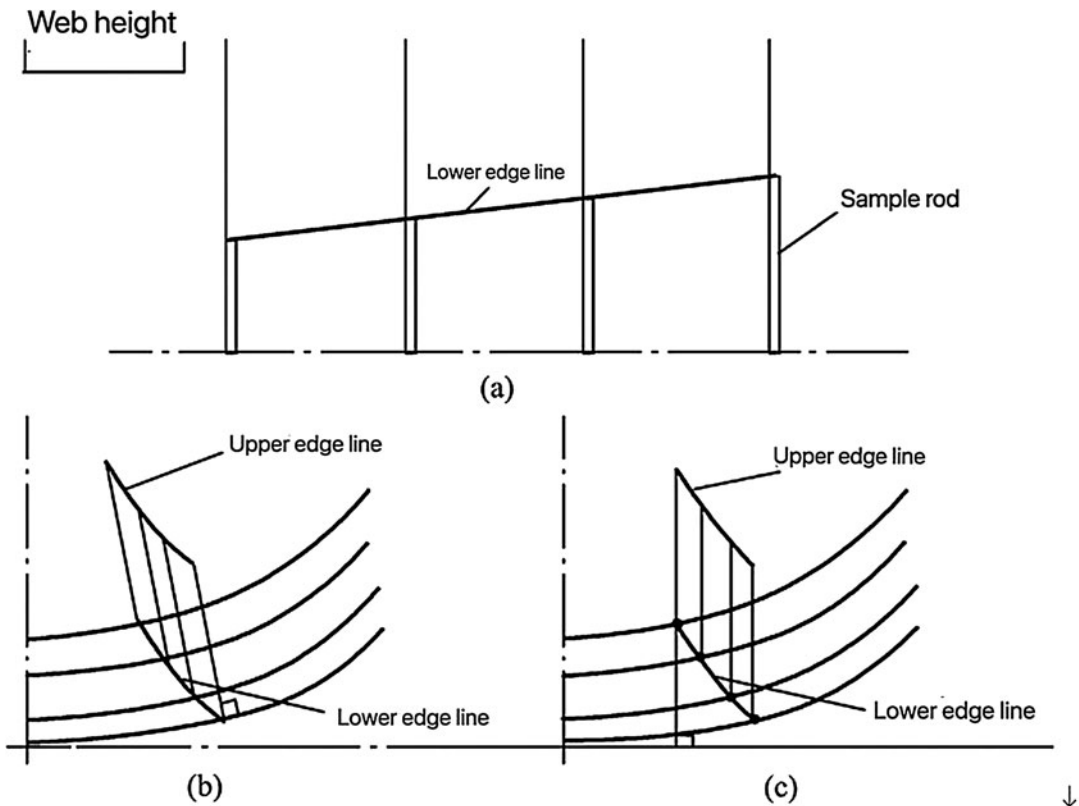
Hull longitudinal members could be categorized into two types: first, frame section perpendicular to the frame line; second, frame section perpendicular to the base line. The two types of longitudinal members could both be developed with mean normal method, in which the projection lines of the datum lines of the members to be developed are first drawn in the frame body plan, then the actual lengths of the member-longitudinal seams and the datum lines are found, and finally the hull member development drawing is drawn based on the datum line development (Figs. 7 and 8).

The center spot-squaring method used to develop hull plates is one type of mean normal method. In the frame body plan, if the frame chord lines of the plate are parallel or approximately parallel to each other, then a perpendicular line of the chord line could be drawn through the

midpoint of the middle frame line, intersecting with frame lines. This perpendicular line is called cross line (or cross hair) (Huang 2013).

The geodesic line used to develop hull plates is the shortest curve connecting two points on a curved surface. If the curved surface is developable, its geodesic line in the development drawing is a straight line. However, usually, the hull surface is undevelopable, with a mild curvature degree, and its geodesic line can be approximately regarded as a straight line in the development drawing and be used as a datum line to facilitate the development of hull plates.

The propping line used to develop hull members is a diagonal line added to each quadrilateral, which is approximately considered to be a straight line after development. The diagonal lines can “prop” the quadrilateral and set it into a unique fixed shape in making development drawing.



Ship Hull Lofting and Marking-Off, Fig. 6 Lofting of longitudinal members

Templates and Marking-Off

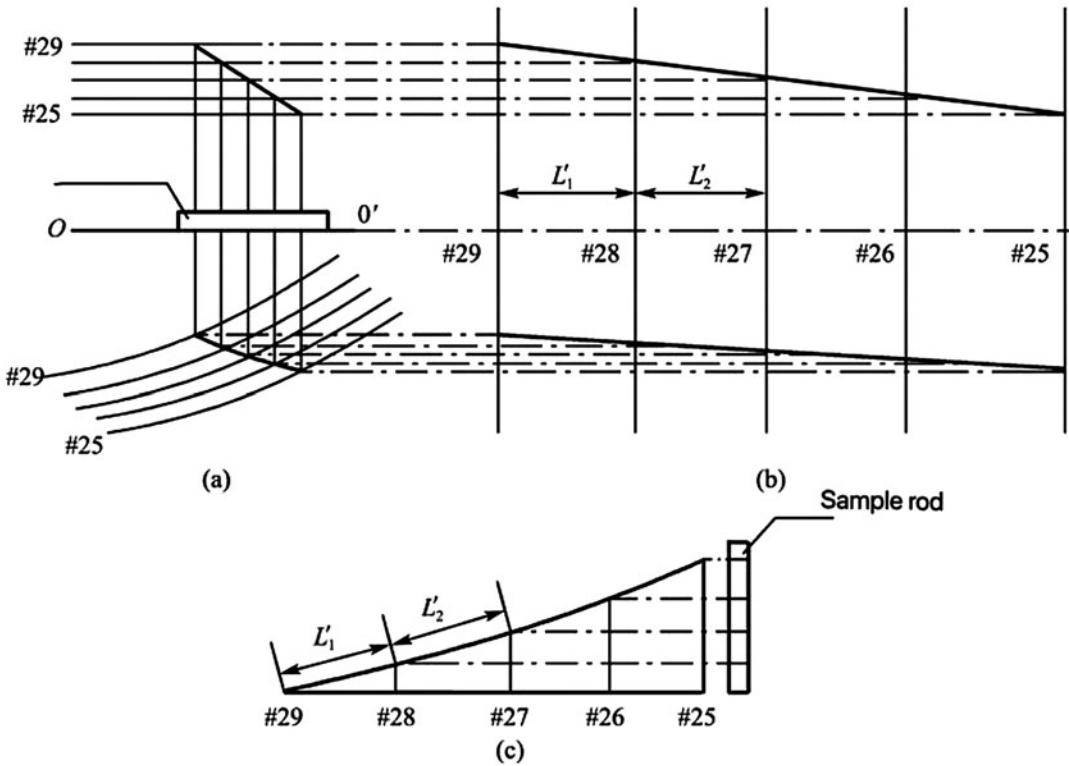
After the hull member development, based on developed shapes of each hull member, various sketches will be clearly drawn and various templates and mock-ups will be nailed with high precision, as the construction basis and quality standard for marking-off, processing, erection, and inspection (M 2012).

Templates can be categorized into different types by different criteria. By their usages, there are templates for marking-off, section mold, templates for assembly marking, templates for assembly angular, etc. By their spacial shapes, there are flat templates and stereotemplates (mock ups). By production materials, there are wood templates, flat iron templates, plastic templates, asphalt felt templates, cardboard templates, sheet steel templates, etc. By their generality, there are fixed templates, flexible templates, etc. (Fig. 9)

In the production of templates, the edge line of the curved side of the mold is the line of a certain frame on the plate. A set of such triangular templates will be produced according to each frame line of the plate to be processed. When being used, the triangular templates will be placed in their corresponding positions as indicated in the frame body plan (Figs. 10 and 11).

In the production of the mock-up, the mock-up is a curved template produced in the form of a fixed three-dimensional frame, which is equivalent to a three-dimensional skeleton cut off from the hull and connected to the plate. The curved surface of the mock-up represents the theoretical surface of the hull.

The cutting method of the mock-up base level includes the cutting perpendicular to frame section (tangent template) and the cutting not perpendicular to frame section (chamfering template).



Ship Hull Lofting and Marking-Off, Fig. 7 The development of the untwisted bottom-side girder

Sketches for marking-off refer to the sketches that can replace the template record and reveal the drawing information. The sketches have the advantages of saving material for template making and of being restored easily, thus being suitable for small-batched and simple-shaped hull members.

While drawing a sketch, first, the Cartesian coordinate system is determined on the drawing paper (generally, the developed straight directrix is used as the horizontal axis). The member shapes are not necessarily drawn strictly in proportion, where based on the basic shapes shown, the accurate graphics are determined through marking the dimensions. The sketch should also contain the necessary marks, symbols, and instructions.

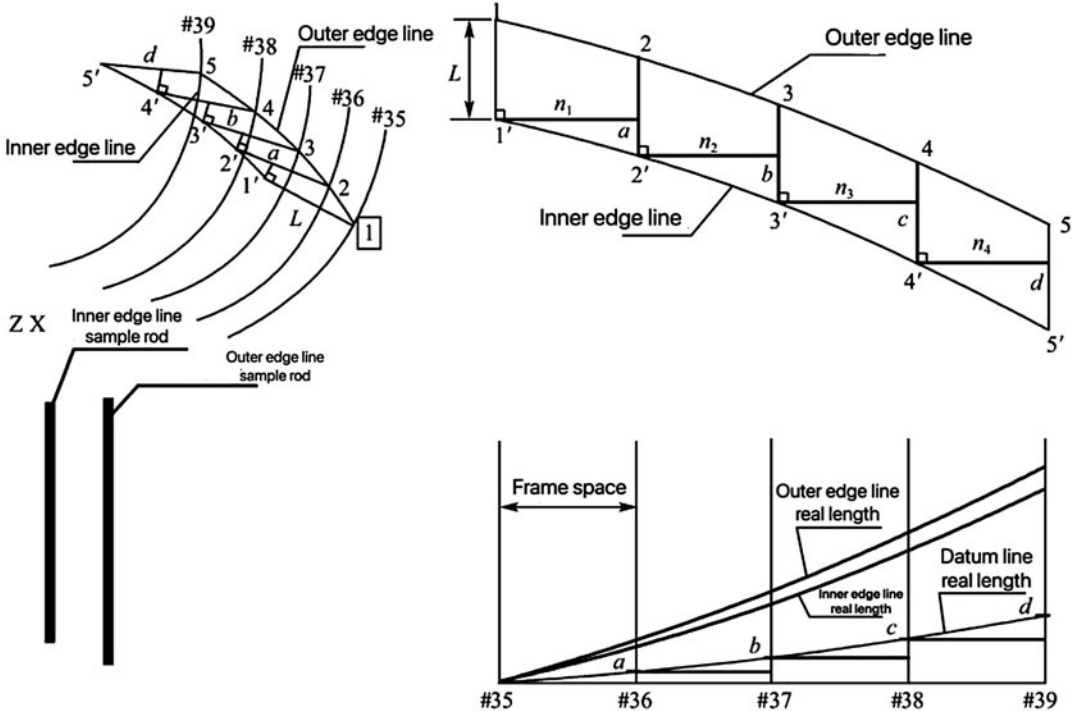
Marking-off is the drawing of cross lines and processing lines of the member on the flat steel plate and profile, according to the member templates, sketches, sample rods, and data provided by lofting.

Manual marking-off is divided into two major types – marking from templates and marking from sketches.

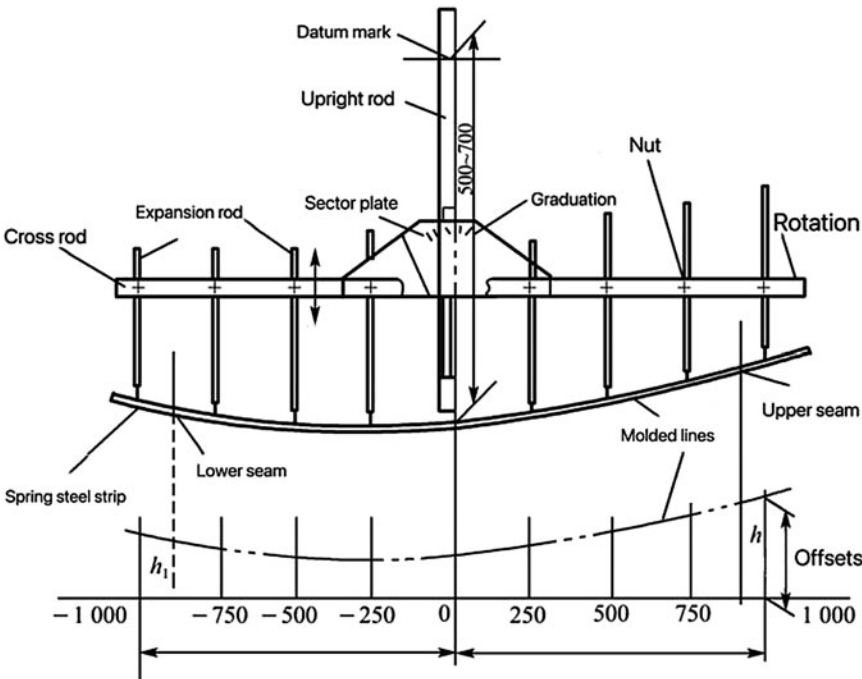
Marking from templates is the technique to directly draw the actual contour lines, member setup lines, reference lines for inspection, surplus lines required, etc., of the hull members on the steel or profiles by means of templates and to make necessary marks, symbols, etc. In order to retain these lines and marks, the permanent marks must be drawn with colored paint, or be carved with a punch or chisel, etc.

Marking from sketches is the technique to reproduce the actual shapes by means of drawing on the steel plate, based on the basic shape and precise dimension recorded in the sketch. The basic requirements are the same as those of the template marking-off (Fig. 12).

Optical projection marking-off refers to the procedure where, based on proportional lofting, after nesting, the developed member graphics are

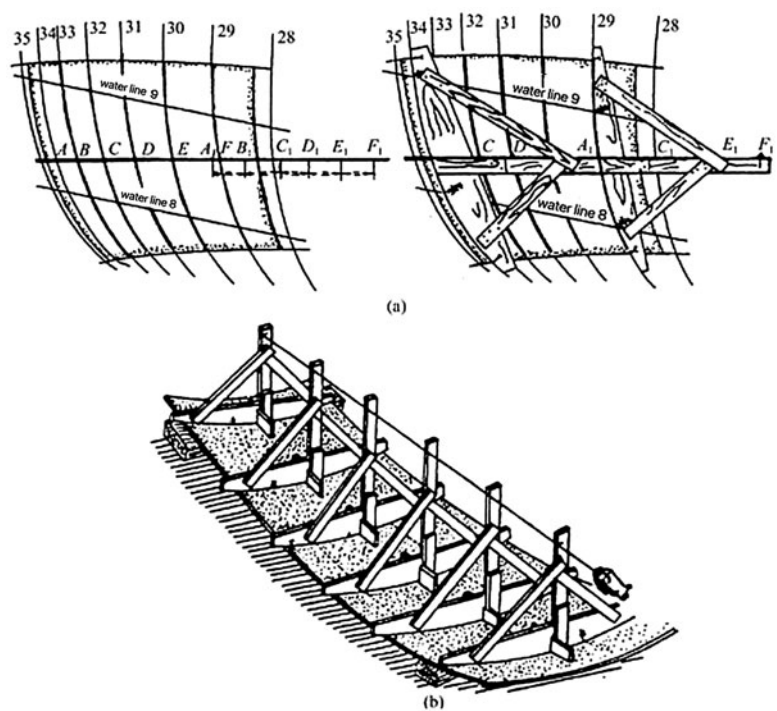


Ship Hull Lofting and Marking-Off, Fig. 8 Development of twisted shell-side-longitudinal girder



Ship Hull Lofting and Marking-Off, Fig. 9 Flexible triangle templates

Ship Hull Lofting and Marking-Off, Fig. 10 The production of triangular templates



Ship Hull Lofting and Marking-Off, Fig. 11 The use of templates to test curved plates forming

drawn into an accurate projection base map or photographed into a projection film with a smaller proportion, then enlarged with the optical projection-amplifying device and projected into a 1:1 member pattern on the steel plate, and finally the projected lines of the member are manually marked off.

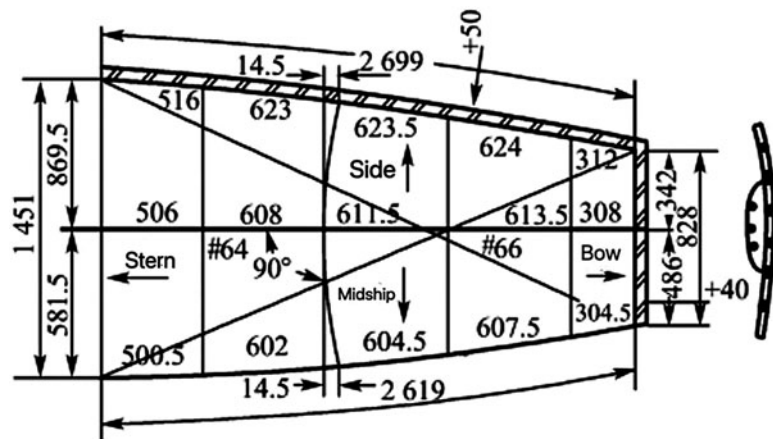
Electroprinting marking-off refers to the procedure where, based on optical projection

marking-off, the image of the member is projected with an amplification of 1:1 onto the marking-off steel plate covered with photoconductive powder with negative charge, so as to achieve exposure, development, and marking, leave the stitch of the marking-off, and thus complete the automatic marking-off.

Automatic nesting refers to the procedure where several parts of different shapes are

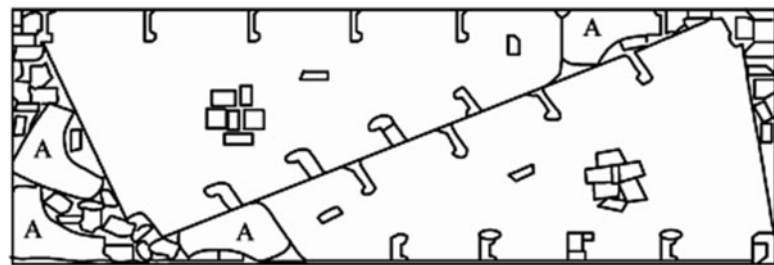
Ship Hull Lofting and Marking-Off,

Fig. 12 Marking-off from sketches for hull plates



Ship Hull Lofting and Marking-Off,

Fig. 13 Automatic nesting drawing



arranged in sequence on a set of rectangular steel plates according to the nonoverlapping layout requirements. The nesting diagram generated by the automatic nesting software is further optimized and corrected manually, and the result is processed by the computer, and the NC-cutting program is output for the use of NC-cutting machine (Fig. 13).

Hull-Mathematical Lofting

Hull-mathematical lofting is the use of the mathematical equation to define the hull lines or the hull surface, so as to establish a mathematical model and to complete the hull-lofting work through the high-speed calculation of the computer.

In the hull 3D-fairing calculation, according to the calculation sequence and calculation method arranged in advance by software, the drawing of hull lines, the offset table, and the matching relationship, between the shapes of the bow and stern and the middle of the hull, are all converted into

data in a certain format required for 3D-fairing calculation sequence and sent to the computer in the prescribed sequence.

The shell plate development calculation is carried out after the 3D fairing of hull lines, interpolation-producing frame offsets, and the hull structure ranking. Its main tasks are to generate the shell-plating longitudinal seam, to calculate developed plate parts, to provide various data charts and NC-cutting data required for the processing of the plate parts, and to provide data and information for the management of shipbuilding projects.

The mathematical lofting of the hull structure is to use the computer to generate the hull structure parts, and create the part drawing, the part nesting drawing, the NC-cutting program, and the data for processing and production management. The mathematical lofting of the hull structure includes at least the following three parts: structure line generation, part generation, and nesting (DM et al. 2015).

Conclusion

Ship hull lofting and marking-off are highly technical, difficult, and precise techniques. It is not only the first process of hull construction, but also provides a variety of accurate and reliable construction basis for subsequent processes of hull construction.

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Ship Loses Restoring in Waves

► Pure Loss of Stability

Ship Machinery Design

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Definition

Ship machinery mainly includes propulsion system, auxiliary power plants, and other auxiliary

machinery. Ship machinery design is one of the important branches of ship design, which is carried out through a specific process to guarantee functional stability, reliability, controllability, maintainability, and operational efficiency of ship. Ship machinery design is currently based on prescriptive regulations embodied in SOLAS and on rules issued by classification societies.

Scientific Fundamentals

Composition of Ship Machinery

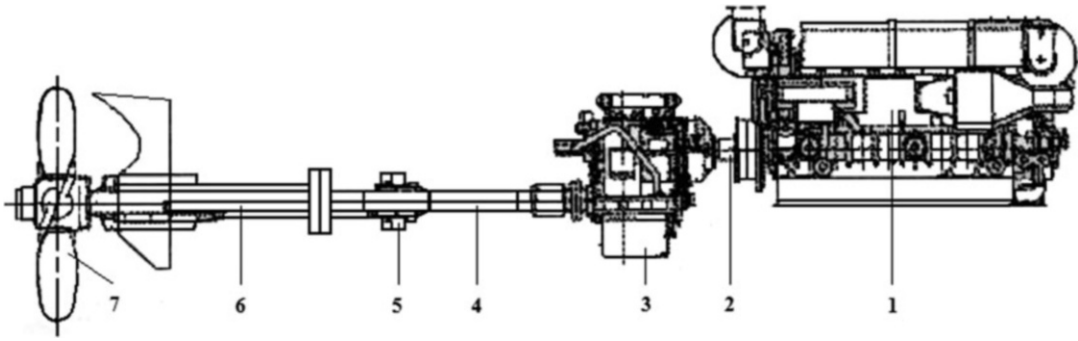
Main Propulsion Devices

Ship propulsion plants are the most important part of the ship machinery. Their function is to transmit power from the main engine of the ship to the propeller through transmission mechanism and shaft system and at the same time transmit the thrust generated by the rotation of the propeller to the hull. Ship propulsion plants generally include the main engines, the transmission mechanism, the shaft system and the propeller, etc., as shown in Fig. 1.

The form of power plants of modern ship can be divided into five categories: diesel engine propulsion power plants, steam turbine propulsion power plants, gas turbine propulsion power plants, nuclear power propulsion power plants, and combined power propulsion plants (Li 2010).

Generators

Generators are the main source of electricity for the motor drive of many auxiliaries, deck machinery, lighting, ventilation, and air conditioning equipment. A constant supply of electricity is essential for safe sailing and machinery operation, and therefore, the standby or additional capacity is necessary together with the emergency supply equipment. In the event of a main generating system failure, an emergency supply of electricity is required for essential services. This can be supplied by batteries. Batteries are usually served as the emergency supply, but most merchant ships have emergency generators. Emergency generators are generally diesel driven and located outside of the machinery space (Jiang 2014).



Ship Machinery Design, Fig. 1 Schematic diagram of propulsion plants: (1)main engine, (2) clutch, (3) gear box, (4) intermediate shaft, (5) bearing seat, (6) propeller shaft, (7) propeller. (China Maritime Service Center 2008)

Boilers

Boiler in a variety of forms can be found on different types of ships. If the main machinery is steam powered, one or more large water-tube boilers will be fitted to produce steam at very high temperatures and pressures. On a diesel engine ship, a smaller boiler (usually fire tube type) will be fitted to provide steam for other ship devices (Zhao 2012).

Steering Gears

The direction of the ship is controlled by the steering gear. As the ship moves in the water, the angle of the rudder at the stem determines its direction of movement. Modern ships are so huge that moving the rudder necessitates the use of hydraulics or electrical power. The steering gear provides a movement of the rudder in response to a signal from the bridge. The total system may be considered to be made up of three parts: control equipment, a power unit, and a transmission to the rudder stock. The control equipment conveys a signal of desired rudder angle from the bridge and activates the power unit and transmission system until the desired angle is reached. The power unit provides the force to drive the rudder to the desired angle. The transmission system is the means by which the movement of the rudder is accomplished (Jiang 2014).

Windlasses

The windlass is a usual anchor handling device used to handle both anchors. A new type of windlass, particularly used on larger vessels, is the split

windlass where a machine is used for each anchor. There are some variations in detail design of cable lifters and in their drives. Due to the low speed of rotation of the cable lifter and heaving anchor, a high gear reduction is needed when the windlass is driven by a high-speed electric or hydraulic motor. This is generally obtained by using a high ratio worm gear followed by a single step of spur gears between the warp end shaft and cable lifters. Alternatively, multi-steps of spur gears are used. Capstans with this type of equipment is situated below deck, and the cable lifters are mounted horizontally (Jiang 2014).

Due to the limited number of words, this article only briefly introduces the main auxiliary machinery on board. For more detailed introduction, please refer to *Marine Auxiliary Machinery* written by McGeorg (1995).

Ship Machinery Design Requirements

1. Principal requirements

Economical operation: The overall economical performance of the power plant should be designed to be as better as possible, and the operating cost of the entire ship should be as little as possible.

Reliability: The selection of equipment and the design of systems should be highly reliable, with necessary backup or redundancy.

Manipulability: In order to obtain convenient operation, the use of equipment and systems should be as simple as possible.

Maintainability: The structures of the equipment and the layouts of the piping system should be easy to maintain.

Economical construction: The cost should be as little as possible, and the construction should be simple.

Weight and size target: The weight of the ship's machinery should be as light as possible, and the space occupied by the machinery should be as small as possible.

Vibration and noise level: The vibration and noise should be as small as possible (CSSC 1999).

2. Specific requirements

Meet the requirements of the power plant stated in the mission book or specification book or contract book.

Meet the requirements of the overall performance of the ship on the power plant.

Meet the requirements of the classification society on the ship.

Meet the relevant regulatory requirements of the flag state.

Meet the requirements of the international conventions and regulations.

Meet the requirements of the waters and ports involved in the navigation.

Meet the system design requirements specified or recommended by the equipment manufacturers (CSSC 1999).

Main Propulsion Devices Design

On modern civil merchant ships, the most popular power plants are diesel engine ones, and a small part are steam turbine ones (on a few LNG ships). Gas turbine propulsion power devices, nuclear power propulsion power devices, and combined power propulsion devices are mainly used for military ships, currently not used on civil merchant ships (CSSC 1999). So we focus on the diesel engine ones when designing main propulsion plants.

Main Propulsion Devices Evaluation Indexes

Technical Indexes The rated power of the main engine required by unit DWT and unit ship speed P_a (kW/(t * kn)) is:

$$P_a = \frac{P_{me}}{DWT * V_s}$$

where P_{me} is the rated power of the engine (kW), DWT denotes the ship load capacity (t), and V_s is the ship service speed (kn).

The weight of unit main engine power g_{me} (t/kW) is:

$$g_{me} = \frac{G_{me}}{P_{me}}$$

where G_{me} is the mass of main engine (t) and P_{me} is the rated power of the engine (kW).

The weight of unit power plant power g_{pp} (t/kW) is:

$$g_{pp} = \frac{G_{pp}}{P_{me}}$$

where G_{pp} is the total weight of power plant (including main engine, auxiliary machinery, shaft system, boiler, piping system, etc.).

The relative weight of the main engine α_{me} is given as:

$$\alpha_{me} = \frac{G_{me}}{DWT}$$

The relative weight of the power plant α_{pp} is written as:

$$\alpha_{pp} = \frac{G_{pp}}{DWT}$$

Area saturation K_{er} (kW/m²) is defined as:

$$K_{er} = \frac{P_{me}}{S_{er}}$$

where S_{er} is the area occupied by the engine room (m²).

Daily fuel consumption of the main engine G_{FOC} (t/d) is given as follows:

$$G_{FOC} = 24 * 10^{-3} * P_{CSR} * g_{CSR}$$

where P_{CSR} is the power of main engine at CSR (continuous service rating) (kW) and g_{CSR} is the fuel consumption rate of main engine at CSR (kg/kW*h).

Daily fuel consumption of the ship $G_D(t/d)$ is:

$$G_D = G_{\text{FOC}} + G_{\text{Gd}} + G_{\text{ab}}$$

where G_{Gd} is the daily fuel consumption of generators (t/d) and G_{ab} is the daily fuel consumption of boilers (t/d).

Total fuel consumption per nautical mile $G_n(t/n \text{ mile})$ is:

$$G_n = \frac{G_D}{V_s}$$

where V_s is the ship speed (kn).

Performance indexes Maneuvering ability: the ability of the power plant to quickly change operating conditions, which is usually indicated by the length of time for starting, acceleration, reverse, and parking.

Reliability: the ability of the power plant to work normally and stably during the specified maintenance, repair, use conditions, and periods. Redundancy theory and techniques should be used to maintain the propulsion and power supply of a power plant when failures or damages occur in one or more components of its equipment.

Automation degree: The level of automation and remote control of mechanical and electrical equipment in the engine room. It is generally divided into two categories: manned engine room and unmanned engine room.

Control of noise and prevention of harmful vibrations: the ship's power plant must meet the control criteria of the vibration and noise (CSSC 1999).

Selection of Main Engine

Selection of a suitable main engine to meet the power demands of a given project involves proper tuning in respect to load range and the influence of operating conditions which are likely to prevail throughout the entire life of the ship (Wärtsilä NSD Switzerland Ltd. 1998).

Every engine has a layout field within which the power/speed ratio can be selected. It is limited by envelopes defining the area where 100% firing pressure (i.e., nominal maximum pressure) is

available for the selection of the contract maximum continuous rating (CMCR). Contrary to the "layout field," the "load range" is the admissible area of operation once the CMCR has been determined (Wärtsilä NSD Switzerland Ltd 1998).

In order to define the required CMCR, various parameters such as propulsive power, propeller efficiency, operational flexibility, power and speed margins, possibility of a main-engine driven generator, and the ship's trading patterns need to be considered (Wärtsilä NSD Switzerland Ltd 1998).

Selecting the most suitable engine is vital to achieving an efficient cost/benefit response to a specific transport requirement (Wärtsilä NSD Switzerland Ltd 1998).

Transmission System Design

The transmission system transmits power from the engine to the propeller. It is made up of shafts, bearings, and the propeller itself. The thrust from the propeller is transferred to the ship through the transmission system (Zhao 2012).

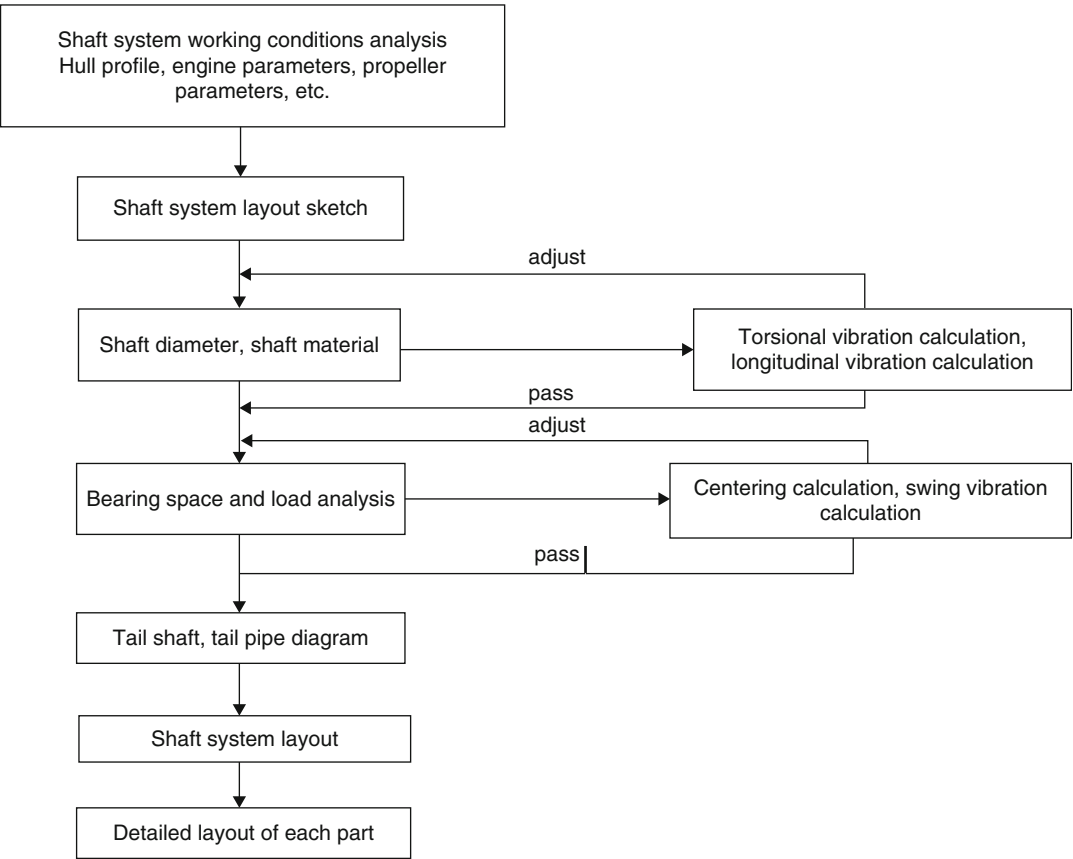
The different components in the system include the thrust shaft, the intermediate shafts, and the tail shafts. These shafts are supported by the thrust block, intermediate bearings, and the stern tube bearing. A sealing arrangement is provided at both ends of the tail shaft with the propeller and cone completing the arrangement (Zhao 2012).

The task of the shaft system design is to determine the position of the propulsion shaft and the size and material of each part. Shaft system design is a process that requires repeated adjustments. The steps of the shaft system design are shown in (Fig. 2).

According to the requirements of the classification society, the basic diameter of the shaft is calculated by the following formula:

$$d = F * C * \sqrt[3]{\frac{P_e * \left(\frac{560}{\sigma_b + 160}\right)}{n_e}}$$

where d is the basic diameter of the shaft (mm), F is the propeller coefficient, P_e is the shaft transmission rated power (kW), n_e is the rated speed (r/min), σ_b is shaft material tensile strength



Ship Machinery Design, Fig. 2 Shaft system design steps. (CSSC 1999)

(N/mm²), and C is the design characteristic coefficient.

If the shaft is hollow with a hole diameter $d_0 > 0.4d$, the shaft diameter should be corrected according to the following formula:

$$d_c = d * \sqrt[3]{\frac{1}{1 - \left(\frac{d_0}{d_a}\right)^4}}$$

where d_a is the actual outside diameter of the shaft.

Intermediate shafts, tail shafts, and propeller shafts in ship shaft system are generally forged from high-quality carbon steel. Shaft couplings should also be forged from high-quality carbon steel. In some cases they can also be made of ductile iron.

The torque transmitted by the shaft system can be calculated based on the power it transmits and the shaft rotation speed:

$$M = \frac{9550P}{n}$$

in which M is the shaft transmission torque (N • mm), P is the shaft transmission power (kW), and n is shaft speed (r/min).

The thrust transmitted by the shaft system can be calculated based on the propeller power absorbed, propeller efficiency, and ship speed:

$$T = 1.94 \times \frac{P_p}{v} \times \eta_p$$

where T is the thrust transmitted by the shaft system (kN), P_p is the propeller power absorbed

(kW), v is ship speed (kn), and η_p is the propeller efficiency (CSSC 1999).

Propeller Design

The propeller consists of a boss with several blades of helicoidal form attached to it. It screws through the water in much the same way as a bolt screws through its nut and thus converts the engine torque into a direct thrust to push the ship ahead. The thrust is transmitted along the shafting to the thrust block and finally to the ship structure (Li 2013).

A propeller which turns clockwise when viewed from aft is considered right-handed, and most single screw ships have right-handed propellers. A twin screw ship will usually have a right-handed starboard propeller and a left-handed port propeller.

A solid fixed pitch propeller is an ordinary one. Although usually described as fixed pitch, the pitch does vary with increasing radius from the boss. The pitch at any point is however fixed, and for calculation purposes, a mean or average value is used (Li 2013).

A controllable pitch propeller is made up of a boss with separate blades mounted into it. An internal mechanism enables the blades to be moved simultaneously through an arc to change the pitch angle and the pitch (Li 2013).

The Match of Ship, Engine, and Propeller

When a diesel engine is used as a ship's main engine, the diesel engine will work with a propeller. For a given diesel engine, an appropriate propeller must be fitted. Correct match between the diesel engine and the propeller should not only make full use of the diesel engine power but also make this power in the allowable range within the entire operating speed range of the engine. In this way, the diesel engine can fully utilize, and at the same time, high economic efficiency, reliability, long service life, and low vibration of the engine can be obtained (Lv 2014).

Auxiliary Machinery Design

Ship's auxiliary machinery and its associated piping system are very complex. For different types of ships, their auxiliary machinery can be

arranged and designed with flexibility. The following are the basic principles to be followed in the design of all marine auxiliary machinery:

1. The weight of auxiliary machinery arranged on the left and right sides of the engine room should be kept as balanced as possible. At the same time, in order to increase the stability of the ship, the arrangement of auxiliary machinery should try to lower the height of the center of gravity of the ship.
2. The layout of the auxiliary machinery should enable the crews to manage, check, and maintain the equipment easily, and reasonable personnel access and maintenance space should be taken into consideration by designers.
3. The layout of the auxiliary machinery should reduce the space occupied by the engine room as much as possible to increase the ship's loading capacity and reduce the cost of shipbuilding.
4. Various safety measures should be fully considered, especially the safety of marine engineers.
5. The relative positions of various auxiliary machinery should be reasonable and do not interfere with each other during normal operation.
6. Ship's auxiliary machinery should ensure reliability and continuous operation of the main propulsion devices (CSSC 1999).

Key Applications

The following example explains the main principles in matching a MAN diesel engine with an appropriate propeller (Refer to www.mandieselturbo.com).

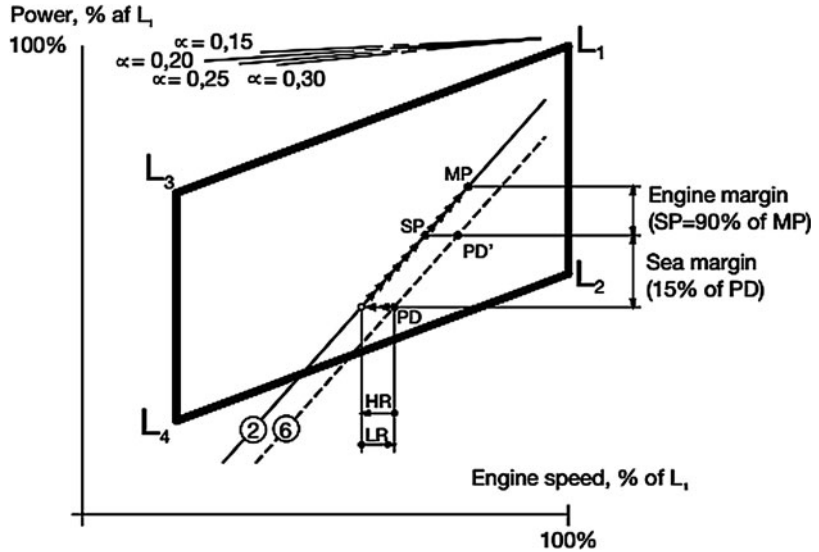
Propulsion and Engine Running Points

For a fixed pitch propeller, the relation between power and propeller speed is:

$$P = c * n^3$$

in which P = engine power for propulsion, n = propeller speed, and c = constant. Normally, estimates of the necessary propeller power and speed are based on theoretical calculations and

Ship Machinery Design,
Fig. 3 Ship propulsion
 running points and engine
 layout



experimental tank tests with assuming optimum operating conditions. The combination of speed and power obtained may be called the ship's propeller design point (PD), placed on the light running propeller curve 6 (Fig. 3). Some shipyards and propeller manufacturers sometimes use a propeller design point (PD) that incorporates all or part of the so-called sea margin described below.

The notations in Fig. 3 are defined as follows:

- Line 2: Propulsion curve, fouled hull, and heavy weather (heavy running), recommended for engine layout
- Line 6: Propulsion curve, clean hull, and calm weather (light running), for propeller layout
- MP: Specified MCR for propulsion
- SP: Continuous service rating for propulsion
- PD: Propeller design point
- HR: Heavy running
- LR: Light running

When the ship has sailed for some time, the hull and propeller become fouled, and the hull's resistance will increase. Consequently, the ship's speed will be reduced unless the engine delivers more power to the propeller, i.e., the propeller will be further loaded and will be heavy running (HR).

As modern vessels with a relatively high service speed are prepared with very smooth

propeller and hull surfaces, the gradual fouling after sea trial will increase the hull's resistance and make the propeller heavier running.

When the weather is bad, the ship's resistance may increase compared to operating in calm weather conditions. When determining the necessary engine power, it is a normal practice to add an extra power margin, the so-called sea margin, which is traditionally about 15% of the propeller design (PD) power.

When determining the necessary engine layout speed that considers the influence of a heavy running propeller for operating at high extra ship resistance, it is (compared to line 6) recommended to choose a heavier propeller (line 2). The propeller curve for clean hull and calm weather is line 6 representing a "light running" (LR) propeller.

Compared to the heavy engine layout line 2, we recommend using a light running of 3.0–7.0% for design of the propeller.

Besides the sea margin, a so-called engine margin of some 10% or 15% is frequently added. The corresponding point is called the "specified MCR for propulsion" (MP) and refers to the fact that the power for point SP is 10% or 15% lower than that for point MP.

Point MP is identical to the engine's specified MCR point (M) unless a main engine driven shaft generator is installed. In such a case, the extra

power demand of the shaft generator must also be considered.

The constant ship speed lines α are shown at the top of the figure. They indicate that the power required at various propeller speeds keep the same ship speed. It is assumed that the optimum propeller diameter is used for each ship speed.

Propeller Diameter and Pitch, Influence on the Optimum Propeller Speed

In general, the larger the propeller diameter D is, the lower optimum propeller speed will be obtained. The kW required for a certain design draught and ship speed can be seen as curve D in the figure below.

The maximum possible propeller diameter depends on the given design draught of the ship and the clearance needed between the propeller and the aft body hull and the keel.

The example shown in Fig. 4 is an 80,000 dwt crude oil tank with a design draught of 12.2 m and a design speed of 14.5 knots.

When the optimum propeller diameter D is increased from 6.6 to 7.2 m, the power demand is reduced from about 9,290 to 8,820 kW, and the optimum propeller speed is reduced from 120 to

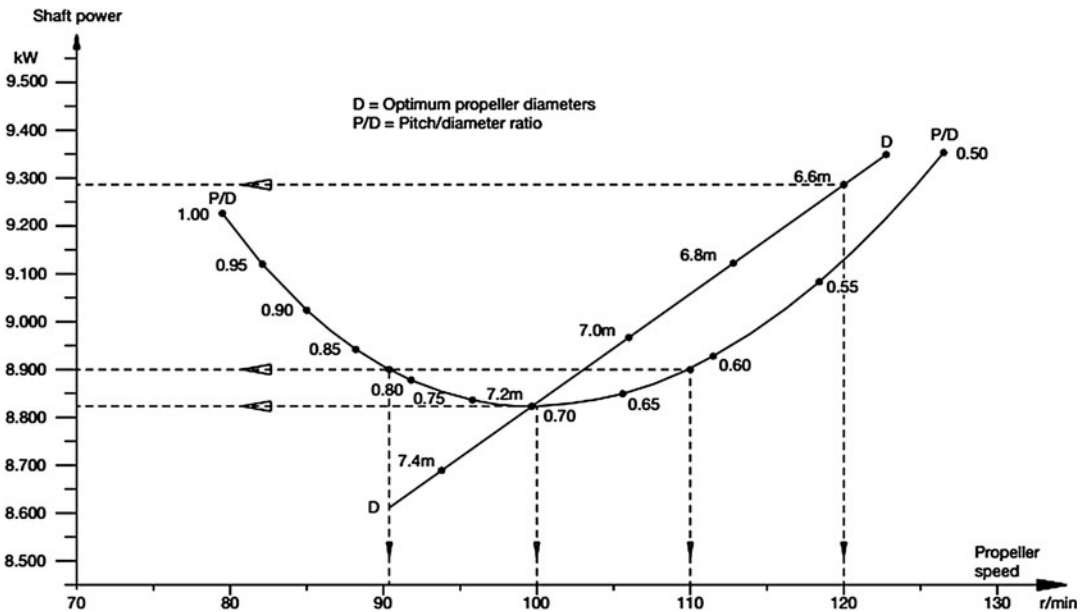
100 r/min, corresponding to the constant ship speed coefficient $\alpha = 0.28$.

Once an optimum propeller diameter of maximum 7.2 m has been chosen, the corresponding optimum pitch in this point is given for the design speed of 14.5 knots, i.e., $P/D = 0.70$.

However, if the optimum propeller speed of 100 r/min does not suit the preferred/selected main engine speed, a change of pitch away from optimum will only cause a relatively small extra power demand, as shown in Fig. 4, keeping the same maximum propeller diameter:

- (Case 1) Going from 100 to 110 r/min ($P/D = 0.62$) requires 8,900 kW, i.e., an extra power demand of 80 kW.
- (Case 2) Going from 100 to 91 r/min ($P/D = 0.81$) requires 8,900 kW, i.e., an extra power demand of 80 kW.

In both cases the extra power demand is only of 0.9%, and the corresponding “equal speed curves” are $\alpha = +0.1$ and $\alpha = -0.1$, respectively, so there is a certain interval of propeller speeds in which the “power penalty” is very limited (MAN B&W Germany Ltd 2010).



Ship Machinery Design, Fig. 4 Influence of diameter and pitch on propeller design. (MAN B&W Germany Ltd 2010)

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Ship Machinery Maintenance

- [Ship Construction and Operation Impact on Energy Efficiency](#)

Ship Motion Control

- [Thruster-Assisted Mooring](#)

Ship Navigation

- [Big Data-Based Decision Support Systems](#)

Ship Navigation Sensor

- [Big Data-Based Decision Support Systems](#)

Ship Navigation System

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Synonyms

[AIS \(automatic identification system\)](#); [Bridge system](#); [Inertial navigation system](#); [Radar navigation system](#); [Radio navigation system](#)

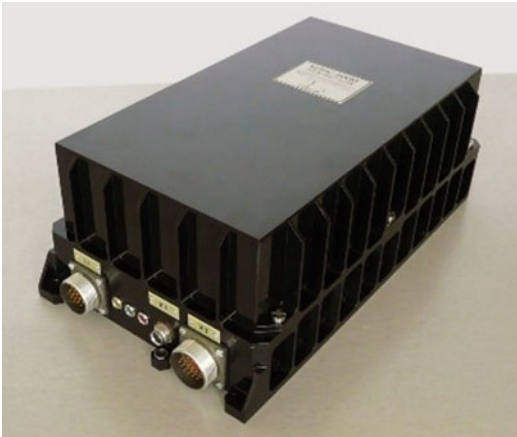
Definition

The shipping navigation system is utilized to assist with the ship navigation; it helps to access precise data about ship performance and surrounding environmental information which may be critical for safe navigation. With the continuous development of marine technology, there are several navigation systems commonly used in ships, such as inertial navigation system, radio navigation system, integrated navigation systems, etc.

Inertial Navigation System

An inertial navigation system (INS) is a self-contained reference system based on inertial measurement provided by accelerometers and gyroscopes (IMO 2007), which used to estimate precise position, velocity, heading, pitch, and roll of a moving ship without the need for external references by applying Newton's mechanics laws in dead-reckoning process. This kind of navigation system is not only used in ship but aircraft, submarine as well.

An example of the inertial navigation system is shown in Fig. 1. The basic components of such a system include a computer, servo systems, core instruments – accelerometers and gyroscopes – and other motion-sensing devices. The initial position, heading, velocity, etc. are set into the



Ship Navigation System, Fig. 1 An example of the inertial navigation system

system from another source of human operator or GPS (Global Positioning System) satellite receiver and thereafter compute the updated position and velocity by processing information received from the motion sensors. Accelerometers detect and measure linear acceleration along the axis in which they lie. For constructing a triaxial inertial platform, three accelerometers are needed, one in each of three perpendicular planes: north-south; east-west; and vertical. Any movement of ships in any plane is continuously sensed by these accelerometers, and the output is sent to the computer to be integrated over time, by which the velocity can be calculated. Then the distance from departure can be worked out by integrating the velocity calculated; thus the actual ship position can be found. Three gyroscopes or a three-degree-of-freedom gyrocompass are maintained to measure rotational motion. Whatever rotational motion happens, the computer generates a torque signal to return this three-gyro-stabilized platform to horizontal position. Therefore, any roll, pitch, and yaw can be sensed, since it is always in the horizontal position and the angle between it and the ship can be used to judge roll, pitch, or heading. Each gyro output signal is amplified through the amplifier to operate servo system motor, which in turn keep the platform stabilized parallel to the earth surface and in the same direction.

As mentioned before, the inertial navigation system is entirely self-contained which means

once the initial information is set, it does not require outside information and radiate energy (IMO 1998). Thus, it shows the security advantage over other types of navigation systems such as radio navigation system for it cannot be detected. Also, it is able to be operated under harsh weather conditions, and data provided has high accuracy. However, the inertial navigation system still has several constraints. Firstly, since INS contains many precision mechanical parts, the entire cost is high. Secondly, it is noteworthy that initial alignment is always necessary which would cost much time. Lastly, the biggest disadvantage is integration drift. Specifically, the errors generated from the measurement of acceleration would gradually become bigger during integration procedure. Due to the new position needs to be calculated from the previous calculated one, the errors will accumulate with time, although these accumulating error characteristics are able to be known and modelled quite well. Therefore, these errors can be fixed from some other types of navigation systems such as satellite navigation system.

Radio Navigation System

The radio navigation system can be used to detect ship's position and guide the ship to sail along the scheduled route using radio technology. Radio waves can travel through the uniform medium in the straight line with the constant speed or be reflected by interface of two different mediums, which is the basic theory about how the system works. According to the propagation characteristics of radio waves, radio wave parameters including frequency, amplitude, propagation time, and phase launched by radio stations can be obtained by ships to calculate geometric parameters of the ship relative to radio stations such as distance and direction, which realizes the function of position and navigation. Since radio waves can travel very long distances, it makes a particularly good navigation system for ships. The common radio systems used worldwide for ships are radio direction finder, OMEGA and LORAN. However, they are too old to be used by modern ships. Thus, radar

navigation system and satellite navigation system have been developed and widely used by current ships.

Radar Navigation System

Radar navigation system is a type of radio navigation using radar for detection and ranging. Shipborne radars are electronic equipment which can use a rotating antenna to produce pulses of high-frequency electromagnetic waves around the water surface surrounding the ship to the horizon, listening for the waves reflected from objects to detect targets and thence displaying a picture of the ship's surroundings on a screen with cathode-ray tube (picture tube). The beam sent by the transmitter is narrow in the horizontal plane but wide in the vertical plane, which would make more objects caught when ship rolls. The shipborne radar can be classified under the X-band (9320–9500 MHz) or S-band (3000–3247 MHz) frequencies. If the size of antenna is constant, the X-band, being of higher frequency, is used for a crisper image and better resolution which reveals advantages in close-range detect, while the S-band is used especially when in rain or fog as well as for long-range detect since the higher sensitivity. Generally, ship with large size will be fitted with both X- and S-band radars, whereas smaller one will only take the X-band radar (Ludeno et al. 2013). A radar antenna mounted on a ship is generally on the high place, ensuring better transmitting and receiving ability. The strength of echo is related to the reflectivity of the surface of the reflecting object and the distance the waves travel. In most situations, the echo strength reflected from water is quite weak unless sea condition is very rough, while there will be strong reflection from ships, cliffs, and offshore constructions. By using radars, the mariners are able to maneuver ships irrespective of visible conditions, and it is, therefore, safe for ships to navigate even in thick fog.

The radar display screen is a vital part of the system, while a typical screen display is shown in Fig. 2. On the screen, ship position is located in the center. The scanning line is a radial straight



Ship Navigation System, Fig. 2 An example of the radar navigation system screen display

line drawn from the center outward, rotating synchronously with the transmitter. When electromagnetic waves that bounced off objects return to receiver, the image on the picture tube brightens (Ma et al. 2015). The positions of any reflecting objects can be seen on the screen, allowing the mariners to analyze the situation from the image.

In marine navigation, the most crucial function of the radar navigation system is avoiding collision, and with the evolution of technology, the ARPA (automatic radar plotting aid) has been used on today's ships. ARPA's introduction results from rapid advances in computer technology which can improve the conventional radar system. Plotting the targets also can be processed automatically instead of convention radar's manual plotting, tracking targets' bearing, speed, and closest point of approach (CPA), thereby calculating time of closest point of approach (TCPA). If the targets are under risk of collision or are navigating in a risky manner, dangerous alarms will be given to operators to avoid collision with ships or other offshore constructions. This intelligent system can alleviate workload of the ship's officers on the bridge.

However, radar and ARPA have limitations which influence their reliability. Radar sometimes may fail to detect small boats, floating ice, or other objects. Blind or shadow sector of the radar is caused by obstructions near the antenna such as masts and other structure or objects on the ship. Bearing discrimination usually happens when

radars need to distinguish between two small targets of same range but slightly different bearings, and they may be reported as the same. False echoes such as double echoes and multiple echoes are always existed resulting in detection errors.

Satellite Navigation System

The satellite navigation system is a particular type of radio navigation system. Unlike conventional ground-based radio navigation, the radio stations are moved to satellites on the ultimate high ground of space. Without concern about limitation of effective range, they are able to be “visible” from anywhere on Earth, so-called global navigation satellite system (GNSS). Such wide spatial converge is the most significant advantage of satellite navigation system among others. There are four main GNSS systems operating: the United States’ Global Positioning System (GPS), Russia’s Global Orbiting Navigation Satellite System (GLONASS), European Union’s Galileo, and China’s BeiDou Navigation Satellite System (BDS). Among them, GPS is the most used satellite navigation system worldwide.

The satellite-based navigation system can be used to calculate ship’s position by integrating information from a network of satellites. Every satellite involved in satellite network emits coded signals at precise intervals to receivers equipped on ships, and then receivers decode the signal information into position, distance between user satellite, and time estimates. This information can be utilized to measure the accurate transmitting satellite’s position and the distance between it and the receiver (production of the transmission delay time and speed of light). Calculating signal times of arrival from four or more satellites enables the receiver to determine its position.

In addition to position, the satellite navigation system can also plan a route for ships. The system is able to store thousands of waypoints. When choosing a waypoint, the distance from the current position, the bearing, as well as the time needed to arrive it at present speed can automatically be obtained. The hydrographic information and coastal features of the waypoints can be easily

found which can be the reference of waypoints selection.

The satellite navigation system has better accuracy and wider converge than previous land-based radio navigation system. Because of these advantages, the satellite navigation system has become the most mainstream navigation system in marine navigation, although it still has disadvantages. The operation of the satellite navigation system entirely relies on receiving satellite signals by radio, which means noise and interference may result in the failure. Moreover, sometimes satellite signals may not be precise owing to extreme atmospheric conditions such as geomagnetic storms.

Automatic Identification System

The automatic identification system (AIS) is a navigation system using shipborne transponders which operate in the VHF mobile maritime band to transmit information about the ship to other ships and to coastal base station automatically, contributing to reduce the risk of ship collisions and bolster the security of ports (Habtemariam et al. 2015). As a kind of self-reporting system, the data provided by AIS consists of both static data such as ship name, call sign, registration number, MMSI (Maritime Mobile Service Identity), and dynamic data such as position, heading, and speed, which can be learned on a display screen, all on a real-time, utterly automated basis. In addition to help masters to know about information of ships in the vicinity, AIS also can be used by maritime authorities to identify, locate, and monitor ships. The information of position, heading, and speed is mainly derived from GPS, while other information broadcast by the AIS is from shipboard electronic equipment and sensors, and all information is able to be uploaded automatically. According to the rules from IMO, AIS is mandatory for ships of 300 gross tons and upwards in international navigation and 500 tons and upwards for cargoes not in international waters and all passenger ships. A typical example of AIS is shown in Fig. 3.



Ship Navigation System, Fig. 3 An example of the automatic identification system

The AIS coverage range basically is dependent on the height of the antenna of both ships and base stations, and this range is limited to VHF range, around 20 nautical miles. To fix the problem, recently, it has become possible to receive AIS information by satellite in low earth orbit which are equipped with receivers. Such satellite-based AIS is enabled to monitor the location of all AIS-equipped ships in the most remote areas that may be beyond the reach of terrestrial-only AIS systems.

As a navigation aid, AIS is widely applied in collision avoidance and ship surveillance. For collision avoidance, AIS can track and monitor targets, while CPA and TCPA can be calculated automatically based on information of ships' movements, and alarms will be sent out if ships may encounter potential hazard. The system can improve situational awareness and be vital for navigators' decision-making process of collision avoidance (Perera and Soares Guedes 2015). As a surveillance tool, AIS enables shore side authorities to detect specific ships and their movements of vessels through the area. Shore side stations can also use the AIS channels to transmit current information on tides and locate weather conditions.

However, AIS still has several limitations. Firstly, since the information received by AIS is from other external sensors, thus the precision of AIS information depends on the accuracy of other

sensors. Secondly, not all ships have fitted with AIS, and the AIS transponder could be intentionally turned off. Thirdly, security of the ships will be challenged if the information transmitted is used illegally, especially in areas where piracy is rife.

Integrated Navigation System and Bridge System

As advances in sensor technology, diverse systems are being developed independently. However, due to the rapid development of modern marine technology, ships tend to be huge-sized, high speed, and professional, thereby single navigation system is unable to meet the needs of current navigation. The integrated navigation system (INS) is the combination of several navigation systems, using computer technology to integrate the data and information from various systems (IMO 1998). Generally, the INS consists of radar/ARPA, ECDIS, GPS, AIS, conning, gyrocompasses, antennas, and other sensors, by which the information of uninterrupted position, speed, heading, and further navigation safety data can be obtained. The system also contains capability of generating alarms and indications when the ship is in proximity to dangers. The motivation of INS's introduction is to promote the ship navigation safety through supplying integrated and augmented information and functions to assist navigators in geographic, traffic, and environmental hazards avoidance.

ECDIS refers to Electronic Chart Display and Information System. The introduction of INS, at large content, owes to the advancement of ECDIS, which is also the core of the system. ECDIS is developed to displace conventional paper nautical chart, which is also the requirements from SOLAS, aiming to ease the mariner's workload and enhance the navigation safety. It possesses automatic capabilities of route planning, ship monitoring, and automatic ETA (estimated time of arrival) calculation with the input of other systems such as radar, GPS, depth sounders, and gyrocompasses. Two widely used types of ECDIS charts are raster chart and vector chart.

Raster charts are raster graphics charts which are generated by transmitting paper charts to digital image through an accurate, detailed scanning process. Raster charts, therefore, are the direct copy of paper charts. This kind of chart only grows larger or smaller as per the zooming, and while the chart is rotated, everything on the chart rotates. Vector charts generated by computers are graphic representations of databases, consisting of digitized data necessary for safe navigation, such as coastlines, bathymetry, buoys, etc. The navigation systems using vector charts can be programmed to send out warns of impending danger using those data and information. All of the same information that is available on a raster chart can be obtained on a vector chart, moreover, with some useful additional data. Thus, the usage of vector charts is preferable to raster charts. According to IMO (1998), raster charts are permitted to be used in the absence of vector charts.

The latest performance standards for INS are detailed in the IMO (2007) and apply to all newbuildings since 1st January 2011 where INS are fitted. The standards told that an INS is required to integrate the tasks of collision avoidance, ship monitoring, route monitoring, route planning, navigation control data status display, and a centralized human-machine interface for alert management on multifunctional displays. Moreover, alert management is one of the most critical parts of the INS, enhancing navigation safety. Every alert or abnormal condition is distinguished between the three priorities by the INS, on the basis of urgency, as an alarm, a warning, or a caution. Each of the three types is reported diversely to navigator by sending out visual and audible signals and displaying messages. Alarms which may represent highest urgency are usually must include danger of collision and grounding. ECDIS also provides a critical integration platform for multi-sensor fusion, which is significant for INS.

According to INS performance standards, INS can be classified into three categories, as INS(A), INS(B), and INS(C). INS(A) is required to at least provide information of ship position, speed, course, and time. Also, alarm signal needs to be emitted in the situation of any sensor failure. For

INS(B), in addition to possess functions of INS (A), it is required to provide relative information for hazard avoidance. This is done by marking information of position, speed, course, depth, and other potential hazards on ECDIS real time and autonomously. INS(C), which has functions of INS(B), need additionally implement automatic navigation control functions as well as ship performance monitoring.

Since the multisource data need to be integrated by INS, the information fusion technology reveals the significance for constructing this centralized fusion system. The information from various sources (homogeny, heterogenic) will be preprocessed, linked, and refined and subsequently obtain a newly higher quality. For instance, AIS and radar navigation system are two representative navigation systems with their own limitations, and these limitations can be made up by a level of information fusion. Presenting such integrated information from AIS and radar on ECDIS is of paramount importance for improving detection precise and reducing navigation risk. Generally, there are two main types of approaches to fuse information- decentralized and centralized approaches. For the former, sophisticated data is initially processed in each sensor and subsequently fused using a proper algorithm. For the latter, default data from each sensor, after a preprocess, is transmitted to a fusion center and then processed together also using the appropriate algorithm. The widely used algorithms for information fusion are weighted average method, Kalman filter, Bayesian estimation and neural network, etc.

Integrated bridge system (IBS) as shown in Fig. 4 is defined as a navigation management system which developed from integrated navigation system (INS). IBS can be considered as a combination of systems, which are interconnected to provide centralized access to navigational, control, propulsion, and monitoring information, aiming to enhance navigation safety and promote efficiency of ship navigational capabilities. For today's IBS technology, it may consist of autopilot, gyro, ECDIS, AIS, dynamic positioning system, power distribution system, meteorological instruments, conning display, and GMDSS



Ship Navigation System, Fig. 4 An example of the integrated bridge system

(Global Maritime Distress and Safety System), and all systems or instruments interact in a complete network which can be divided into two sub-networks, namely, navigation system and automation system. The navigation system is responsible for navigation safety, functioning like INS, while the automation system is in charge of power management and propulsion control as well as additional task such as bilge and ballast control, alarm, and monitoring systems. Conning displays of IBS can present navigation information such as ship position, speed, course, and rudder and propulsion data such as propeller situation, thruster, power, and states of wind and wave to operators and assist their decision-making.

Cross-References

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Integrated Navigation](#)

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Ship Operational Environment

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Synonyms

[Sea environment](#); [Significant wave height](#); [Spatiotemporal model](#); [Wind speed](#)

Definition

Stricter environmental regulations on shipping, the maritime community is eagerly seeking for solutions to increase a ship's energy efficiency. Furthermore, the climate targets set up by authorities, such as in EU and some western countries, require that at least 10% of the transport fuels should come from renewable sources by 2020. This climate ambition triggered intensive research and innovation activities to promote auxiliary propulsion using renewable energy such as wind and waves and to develop energy efficiency measures. Any maritime-related research and development activities must rely on the correct description of Meteorological and Oceanographic (MetOcean) environment (wind, wave, etc.) that a marine structure may encounter. Therefore, large instruments, such as satellite, climate ocean models, buoys, and even expensive ship onboard measurement devices, are launched to measure the safety and energy performance of marine systems and their surrounding sea environments.

Weather conditions at sea vary in space and time resulting in variable environmental loads acting on ship structures. The load variability can cause material deterioration and may lead to mechanical/structural failures or breakdowns in extreme sea conditions. Thus, accurate models describing the variability of MetOcean conditions are essential for safe maritime operations. Statistics and probability theory are mathematical tools to derive useful models describing the variability of sea environments a ship encounters. Hierarchical models can be used to represent variability at different time scales, hours, days, seasons, and years, while in spatial scales ranging from a few to 1000 kilometers.

The detailed spatial-temporal environmental models for short- and long-term descriptions of the variation and correlation of sea environment give tools to generate the weather conditions for the whole service time of ship. These conditions can be used to estimate a ship's energy performance and efficiency for the design, validation, and implementation of energy efficiency measures in ships (such as wing sails, voyage optimization systems). Additionally, those models give

means to estimate risks for system failures and methods to calculate payback time of the energy efficient/harvest measures.

The Sea Surface Elevation and Wind Gusts

In the finest scale, environmental loads applied on ships and generated by waves and wind gusts are stationary and homogeneous. In order to estimate the loads, one needs dedicated software to evaluate responses of structures sea surface and wind pressure variability. The Gaussian distribution is often used to describe the sea surface and wind variability. (In shallow water locations or in extreme seas, nonlinear quadratic corrections are added to the Gaussian model.) The Gaussian models are very convenient since the stationary and homogenous sea surface and winds are then uniquely described by their average values and the power spectral densities (equivalently variances and correlations) (Lindgren 2013). Tools to find various load characteristics (wave-length, period amplitudes, etc.) are developed (Brodtkorb et al. 2000; Podgórski et al. 2000).

Several parametric models were proposed for the sea and wind spectra, e.g., Pierson Moskowitz, Jonswap sea spectra, and Davenport, Harris, and Kaimal spectra for gust wind variability. For example, the Pierson-Moskowitz spectrum is often used to model fully developed sea as by

$$S(\omega) = c\omega^{-5} \exp\left(-1.25\omega_p^4/\omega^4\right), \quad (1)$$

$$c = 0.3125H_s^2\omega_p^4, \quad \omega_p = 2\pi/T_p.$$

The important parameters describing an ocean wave spectrum are integral of the spectrum, i.e., signals variance equal to $H_s^2/16$ and the peak frequencies T_p (or wave number). For wind spectrum description, it is variance and an average wind speed W . Spectral parameters also called environmental parameters define sea conditions. Typical power spectral densities $S(\omega)$ and means m of sea surface elevation (wind speeds) define Gaussian models, which describe short-term variability of a sea surface and wind velocity in time

and/or space. For example, given a spectrum $S(\omega)$, a time series $X(t)$ can be generated using the spectral representation, viz.,

$$X(t) \approx m + \sum_i \sqrt{S(\omega_i) \Delta \omega} R_i \cos(\omega_i t + e_i),$$

where $\Delta \omega = \omega_{i+1} - \omega_i$; R_i , e_i are independent Rayleigh and uniformly distributed variables, respectively (see Brodtkorb et al. (2000) and Lindgren (2013) for more detailed presentation).

The spatiotemporal models can be used to describe environmental parameters H_s , T_p , and W in larger scales when the condition of stationarity and homogeneity of sea conditions cannot be assumed. Since the parameters changes relatively slowly, they could be modeled as correlated random variables. This section will introduce random models describing the variability of environmental parameters in geographical locations and with seasons, i.e., spatiotemporal models of $H_s(t, x, y)$ and $W(t, x, y)$ where t has units days and x degree of longitude, while y degrees of latitude. The parameter $T_p(t, x, y)$ is at present modeled conditionally on $H_s(t, x, y)$ values.

Spatiotemporal Local Models for Significant Wave Height H_s

The variability of H_s is described by means of probability distribution. In the literature, the

H_s distribution is typically understood as the long-term distribution of significant wave height at some location or region. The long-term distribution can be interpreted as variability of H_s at a randomly taken time during a year. Often Weibull distribution fits well to the data. Limiting time span to a specific time window such as January month will affect the H_s distribution, because the variability of H_s depends on season. By shrinking the time span to a single moment t and geographical region to a location (x, y) , the distribution of $H_s(t, x, y)$ one obtains can be obtained. This will be the meaning of spatiotemporal distribution of H_s as used in this section. According to the study in Baxevari et al. (2005), for time period limited

to couple of weeks and regions of about 4 degrees in diameter, the distribution of H_s is log-normal, i.e., $\ln H_s(t, x, y)$ is normal with mean $m(t, x, y)$ and variance $\sigma^2(x, y)$, in Mediterranean and central North Atlantic. At fixed position denoted by $\mathbf{p}$, the temporal variability of mean $m_p(t)$ was modeled by means of annual cycle

$$\begin{aligned} m_p(t) &= \beta_0 + \beta_1 \cos(\phi t) + \beta_2 \sin(\phi t) \\ \phi &= 2\pi/365.2 \end{aligned} \quad (2)$$

time measured in days, and parameters β all depend on geographical location $\mathbf{p}$. Furthermore, it is found in Baxevari et al. (2005) that locally, i.e., in a neighborhood of location $\mathbf{p}$, the spatial correlation between the logarithms of significant wave heights with positions (x, y) away could be sufficiently accurately described by means of

$$\rho_p(x, y) = \exp\left(-(x^2 + y^2)/2L_p^2\right),$$

where L_p called correlation length depends on geographical location of the small region and with month. The $\ln H_s(x, y)$ has a two-dimensional spectrum

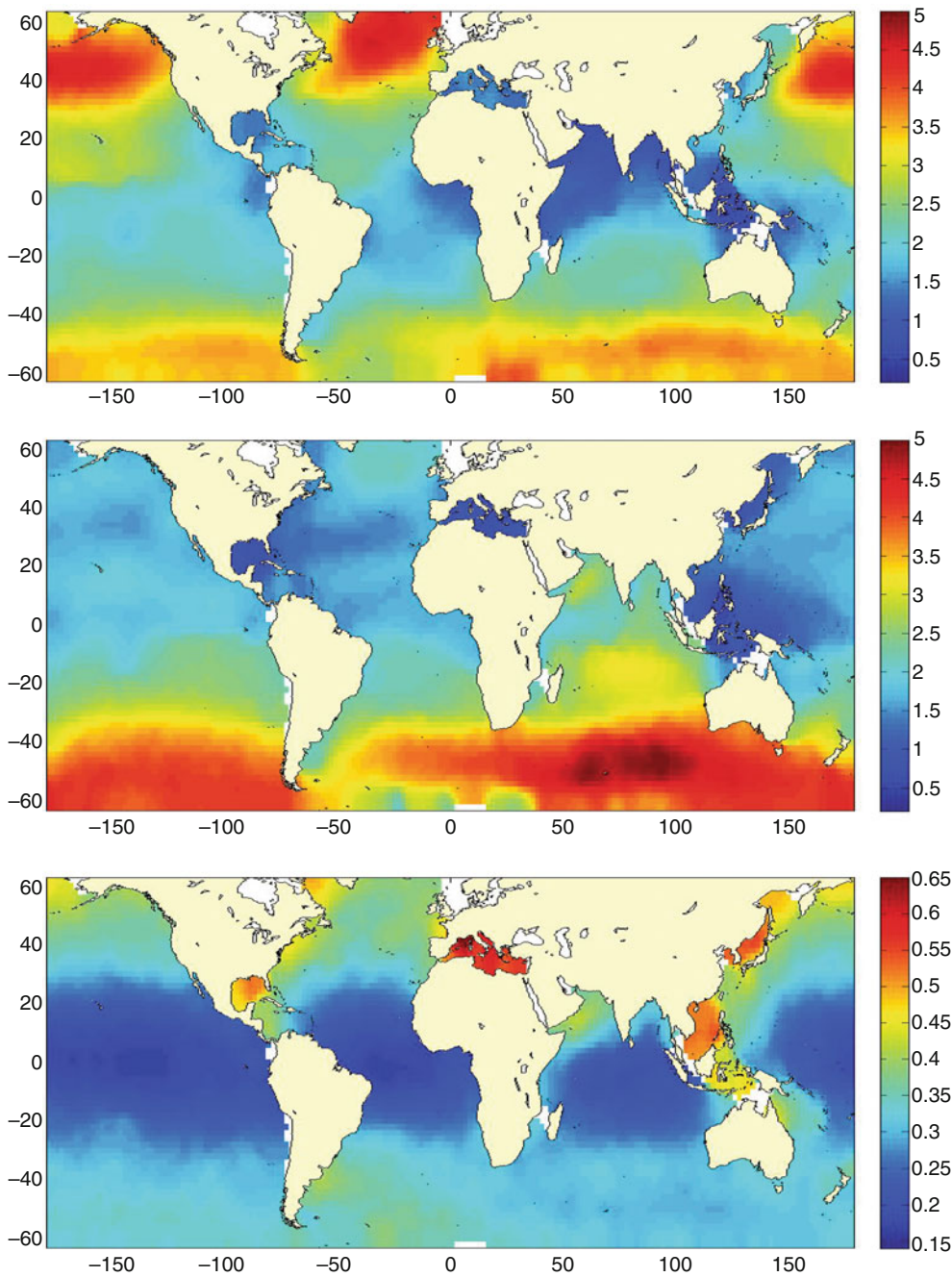
$$S_p(\omega_1, \omega_2) = \frac{\sigma_p^2 L_p^2}{2\pi} e^{-L_p^2 \frac{\omega_1^2 + \omega_2^2}{2}},$$

and the local spatial field in a neighborhood of a geographical location $\mathbf{p}$ can be approximated by

$$\begin{aligned} X(x, y) &= \ln H_s(x, y) \\ &\approx \Delta \omega \sum_{ij} \sqrt{S_p(\omega_i, \omega_j)} R_{ij} \cos(x\omega_i + y\omega_j + e_{ij}), \end{aligned} \quad (3)$$

where $\Delta \omega = \omega_{i+1} - \omega_i$, R_{ij} , e_{ij} are independent Rayleigh and uniformly distributed variables, respectively.

Note that (x, y) are in equivalent degrees so that degree in both longitude and latitude stands for the same distance when measured in kilometers. Finally, spatial models for the variability of the significant wave height on the world oceans can be accessed in Baxevari et al. (2008a). An



Ship Operational Environment, Fig. 1 Estimates of median significant wave height in February (top plot) and August (middle plot) and estimates of variance of $\ln H_s$ (bottom plot). Note the variance is independent of a season

example of the median value is presented in Fig. 1. The parameters β and σ^2 and 12 correlation lengths L have been estimated at 1861 different 4×4 degree regions.

Local H_s Dynamics

The variability of significant wave height in time and space has been studied in Baxevani et al. (2008b), where a local spatiotemporal $X(t, x,$

$y) = \ln H_s(t, x, y)$ proposed. Following the work carried out by Baxevari et al. (2003) and Longuet-Higgins (1957), a median velocity (v_x, v_y) is modeled by a moving random Gaussian field. A first-order autoregressive model, AR(1), was used to model “memory” of the field, leading to the following spatiotemporal correlation function

$$\rho(t, x, y) = \exp\left(-\left((x - v_x t)^2 + (y - v_y t)^2\right)/2L^2\right) \exp(-\lambda|t|). \quad (4)$$

This type of spatial-temporal covariance has been originally used in Cox and Isham (1988) and Gupta and Waymire (1987) for modeling rainfall.

The velocity field (v_x, v_y) was estimated using ERA40 data set, while the parameter λ was estimated using time series of buoy H_s observations. This was done by fitting the temporal correlation

$$\begin{aligned} \rho(t, 0, 0) &= \exp(-t^2/2T^2) \exp(-\lambda|t|), T \\ &= L/\|\mathbf{v}\|, \end{aligned} \quad (5)$$

to observed autocorrelation in measured time series by buoys. Here $\|\mathbf{v}\| = \sqrt{v_x^2 + v_y^2}$. The spatiotemporal field $X(t, x, y)$ can be simulated in a recursive way. For any $t = i \, dt$, $i > 0$, $X(0, x, y) = X_0(x, y)$,

$$\begin{aligned} X(t, x, y) &= \rho X(t - dt, x - v_x dt, y - v_y dt) \\ &+ \sqrt{1 - \rho^2} X_i(x, y), \end{aligned} \quad (6)$$

where $X_i(x, y)$ is independent simulation of the spatial field $X(x, y)$ by Eq. (3). Here $\rho = \exp(-\lambda \, dt)$.

Global Model

In addition, the local spatiotemporal fields have been further generalized to the global non-homogeneous and nonstationary fields in Baxevari et al. (2008b). This involved evaluation of a flow φ defined by the velocity fields. Clearly, $\mathbf{p} = \varphi(\mathbf{q}, t, s)$ is the position at time s of the point, and at time t , the position moves to q . Having the

flow the recursion is easily adopted to a non-homogeneous case (Baxevari et al. 2008b).

The model has been simplified in Podgórski and Rychlik (2014) by approximating the local correlation (5) as follows:

$$\begin{aligned} \rho_{\mathbf{p}}(t, 0, 0) &\approx \exp(-t^2/2\tau^2), \\ \tau &= \pi\left(-\lambda T^2 + \sqrt{\lambda^2 T^4 + T^2}\right), \end{aligned} \quad (7)$$

The parameter τ depends on location $\mathbf{p}$ and season.

Then the local spatiotemporal correlation ρ (4) in a neighborhood of a location $\mathbf{p}$ can be written as follows:

$$\rho_{\mathbf{p}}(t, x, y) = \exp\left(-(t, x, y) \Sigma_{\mathbf{p}}(t, x, y)^T / 2\right), \quad (8)$$

where $\Sigma_{\mathbf{p}}$ is a positive definite matrix with elements depending on L, v_x, v_y , and τ . Now the covariance function between $\ln H_s$ at any locations on the oceans $\mathbf{p}$ and $\mathbf{q}$ is given by

$$\rho(\mathbf{p}, \mathbf{q}) = \sqrt{\frac{2^3 |\Sigma_{\mathbf{p}}|^{-1/2} |\Sigma_{\mathbf{q}}|^{-1/2}}{|\Sigma_{\mathbf{p}}^{-1} + \Sigma_{\mathbf{q}}^{-1}|}} e^{-\mathbf{z}(\Sigma_{\mathbf{p}}^{-1} + \Sigma_{\mathbf{q}}^{-1})^{-1} \mathbf{z}^T}, \quad (9)$$

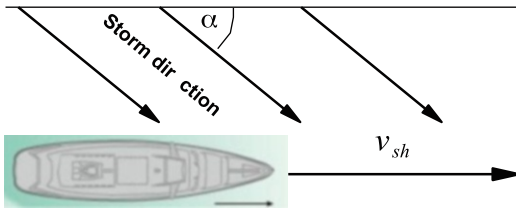
where $\mathbf{z} = \mathbf{q} - \mathbf{p} = (t_{\mathbf{q}} - t_{\mathbf{p}}, x_{\mathbf{q}} - x_{\mathbf{p}}, t_{\mathbf{q}} - y_{\mathbf{p}})$ and $|\Sigma|$ is the determinant of Σ . Consequently for any collections of locations, $\mathbf{p}_i, i = 1, \dots, n$, $\ln H_s(\mathbf{p}_i)$ is a vector of Gaussian variables with means $m(\mathbf{p}_i)$, variances $\sigma_{\mathbf{p}_i^2}$, and correlations $\rho(\mathbf{p}_i, \mathbf{p}_j)$ as in Eqs. (2) and (9). Some further developments of the global model have been done in Ailliot et al. (2011).

Simulation of Encountered H_s

For a vessel moving with velocity $\mathbf{v}_{shp}$ as in Fig. 2, the local covariance between logarithms of encountered H_s is of the following form as in Eqs. (5), (6), and (7)

$$\rho(t) = e^{-\frac{|\mathbf{v}_{shp} - \mathbf{v}|^2 t^2}{2L^2}} e^{-\lambda|t|} \approx e^{-\frac{t^2}{2\tau^2}},$$

where



Ship Operational Environment, Fig. 2 Heading angle α of a ship

$$\tau = \left(-\lambda T_v^2 + \sqrt{\lambda^2 T_v^4 + T_v^2} \right), \quad (10)$$

$$T_v = L / \left\| \mathbf{v}_{shp} - \mathbf{v} \right\|.$$

In general, τ depends on route (season and position of vessel and speed computed from the route), which is a function of time t . The correlation between logarithms of encountered significant wave heights at time t, s $X(t), X(s)$ is given by (Podgórski and Rychlik 2014) as:

$$r(t, s) = \sigma(t)\sigma(s) \sqrt{\frac{2\tau(t)\tau(s)}{\tau(t)^2 + \tau(s)^2}} e^{-(t-s)^2 / (\tau(s)^2 + \tau(t)^2)}. \quad (11)$$

Below, the use of the proposed model is demonstrated to simulate the significant wave heights encountered along the route presented in the Fig. 3. The mean values $m(t)$ and $\sigma^2(t)$ are shown in Fig. 1. The average duration of encountered storm (in hours) $\tau(t)/\pi$ is given in Fig. 4. Here a storm duration is defined as time the encountered H_s stays above the median. One can see that the time duration of storms/calm periods on the route from Europa to America is about 40 h, while the sailing time is about 160 h. Thus one can expect to meet three storms on this route. On the other hand, the route from America to Europe took about 125 h, the average storm period is 130 h, and hence one can expect to meet one storm.

We turn now to more detailed description of the simulation algorithm. The inputs are functions $m(t)$, $\sigma^2(t)$, and $\tau(t)$ presented in Figs. 1

and 3. The functions are sampled with time step 0.5 h (the same as frequency of on board measured H_s). In the following, we denote by $\mathbf{t}, \mathbf{s}$ vectors of time points sampled with frequency two per hour along the route. Furthermore, the significant wave height observed at points $\mathbf{t}$ is denoted by $H_s(\mathbf{t})$, while their logarithms by $X(\mathbf{t})$ and the vector of their means by $m(\mathbf{t})$. The covariance matrix between $X(\mathbf{t}), X(\mathbf{s})$, with entries $r(t_i, s_j)$ computed using (11), is denoted by $r(\mathbf{t}, \mathbf{s})$. Using the introduced notation, the samples of $H_s(\mathbf{t})$ can be simulated as follows

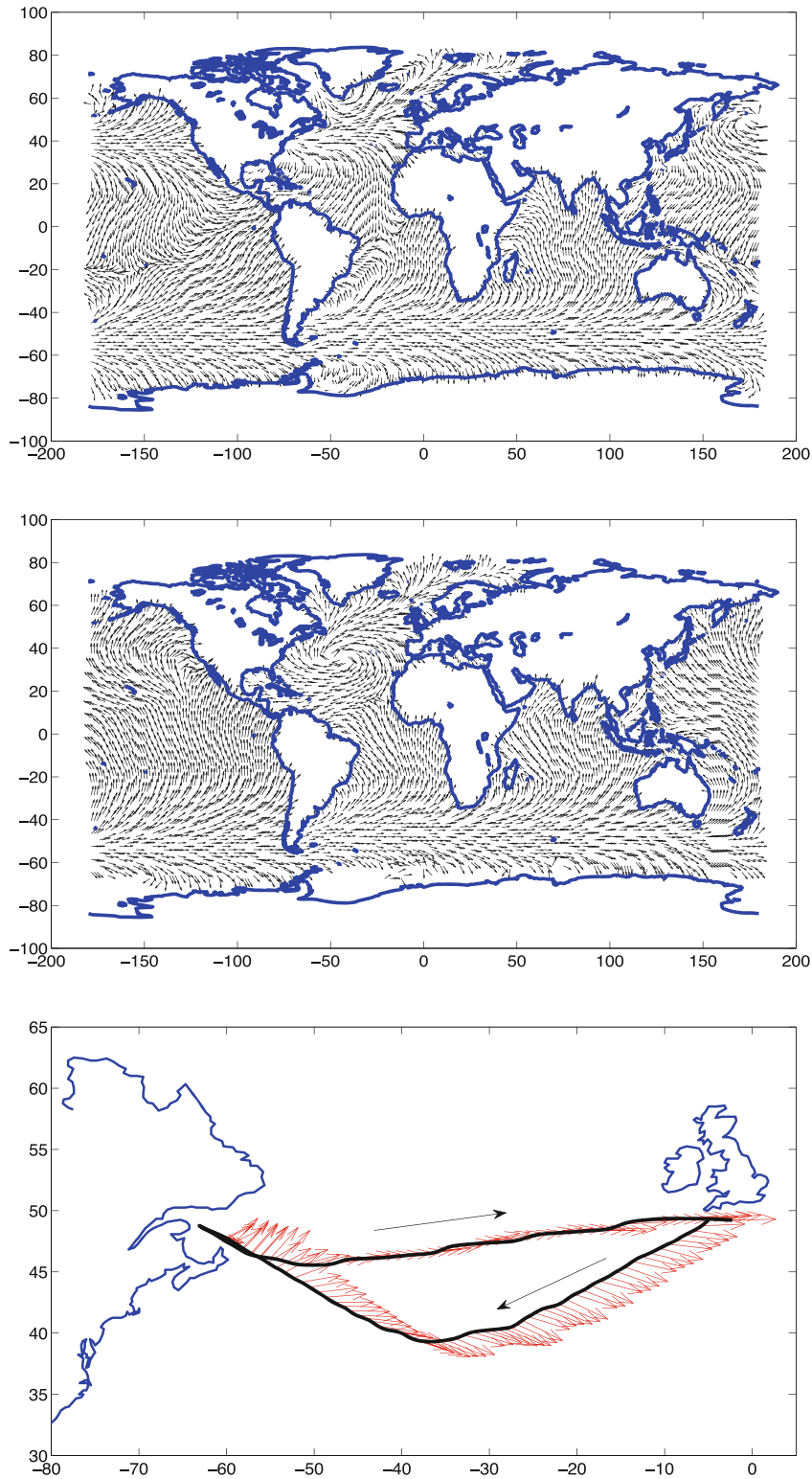
$$H_s(\mathbf{t}) = \exp \left(m(\mathbf{t}) + \sqrt{r(\mathbf{t}, \mathbf{t})} \mathbf{Z} \right),$$

where $\mathbf{Z}$ is a vector of independent standard normal and $\sqrt{A}$ is any matrix such that $A = \sqrt{A}\sqrt{A}^T$.

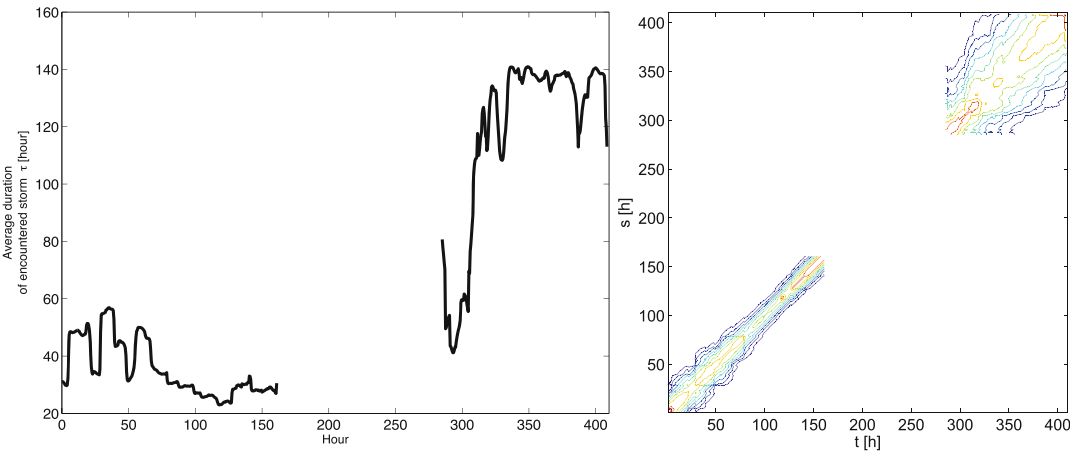
For the ship routes presented in Fig. 3, three simulated H_s encountered along the route are compared with the measured H_s as in Fig. 5. The simulated values can well reflect the characteristics of the significant wave height measured along the ship routes. In the following subsection, we will show a solid example of using the model and simulation to estimate the ship's performance, i.e., damage accumulation, during the measured route. The estimated damage results will be compared with the full-scale measurement to validate the capability of the model for other similar maritime application.

Expected Fatigue Damage During a Voyage

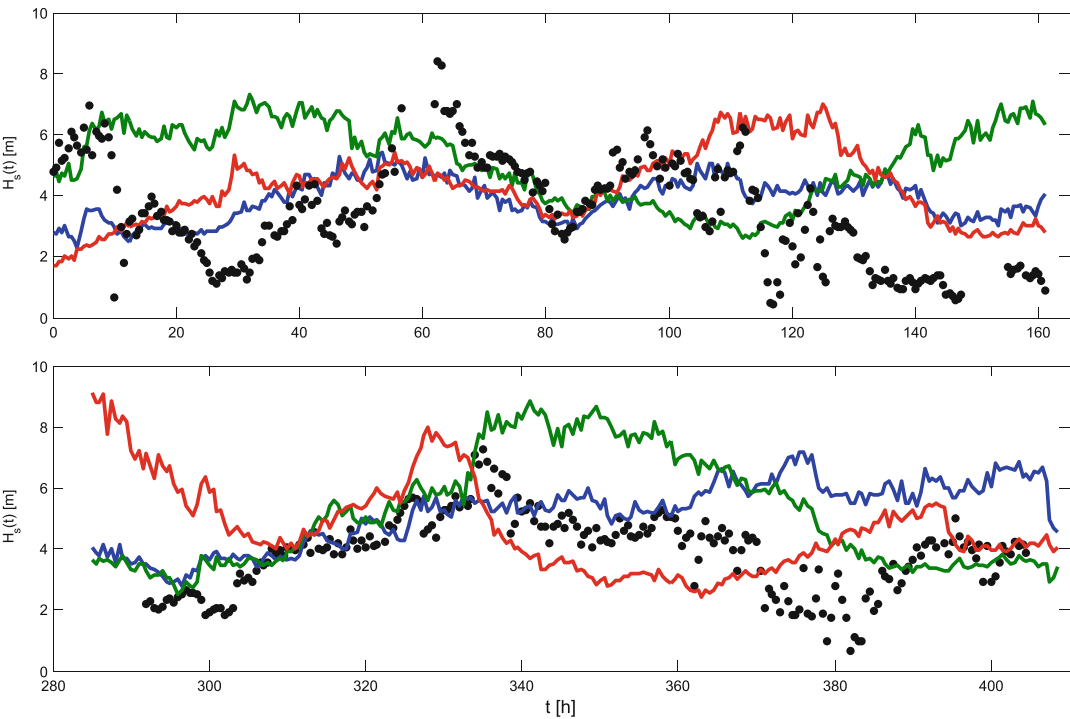
In the maritime industry, safety and energy efficiency are two fundamental issues that have to be seriously taken into account in all maritime-related activities. For the evaluation of ship safety and energy consumption-related parameters, to get reliable sea conditions is one of the key elements in the evaluation process. For example, estimation of expected fatigue $\mathcal{D}_\beta$ for a specific trade route is a central issue in the reliability analysis to ensure a ship's structural integrity. A common approach is to represent encountered loads as a sequence of stationary sea states during which the expected damage increases at a rate



Ship Operational Environment, Fig. 3 Estimates of the median H_s velocity (v_x , v_y) for the wave fields in February (top) and August (middle). The lengths of vectors are not up to scale. Bottom: estimates of the median velocity (v_x , v_y) along the route in January



Ship Operational Environment, Fig. 4 Parameters of $X(t) = \ln H_s(t, \mathbf{p}(t))$ for the route presented in Fig. 3 (bottom). Left: the expected duration τ / π of encountered storm. Right: covariance Eq. (11) between $X(t)$ and $X(s)$. The covariance is zero for $160 < t < 270$ or $160 < s < 270$ since the vessel is close to or in the harbor



Ship Operational Environment, Fig. 5 Three simulated H_s encountered along the route presented in Fig. 3. The simulated values are exhibited in solid lines, while dots represent measured H_s on board of a vessel sailing the same route

depending on shipping conditions and sea states parameters. The damage accumulation rate at time t is denoted by $d(t)$.

The method introduced in Mao et al. (2010) is illustrated for a container ship traveling regularly between Europe and North America. The fatigue

damage rate $d(t) = d$ in a container ship was given, viz.,

$$d = \frac{0.47C^\beta}{a} \times \left(\frac{H_s^{\beta-1/2}}{3.75} - \frac{2\pi}{g} \frac{H_s^{\beta-1}}{3.75^2} \|\mathbf{v}_{sh}\| \cos \alpha \right) = d(H_s, \mathbf{v}_{sh}, \alpha), \quad (12)$$

where H_s is significant wave height encountered by the ship, $\mathbf{v}_{sh}$ is her velocity, and α is the heading angle as given in Fig. 2 or by using the velocity of $\mathbf{v}$ of H_s field through $\cos \alpha = \frac{\langle \mathbf{v}_{sh}, \mathbf{v} \rangle}{\|\mathbf{v}_{sh}\| \|\mathbf{v}\|}$, where $\langle \cdot, \cdot \rangle$ is the inner product between vectors. For compactness of the formulation, the dependence on time t and a position of the ship $\mathbf{p}(t) = (x(t), y(t))$ (a route of the ship) is not explicitly shown. In the formula a is fatigue strength of material taken from S-N curve of parameters $a = 10^{12.76}$ while $\beta = 3$. The constant C depends on the type of a ship and the geometry of ship detail: ship speed and heading angle. It can be evaluated using linear strip code, beam theory, and stress concentration factor, here $C = 20$. It is also assumed that actual encountered waves travel in the same direction as that the field H_s moves in.

The method is constructive and requires a detailed shipping plan which consists of voyages between harbors. A single voyage is then represented by a route, i.e., a sequence of times t_i (in hours) of the (approximately) constant sea states and corresponding positions in space $\mathbf{p}_i = \mathbf{p}(t_i)$ (in degrees). In Fig. 3, *bottom*, a route that will be considered in examples is presented as a black line. The route started close to England and took 408 h to sail (time in an American harbor included).

Now for a given route which sails in T hours the accumulated damage is

$$\mathcal{D} = \int_0^T d(t) dt \approx \sum d(t_i) \Delta t, \quad (13)$$

and the expected damage $E[\mathcal{D}] \approx \sum E[d(t_i)] \Delta t$ is used to estimate risk for cracking of ships details.

Here expectation $E[d(t_i)]$ averages the variability of significant wave heights, heading angle, and the ship velocity at time t_i when ship is at position $\mathbf{p}(t_i)$. In order to find the distribution for any route, one needs atlases of distributions of $H_s(\mathbf{p})$ at any position and season (here represented by a month). Several such atlases exist, and an example of such maps is presented in Fig. 1 one estimated from H_s recorded by satellites.

The second problem is description of variability of the ship velocity $\mathbf{v}_{sh}$ and heading angle α . Here the ship's velocity and heading along the route are approximated by their average values that are derived as follows. The speed is taken as the "design speed," and the direction is evaluated as the tangent to the route. The heading is taken as the angle between the average ship velocity and the median velocity of H_s at the position as shown in Fig. 3.

Here we shall illustrate the presented methods by estimating the expected damage accumulation rate along the route presented in Fig. 3 (*bottom*) that took 17 days to sail in January month. The damage rate is given by Eq. (12) with ship velocity and H_s field velocity replaced by their median values (see Podgórski and Rychlik (2014) for more details). The average H_s field velocities along the route are available and presented in Fig. 3. What remains is computations of expectations of H_s^2 and $H_s^{2.5}$. For log-normally distributed H_s defined by $m = E[\ln H_s]$ and $\sigma^2 = \text{Var}(\ln H_s)$, the average value of H_s^r is given by

$$E[H_s^r] = \exp(rm + r^2\sigma^2/2).$$

Since the duration of the voyage is less than 1 month, the dependence of distribution of H_s on time can be neglected, and hence the damage rate can be computed when the parameter values $m(\mathbf{p})$ and $\sigma^2(\mathbf{p})$ are known for locations $\mathbf{p}$ along the route. This can be done using the model proposed in Baxevari et al. (2005) and fitted to all locations on oceans in Baxevari et al. (2008a). Typical fits following this approach see Fig. 1.

Using the evaluated damage rates presented in Fig. 6, the accumulated damage for the entire voyage can be computed. The average damage

rate (dashed line) results in expectation that 0.99% of the total fatigue life has been “consumed” in this 17 days long voyage. This estimate applies to a midship location in 2800 TEU container ship (see Mao et al. (2010) for more details). During six winter months, between 6 and 8 such voyages can be undertaken which indicates high risk for cracking in some ship details before the design life of 20 years.

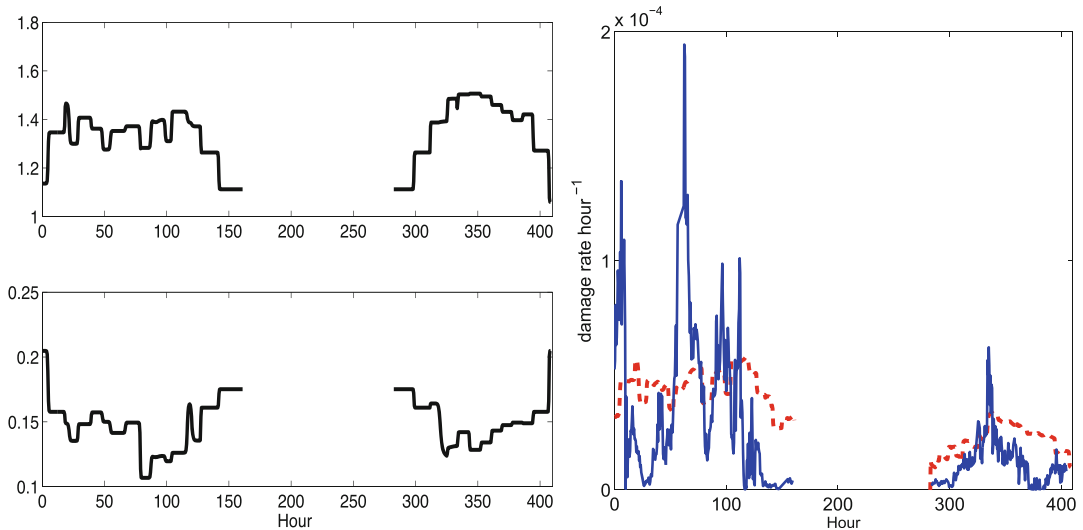
The observed damage rates, although uncertain due to large estimation errors and missing values, give accumulation of 0.72% of the life during the voyage. The difference between observed and the expected damage is caused not only by randomness of the sea conditions but also by captain decisions and routing programs. A wish to avoid predicted large storms resulted in the decision of taking longer route to the south to avoid the worse sea conditions, but despite these precautions, still three storms have been met on route to America.

Since on the route back to Europe one is sailing in similar direction as the motion of storms, the damage rate is much smaller. The weather in the middle of this route is harsher than the conditions that can be met on the chosen route to America as seen from the parameters m and σ^2 in Fig. 6 (left). Mathematically, it is the term $\|\mathbf{v}_{sh}\| \cos \alpha$ that

makes the damage rate smaller, and as a result, 80% damage were accumulated on the way to America and only 20% on the way back. The difference is a well-known fact, and hence sailing from America to Europe is often taken along the shortest route with a maximum speed.

Spatiotemporal Models for Winds

There are tremendous renewable wind powers in the open sea. For the development of auxiliary propulsion, using wind power in ships relies on accurate description of wind conditions along a ship’s actual sailing route. In such application, the wind conditions often relate to the variation of wind speeds encountered by ships. A good way to model the variation is by developing some statistical spatiotemporal wind models. Development of a fully spatial-temporal model to describe the variation of wind velocity seems to be a difficult problem (see Monbet et al. (2007) for a review). A simpler way is to derive conditional distributions of wind direction given the wind speed. In Bessac (2014), statistical means to derive the conditional distribution close to French coast in January month are given. The



Ship Operational Environment, Fig. 6 Analyses for the route in Fig. 3 (bottom) in dependence on sailing time. Left: mean (top) and variance (bottom) of $\ln H_s(t)$.

Right: the expected (mean) damage rate (dashed line) vs. the observed damage rate (solid line)

spatiotemporal model for wind velocity was discussed in Bessac et al. (2016a, b). The approach used hidden Markov processes to model properties of wind velocity on a grid. This is numerically a very intense method, which limits the size of the grid.

Furthermore, the average wind speed is an important parameter in wind gust spectra and models for wind speed variation in both space and time. The spatiotemporal models for wind speeds in North Atlantic and Arctic have been proposed in Rychlik and Mao (2014), Rychlik (2015), and Mao and Rychlik (2018). The proposed models are of the same type as used to describe variability of the significant wave heights. The main difference is that the logarithmic transformation of data is replaced by an exponential transformation proposed in Brown et al. (1984). In Mao and Rychlik (2016), it was also demonstrated that wind speeds encountered by a ship are Weibull distributed. In this work it was also demonstrated that in the Caribbean Sea, the Weibull model does not fit well with the observed data in regions where extreme wind speeds can arise during hurricanes or tropical storms. Rarity of such events makes it difficult to find a suitable transformation so that transformed wind speeds can be model as Gaussian fields, i.e., the model used to describe the variability of significant wave heights.

In Rychlik and Mao (2018), a new model for exponentially transformed wind speeds was proposed. It is a sum of a Gaussian field and several non-Gaussian random fields (with Laplace distributed amplitudes), which are independent and randomly spread in time and space. These non-Gaussian fields describe strength and dynamics of cyclones and tropical storms. In Rychlik and Mao (2018), only local model, valid in geographical region of about 4 degrees radius and for a period of 1 month, was presented. A global model is not developed yet. Work in this direction is undertaken. Furthermore there is no spatiotemporal model for wind speeds over oceans estimated yet. Consequently modeling of spatiotemporal wind speed variability is less developed as the modeling of significant wave heights.

Cross-References

- [New Technologies in Auxiliary Propulsions](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

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Ship Optimization

► Ship Construction and Operation Impact on Energy Efficiency

Ship Overall Design

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Synonyms

Ship general design

Definition

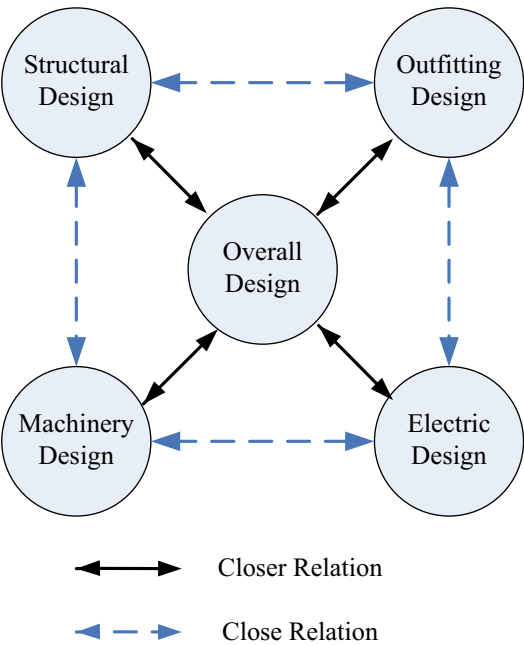
Ship design can be, by discipline, subdivided into overall design, structural design, outfitting design, machinery design, and electric design, wherein the overall design has been recognized to play a dominant role over others. Figure 1 shows the ties among each discipline.

Ship overall design is a process that starts from the determination of ship's main dimensions and generation of hull form, followed by general arrangement and lightweight estimation, based on which the ship's performances such as velocity, maneuverability, sea-worthiness, and stability can be further analyzed. The coverage of ship overall design ranges from model tank testing to delivery of the as-built document, so to speak.

Scientific Fundamentals

Basic Theory

As the fundamental theory and method of ship overall design, the knowledge contained in the



Ship Overall Design, Fig. 1 Status of overall design

following three subjects have to be learned and mastered.

Ship hydrostatics mainly introduces how to generate ship lines and calculate hydrostatic data, floatability (static equilibrium of a ship), initial stability (at a small rolling angle), large angle stability, and launching process.

Ship dynamics based on fluid dynamics elaborates on the basic principle of ship resistance, propulsion, maneuverability, and sea-keeping performance.

Principles of ship design, developed on the ground of above fundamental theory, teach people how to determine the ship's main dimensions, general arrangement, lightweight, capacity, cost, and the like.

Input of Design

Ship overall design has to proceed in accordance with following requirements.

Mission and owner's requirements addressed in contract and ship specification such as cargo category and capacity, principal characteristics, rules and regulations, number of crew, endurance, service speed, specific fuel oil consumption (SFOC), deadweight, as well as the vessel design life.

Classification requirements reflected in rules issued by certain classification Society such as American Bureau of Shipping (ABS) or China Classification Society (CCS) which mission is to provide a ship classification service and supervision over the design quality and shipbuilding process.

Regulatory requirements stipulated in all the documents such as conventions, Clcodes, and resolutions issued by the International Maritime Organization (IMO). The most important two conventions, among those documents, are the International Convention for the Safety of Life at Sea (SOLAS) and the International Convention for Prevention of Pollution from Ships (MARPOL). The former is mainly concerned about the safety of human life onboard, while the latter is much concentrated on the prevention of various pollutions from ships.

ILO Convention No. 92 & 133 published by the International Labor Organization (ILO). The

articles in this convention are mainly about screw's living conditions to be satisfied during the overall design.

Regulations of authorities issued by, for example, USCG (United States Coast Guard) or ACP (Autoridad Del Canal De Panama), etc. For ships intent to go through US waters or Panama Canal, those regulations have to be followed up.

Design Approach

"Design Spiral": An Iterative Method

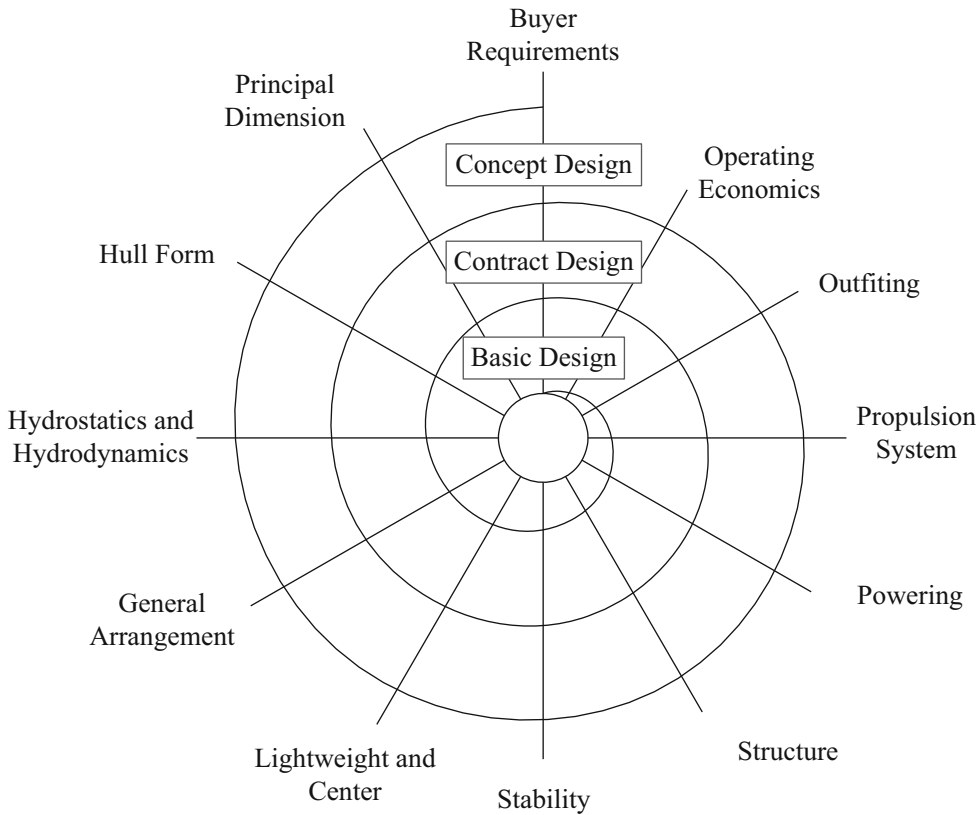
The design spiral shown in Fig. 2 effectively illustrates the sequential course of ship design through the various design steps, the repeating, iterative procedure for the determination of ship dimensions and of other properties, and, finally, the gradual approach to the final stage of detailed ship design.

The usual and recommended steps when the preliminary ship design is elaborated are listed below:

- (a) Critical evaluation of ship buyer's major requirements with emphasis on those which influence the main dimensions' selection.
- (b) Data gathering (by ship type, size, DWT, speed, and power of main engine installed onboard) of built similar ships in available publications, including databases. Designers working in the shipbuilding industry or design offices may exploit available technical information about built ships filed in the design department's records.
- (c) Identification and study of the relative/ corresponding regulations concerning the specific ship type design, construction, and operation: conventions, national and international regulations, class society rules, technical notes, guidelines, and instructions.

"Parent (Basic) Vessel": A Transformation Method

With accumulation of ship design and development, a ship form database composed of variety of excellent ship types has been formed, which can be used as a reference for subsequent ship design. An excellent ship data from such database can be



Ship Overall Design, Fig. 2 Design spiral

chosen as a parent ship for a new ship design, provided both vessels are similar in size and type. This can get rid of a tedious and time-consuming spiral design process and ensure that the new ship is free of risk in performance and safety. Such a result can be concluded even at the very beginning of ship design without detailed calculation. The parent vessel design method is widely recognized and accepted as the first choice for modern ship design.

Key Applications

As a results of key application of above fundamental theory and design approach into ship overall design, some documents (output of ship overall design) would be worked out as shown, taking a document list for typical oil tanker as an example, in Table. 1.

Further interpretations to parts of such documents are given as following.

Lines

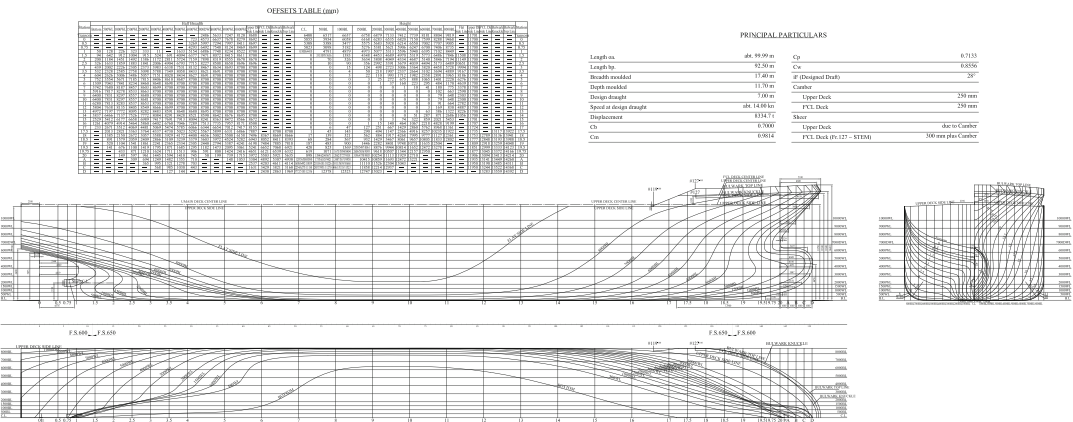
An important drawing used to describe the shape of a ship (shown in Fig. 3). It gives an offsets table and three orthogonal views, the so-called buttock, body, and waterline which worked out from the Offsets. The main dimensions and ship form coefficients are also included in this plan. Lines is considered the basis of all the ship performance calculation as well as an indispensable shape reference for general arrangement and structural design.

General Arrangement

A drawing presenting the overall appearance of the ship consisting of profile view, deck/platform views, and front view (see Fig. 4). It divides the whole ship into several functional zones such as

Ship Overall Design, Table 1 Documents produced in ship overall design

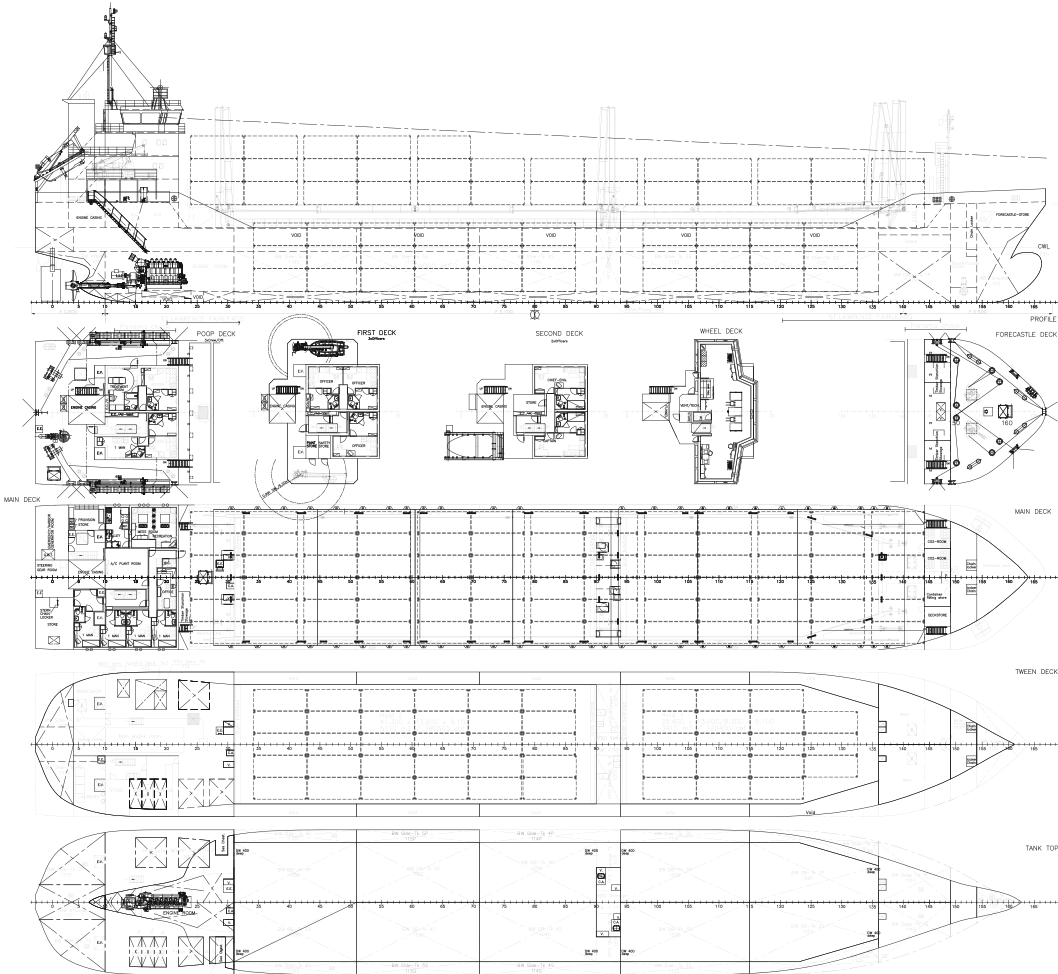
No.	Title of document	Function	Design basis
1	Lines	Design foundation	Mission & Buyer’s requirement
2	General arrangement		
3	Hydrostatic data		
4	Capacity plan		
5	Tank sounding table		
6	Model test report	Performance verification	
7	Procedure of sea trial		
8	Report on sea trial		
9	Freeboard calculation	Safety	SOLAS
10	Visibility plan		
11	Freeboard mark plan		
12	Freeboard plan		
13	Damage control plan		
14	Fire control plan		
15	Procedure of inclining test & lightweight measurement	Stability	Classification society’s requirement
16	Results of inclining test & lightweight measurement		
17	Preliminary trim & stability calculation		IMO A. 749
18	Preliminary damage stability calculation		
19	Final trim & stability calculation		
20	Final damage stability calculation		
21	Loading manual	Operation	SOLAS
22	PSBT calculation (tanker only)	Pollution prevention	MARPOL
23	MARPOL 12A (tanker only)		
24	Ballast water management plan		
25	Docking plan	Maintenance	Buyer’s requirement



Ship Overall Design, Fig. 3 Ship lines

cargo holds, machinery space, fore, and aft parts as well as accommodation zone. Cabins layout on decks in accommodation zone, equipment layout

on platforms in machinery space, subdivision of liquid tanks and cargo holds, layout of mooring equipment on the fore and aft deck, all the



Ship Overall Design, Fig. 4 General arrangement

hatch, stairway and passageway, positions of navigation, and life-saving and communication apparatus are given as a foundation for future development.

In addition, the main particulars of the ship, such as main dimensions, displacement, deadweight, main engine power, main engine revolution, complement (number of crew), endurance, deck height, and classification notation, are also indicated in this drawing.

Hydrostatic Data

A set of hydrostatic data that varies against draft. Usually the following parameters have to be included:

- Molded displacement of volume: V .
- Displacement: D .
- Water plane area: A_w .
- Tons per centimeter immersion: TPC.
- Longitudinal center of floatation: LCF.
- Longitudinal center of buoyancy: LCB.
- Vertical center of buoyancy: VCB.
- Transverse metacenter: KM_T .
- Longitudinal metacenter: KM_L .
- Wet surface area: A_S .
- Moment to change trim 1 cm: MTC.
- Midship area coefficient: C_M .
- Waterplane coefficient: C_W .
- Block coefficient: C_B .
- Prismatic coefficient: C_P .

Capacity Data

A document showing capacity data of all the cargo holds and liquid tanks which would offer necessary data for inclining test, stability calculation and calculation for various loading condition, and so forth.

The data would include cargo hold and liquid tanks capacity, centroid position and free surface inertia moment of each cargo hold, and liquid tank at different load height.

Capacity Plan

A plan presenting the size and position of all the cargo holds and liquid tank of the vessel.

A summary information including category, position, capacity, centroid position, and maximum free liquid surface inertia moment for each cargo hold and liquid tank would be given in this plan.

Tank Sounding Table

A document showing all the liquid volume in tank corresponding liquid depth, offering necessary data for inclining test, damage stability calculation, and ship operation.

Different from the document of capacity data mentioned above, the sounding table shall provide the location and size information of all tank

sounding pipes along with the liquid volume corresponding to each sounding in tanks with even, heel, and trim condition so as to provide the correct capacity information for the chief officer.

Visibility Plan

A drawing (shown in Fig. 5) checking if the blind sector forward of the bow, from the point of conning, satisfies or not the requirement of classification society. All the loading conditions from the preliminary or final stability calculation have to be considered.

Free Board Marks Plan

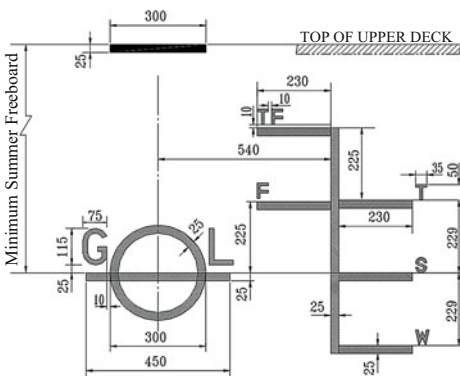
A drawing indicating the assigned freeboard mark and load lines for various drafts as well as the freeboard deck line (shown in Fig. 6).

The freeboard mark is composed of a ring and a horizontal line intersecting the ring. The letters on either side of the ring represent the classification society who assigned this mark on behalf of competent authority.

The load lines according to the ship's navigational area and season are represented by horizontal lines with a letter.

- Letter “S” stands for the summer load line.

Ship Overall Design,
Fig. 5 Visibility plan



Ship Overall Design, Fig. 6 Ship free board marks

- Letter “T” stands for the tropical load line.
- Letter “W” stands for the winter load line.
- Letter “F” stands for the summer freshwater load line.
- Letter “TF” stands for the tropical freshwater load line.
- Letter “WNA” stands for the North Atlantic winter load line.

Damage Control Plan

A drawing presenting all the information pertaining to ship’s damage controlling.

Such information include:

- Watertight boundaries of the ship.
- Locations and arrangements of all cross-flooding systems together with the locations of their valves and remote controls, if any.
- Locations of all internal watertight closing appliances.
- Locations of all the openings and their closure appliance whether on deck or on side shell.
- Location of all bilge and ballast pumps, their control positions, and associated valves.

Preliminary Trim and Stability Calculation

A calculation for all intended loading cases based on general arrangement, capacity plan, and data as well as preliminary lightweight result.

The trim and stability for each loading case with departure, midway, and arrival shall be checked to demonstrate the compliance with the criteria of applicable rules and regulations.

Preliminary Damage Stability Calculation

A calculation for all specified damage scenario based on general arrangement, capacity plan, and data as well as preliminary lightweight result.

The stability and final equilibrium waterline for each damage scenario for every loading case with departure, midway, and arrival shall be checked to demonstrate the compliance with the criteria of applicable rules and regulations.

Loading Manual

A document providing the similar contents as the preliminary trim and stability calculation, however, updated with approved lightweight data.

In addition, this manual should contain more information rather than preliminary trim and stability calculation such as:

- Instruction to Captain.
- Envelop of bending moment and sheer force.
- Draught limitations (in ballast, etc.)
- Load specifications for cargo decks.
- Cargo mass and cargo angle of repose restrictions.
- Cargo density and filling heights for cargo tanks.
- Restrictions to GM-value.

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Ship Propulsion System

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Synonyms

[Air emissions](#); [Automatic identification system](#); [Energy efficiency](#); [Environmental impact](#); [Fuel consumption](#); [IMO \(International Maritime Organization\)](#); [Shipping](#)

Definition

Ship propulsion system is the most vital component of marine power plant system, which

provides propulsion forces for ships to maneuver themselves. Most of ship propulsion systems may consist of main engine, shafting system, and propeller, which convert the energy of the fuel from main propulsion engine into useful thrust to propel the ship and match the ship resistance with the help of the shaft.

Types of Ship Propulsion System

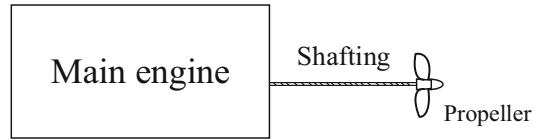
Using the best type of propulsion system is vital to ensure a better safety standard for the marine ecosystem along with cost efficiency. Moreover, the type of propulsion system used in a ship depends on several factors such as speed, power, ship type, etc. According to power transmission mode, there are several forms of propulsion system enumerated as follows (Anthony et al. 2011).

Direct Drive Propulsion System

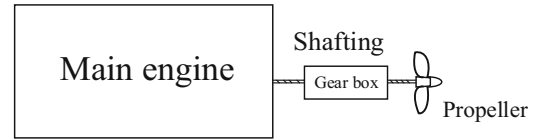
Direct propulsion system is the most commonly used on the ship, in which the power output from engine is translated to propeller directly through the shafting without decelerator or clutch. In this type of propulsion, the rotation speed of main engine and propeller are the same, thus operating of the ship can be done by altering the speed and direction of the main engine. Since the mechanical simplicity and easy maintenance, direct drive propulsion system is the most commonly used system on ships. Also, it has fewer energy losses. However, this type of system usually has big size and weight, and the speed is restricted by main engine when ship needs a very slow speed. Thus, direct drive propulsion system as shown in Fig. 1 is generally applied to large-scale merchant ship which usually incorporates large bore low-speed diesel engines.

Indirect Drive Propulsion System

Unlike direct propulsion system, the power transfer process of the indirect drive propulsion system from main engine to propeller shown in Fig. 2 contains some intermediate conversions, which results from the installation of intermediate devices such as clutch and decelerator. The installation of decelerator reduces the number of revolutions from the engine output in such a way that



Ship Propulsion System, Fig. 1 Direct drive propulsion system



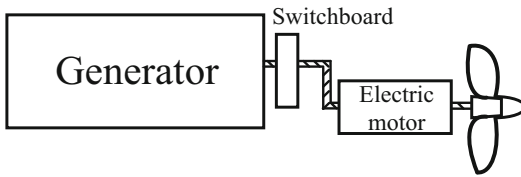
Ship Propulsion System, Fig. 2 Indirect drive propulsion system

the system may derive maximum propeller efficiency.

The advantages of such system are the following: (1) the propeller can keep very low rotation rate without limitation from main engine, (2) because of the existence of clutch, changing over is not needed anymore, and (3) such a system is more convenient for reversing and setting shaft generator. However, with the additional procedures of energy conversion, the energy costs more, and the efficiency of the whole system decreases. Since the pros and cons, the indirect drive propulsion system is more common in small- and medium-sized ships, and it is also suitable for large-sized ships equipped with high-power medium speed diesel engines.

Electrical Propulsion System

In addition to two types of propulsion system mentioned above, another widely used propulsion system is electrical propulsion system. This relative new propulsion mode produces less pollution compared with conventional propulsion system. In this way, the propeller relates to the propulsion motor, which is driven by generator engine through main switch board as shown in Fig. 3. Some redundant shaft and bearing are omitted in electrical propulsion system. Thus, the space is saved. Also, the direction of the rotation of propeller is governed by altering the direction of current of electrical motor, which guarantees the ship maneuverability, and rotation can vary with more extensive range. However, the electrical



Ship Propulsion System, Fig. 3 Electrical propulsion system

propulsion system may cost extremely more than direct and indirect types in the initial cost and maintenance. Moreover, in the energy conversion process of the system, the mechanical energy is converted into electric energy and then into mechanical energy, which contributes to low efficiency of the whole system. Electrical propulsion system is commonly applied to icebreakers or engineering ships such as tug or dredger.

Shafting Transmission System

Shafting transmission system is an essential unit on a power-driven ship, which functions as the transmission converting the power from main engine to the propeller (Lee 2018). The useful thrust from the propeller is also converted to the ship through the transmission system. The shaft is manufactured from forged steel, completed with coupling flanges. Generally, the transmission system consists of various shafts (thrust shaft, intermediate shaft, tailshaft), bearings (thrust shaft bearing, intermediate shaft bearing, tailshaft bearing), and ultimately the propeller itself.

The propeller shaft usually connects the transmission inside the vessel directly to the propeller, passing through a sealing arrangement at the end of the tailshaft which connects directly to the propeller to complete system's arrangement.

Around the shafts are bearings, spacingly fitted along the length of the shaft, to bear the weight of the shaft to prevent shaft bending and constrain its radial motion. The bearings can also raise the shaft's axial bearing capacity while it holds the tensional forces from water repulsive. It is noteworthy that there is a thrust block, a specialized form of thrust bearing installed just aft the main engine, to support thrust shaft, resist the thrust of

the shaft, and transfer it to the hull of the ship. The thrust block itself is carried by bearing pads with the helical oil grooves cut into them. The pads faced with white metal are mounted in holders around a steel support, holding them against a machined collar on the drive shaft. The intermediate shaft bearing only has bottom bearing, but the aftermost tunnel bearing is the only one equipped with both top and bottom bearing shells since it has to offset the propeller mass and bear a vertical upward thrust of the tailshaft. The installation for carrying the tailshaft is called stern tube which is supported by stern tube bearing. The tube consists of a cast iron tube welded into the stern frame. And it is responsible for lubrication and sealing tasks arrangements of tailshaft.

During the time when the ship is moving, the imbalance rotatory motion of the main engine creates excessive excitation forces on the shaft which leads to undesirable axial vibration in the ship body. Meanwhile, shaft suffers from the inconstant cyclic torsional-bending vibrations which attributes to considerable hydrodynamic pressure in the longitudinal direction, so-called longitudinal vibrations. The longitudinal vibrations also cause the wear damage of thrust bearings. Moreover, the increasing torsional vibrations may generate large torsional stress on the shafting, which would be imposed the existing stress from torque transmission, axial stress, and longitudinal stress. The torsional vibrations mainly result from imbalance thrust of propeller. Besides, excessive vibrations in torsional, longitudinal, and axial modes and their coupled vibration forms are hostile and may lead to massive failures when operating shafting (Anthony et al. 2011). Thus, the reliability of the shaft is crucial to the ship navigation safety. The proper estimation of the dynamic performance of shaft is significant to guarantee the optional power delivery to the propeller and to minimize vibrations.

The torque M on the shafting can be calculated as follows:

$$M = 9.55 \frac{P}{n} \times 10^3 \quad (\text{N} \cdot \text{m}) \quad (1)$$

where n is the rpm with unit of r/min and P is the power translated by shafting. Meanwhile, the

thrust T delivered by shafting can be calculated as follows:

$$T = 1.94 \frac{P_p}{v} \eta_p \quad \text{kN} \quad (2)$$

where P_p represents propeller achieved by propeller, v is the ship speed, and η_p is the efficiency of propeller.

The shaft works under severe conditions, bearing force and torsion as shown in Fig. 4. In addition, the increase of load in the shaft system occurs owing to the influence of the propeller weight at the shaft end and effect of other attachments. Meanwhile, the journal rubs against the bearing when system is working. Also, journal and stern tube would be prone to corrosion if stern bearings are lubricated by seawater.

As mentioned above, owing to the rough condition, damage, moreover fatigue, would be frequent (Lee 2018). To reduce damage, the following conditions should be fulfilled: (1) the shaft must have enough strength and rigidity; (2) the conversion of power needs to be efficient; (3) the shaft should have good adaptability to hull deformation; and (4) torsional resonance is forbidden. Within the prescribed range of operation, (5) the shaft system should have good sealing, lubrication, and cooling performance; (6) the shaft system needs to be conveniently maintained and managed.

Propeller

Propeller Geometry

A marine propeller is the most common propulsor on ships, imparting momentum to a fluid which

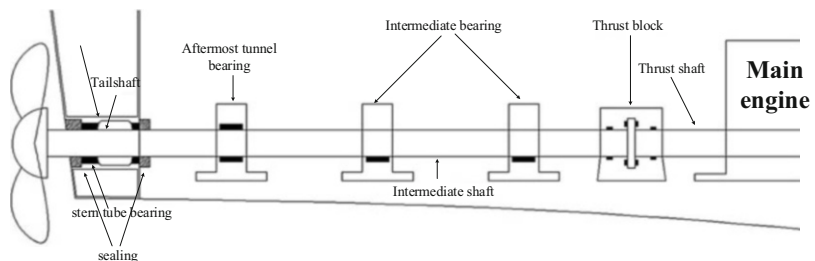
generates the desired thrust to act on the ship (MAN 2018). Based on the torque or power generated, rotational motion will push the water backward and the ship forward.

A generic propeller has two to seven numbers of blades, depending on hull and shafting vibration frequencies, space-equally attached to the boss which is fitted at the end of the propeller shaft. The boss of the propeller is fitted with tapered propeller shaft for tight connection and then jointed with the propulsion mechanical system (Yeo and Ong 2014).

Blades are typically defined by a series of sections stacked at several radial locations. And these sections lie on cylindrical surfaces coaxial with the propeller shaft which also represents the rotation axis. View from aft, the outline of a blade is not symmetric about the generator line. The geometry of the propeller is for one skew angle for various requirements.

When propeller is rotating ahead, the front edge of a blade that enters water initially is leading edge, and the other is trailing edge. These two edges meet at the blade tip which is in the outermost position of the blade. The propeller radius refers to the distance between the tip and the centerline of shaft, while diameter is twice this distance. The size of the propeller is crucial for operator to access the require torque and thrust of the propulsion system. The longitudinal distance in a complete revolution of the propeller is particularly called pitch. Moreover, the governing point to form the pitch over the surface of the rotating cylinder volume as it rotates will form a helix line. The pitch angle is formed by pitch and base of cylinder surface which may have significant influence on the displacement of each rotation; see Fig. 5.

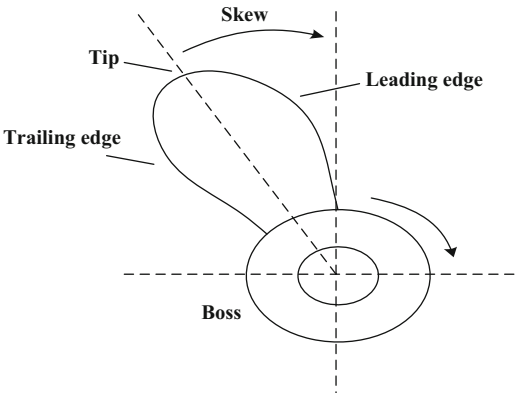
Ship Propulsion System,
Fig. 4 Shaft force transmission system



The face pitch ration which is one of main structural parameters of propeller as in Fig. 6 can be calculated as follows:

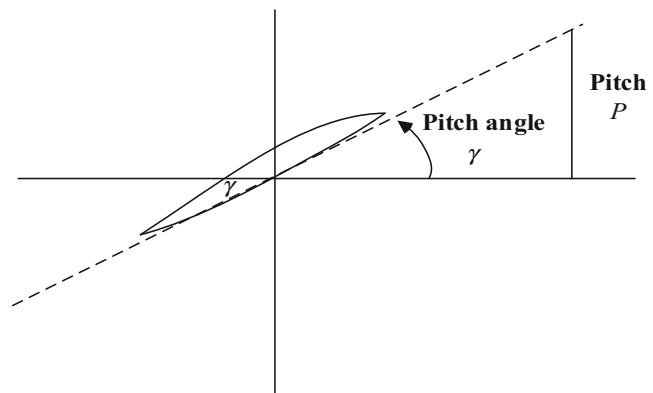
$$P/D = \text{pitch of generator surface/ diameter} \quad (3)$$

The pitch can be fixed or varied across the blade radius. For the latter, the propeller is called controllable-pitch propeller (CPP). A mean pitch may be quoted which is an average value over the blade. It is noteworthy that even for constant radial pitch, the pitch angle γ will change radially, from a large angle at the blade root to a small angle at the tip. Let r represent the local radius. The local pitch angle can be calculated as follows:



Ship Propulsion System, Fig. 5 Propeller geometry (Yeo and Ong 2014)

Ship Propulsion System, Fig. 6 Pitch and pitch angle



$$\tan \gamma = \frac{P}{2 \pi r} \quad (4)$$

Because water is pushed backward a distance by propeller, the actual advance distance of the ship h_a is less than pitch. The slip is $P-h_a$ and slip ratio S can be calculated as follows:

$$S = \frac{P - h_a}{P} \quad (5)$$

Moreover, propeller advance ratio of prop J is determined as:

$$J = \frac{h_a}{D}. \quad (6)$$

Types of Propeller

(1) Controllable-Pitch Propeller

As the name implies, in the controlled-pitch propeller, the pitch can be altered for different encountering conditions which is generally achieved with hydraulic mechanical and hydraulic arrangement or pistons. The changeable pitch to match the required condition is useful in changing the speed of the ship without rising or lowering the speed or rpm of the main engine and drive ship reversely without the necessity to alter the direction of rotation of the engine. Therefore, the maneuverability improves (faster response of speed change), and the loss in efficiency is smaller compared with fixed pitch propeller in astern condition. However, because of mechanical complexity, the cost of installation and maintenance will be higher, and it should be noted that the hydraulic

oil in the boss which is used for controlling the pitch may leak out resulting in pollution. Controlled-pitch propeller is extensively equipped in tugs, ferries, and icebreakers which need variable speed changing during the operation.

(2) Fixed-Pitch Propeller

The fixed-pitch propeller is designed by the way that the blades are permanently attached to the boss with fixed blade angles. The pitch of this type of propeller is unchangeable since the propeller is casted. The fixed-pitch propeller is generally designed for achieving maximum efficiency at only one speed of rotation with one horsepower input amount and one load condition. Meanwhile, the fixed-pitch propeller is robust and reliable since it does not consist of any mechanical and hydraulic arrangement as in controlled-pitch propeller which also tends to decrease costs of maintenance, installation, and operation. However, the maneuverability of fixed-pitch propeller cannot be as higher as controlled-pitch propeller. Thus, the fixed-pitch propeller is usually used to meet requirements of simplicity and lower weight and cost, such as usage in cargo and submarine.

(3) Contrarotating Propellers

These propellers rotate coaxial in opposite directions, as implied in Fig. 7. The upstream propellers are often slightly larger than aft one to take account of the slipstream contraction of the propeller wake and to avoid the aft propeller to

contact with the tip vortex of the upstream one. These propellers can recover rotational losses, more specific, by aft propeller which is often directly coupled with the main engine. Thus, they produce a higher efficiency compared with conventional propellers. But the mechanical installation is more complicated and expensive than conventional propellers as extra weight and complicated gearing and sealing arrangement, as well as more requirements of maintenance. Thus, the contrarotating propellers are hardly used in normal ships but torpedoes, submarines, and some motorboats.

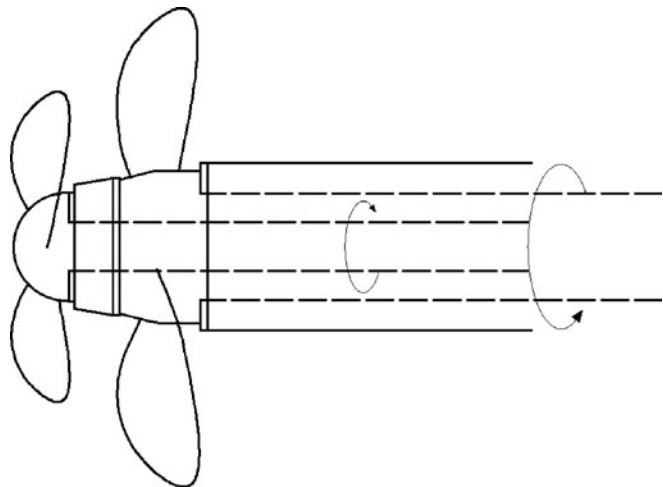
(4) Ducted Propeller

The ducted propeller is the type of propeller fitted with a fixed nozzle. The nozzle is a hydrodynamic designed shroud shaped like a foil encircling the blade tips of a propeller, and it can help in improving the efficiency of propeller.

There are two types of ducts, the accelerating and the decelerating duct. With the former, the duct accelerates the flow inside the duct and hence augments thrust of the propeller. This type usually finds applications in highly loaded propellers and propellers with restricted diameters such as in tugs and trawlers.

With the decelerating duct propeller, the duct circulation reduces the inflow velocity, whereby loss of efficiency and thrust happen, while pressure in the propeller plane increases, resulting in reducing the cavitation risk and noise radiation. Thus, it is used in fast ships or military ships for

Ship Propulsion System,
Fig. 7 Contrarotating
propeller



perspective of cavitation risk and noise minimization.

Propeller Cavitation

Cavitation is the formation of vapor bubbles, which is the consequence of low local pressure level whenever local flow velocities become extreme due to Bernoulli's principle. This phenomenon can be explained as the process of nucleation in a liquid when the pressure drops below the vapor pressure (Pérez-Arribas and Pérez-Fernández 2018). After the pressure again increases to above the critical value, the vapor bubbles that formed around the nuclei begin to implode.

These vapor bubbles collapse within an extremely short time and can cause shock waves against the blades. It is the collapse of the bubbles that can physically damage the propeller blade surface due to very high energy influence and excite vibrations and noise due to pressure variations.

To control cavitation and improve the performance of ships, there are material and industrial ways. Using proper alloys which has great cavitation resistance (high flexibility and strength) can sufficiently reduce risk of cavitation. For industrial process, introducing the air-filled rubber membrane can prevent from cavitation to some extent as in Fig. 8. As it is called, this membrane placed directly after the propeller in the hull uses a pocket of air to prevent cavitation. The large substantial of air bubbles sprayed out over the propeller can be approximated as a single macro

bubble. Since the different stiffness characteristics of air and the small mass of seawater adjacent to the bubble, an isolate zone develops that has the ability to reduce the resonance frequency, which can raise the point at which cavitation is encountered. Thus, the probability of cavitation can be reduced during operation of a marine propeller.

Propeller Characteristics

The thrust, T , and torque, Q , depend on the propeller's diameter, D , revolutions, n , and speed of advance, V , together with the character of the fluid in which the propeller is operating and gravity. Nondimensional relationships between those factors can be expressed as:

$$T = f(D, V, n, \rho) \text{ and } Q = f(D, V, n, \rho) \quad (7)$$

Moreover, according to ship principle, if the engine power and ship velocity are known when the propeller is operating in open water, the propeller thrust T and torque Q can be calculated utilizing the nondimensional coefficients:

$$\begin{aligned} T &= K_T \rho n^2 D^4 \text{ (N)} \text{ and } Q \\ &= K_Q \rho n^2 D^5 \text{ (N} \cdot \text{m)} \end{aligned} \quad (8)$$

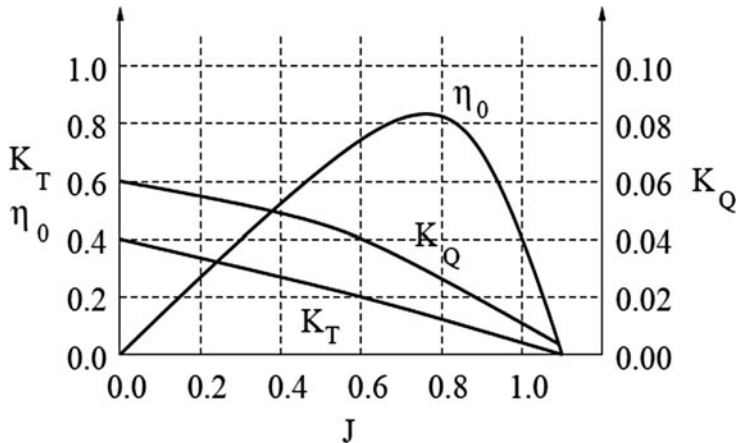
where K_T and K_Q represent propeller thrust coefficient and torque coefficient, respectively, which can be found in propeller data; see Fig. 9. ρ is the density of seawater; n is the propeller rotation

Ship Propulsion System,

Fig. 8 An example of propeller cavitation (Pérez-Arribas and Pérez-Fernández 2018)



Ship Propulsion System,
Fig. 9 An example of the
 K_T - K_Q - J chart



speed; and D is the diameter of the propeller. Propeller efficiency is defined as follows:

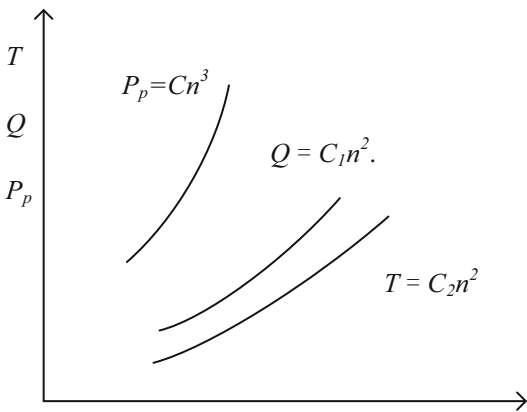
$$\eta_0 = \text{power output/power input}$$

$$= \frac{K_T}{K_Q} \cdot \frac{J}{2\pi} \quad (9)$$

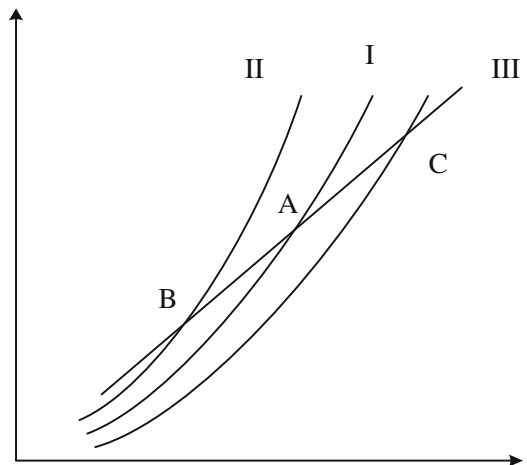
For a propeller with definite geometry, thrust coefficient K_T and torque coefficient K_Q depend on advance ratio J . The varied relationship between these three parameters can be measured in experiment, as shown in Fig. 9.

Thrust and torque coefficients are typically nearly linear over a range of J . When propeller is working in a stable condition, $J(=h_p/D=V/nD)$ is constant and so are K_T and K_Q . The diameter of propeller D and density of water ρ remain unchangeable. Thus, as shown in Fig. 10, the thrust force $T \propto n^2$ and the torque $Q \propto n^2$. The power of propeller can be computed by $P_p = 2\pi nQ$. Hence, it is concluded that $P_p \propto n^3$.

The optimal matching of propeller rate, torque, and power to the main engine is very critical, which may limit the engine working under safe condition preventing it from overload. Diverse propellers with different diameter D and pitch P have different characteristic, which can determine how propeller and main engine work. In Fig. 11, it is hypothesized that the main engine runs at rated power P_b and rated engine speed n_b . The curve of selected propeller characteristic



Ship Propulsion System, Fig. 10 An example of the
propeller characteristics curves



Ship Propulsion System, Fig. 11 Matching of propeller
to main engine

following $P_p \propto n^3$ rule intersects rated power curve at point A. The engine operates stably at condition of point A; meanwhile the propeller achieves complete power output from main engine, represented by the curve I. If the ship encounters severe weather, hull roughness, or propeller roughness which augment the resistance, the propeller characteristic will shift to the left of the curve, such as the line on the curve II. Thus, the operating point moves from A to B which consequently harms the engine since the torque exceeds the limit. It should be noted that, in this situation, the consequences to the engine will result in power loss. The measures such as slowing down the engine need to be taken to prevent damage caused by overloading. Conversely, when assumed that encounters calm water, namely, light load such as in ballast, it is general that the propeller characteristic will move to right, represented by the curve III, and in this condition, the power and speed of engine both exceed rated values P_b and n_b , in which the engine also overload. The same measurement needs to be taken to ensure the engine functions properly.

Cross-References

- [New Technologies in Auxiliary Propulsions](#)
- [Technical and Economical Barriers on Green Energy Utilization in Shipping](#)

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Ship Recycling

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Definition

“Ship recycling” means the activity of complete or partial dismantling of a ship at a ship recycling facility in order to recover components and materials for reprocessing and reuse, while taking care of hazardous and other materials, and includes associated operations such as storage and treatment of components and materials on site, but not their further processing or disposal in separate facilities (International Maritime Organization 2009).

The ship recycling industry is vital for both the shipping industry and the waste recycling industry. On one hand, it helps renew the global fleet of ships and balance the demand and supply of ships in the freight market; on the other hand, it recycles millions of tons of scrap material and contributes to the sustainability. Ship recycling is a complex industrial activity which is required to get rid of end-of-life ships that cannot be used for maritime transportation for various reasons (Jain and Pruyu 2017).

Scientific Fundamentals

Ship recycling is a highly polluting industry. Hazardous chemicals such as asbestos, PCBs, and heavy metals are released during the ship's dismantling and endanger both the environment and the personnel. Moreover, since companies rarely clean end-of-life ships prior to ship recycling, the ships often also exude toxic oils directly into the ocean. And, accidents are very prone to occur in the process of ship dismantling. It needs laws to ensure the implementation of HSE management system.

Contemporary International Law

An integral component of the shipping business, ship recycling is also a global industry. Several international hazardous waste management instruments do great impact on the ship recycling industry. The Basel Convention on the Control of Transboundary Movements of Hazardous Wastes and Their Disposal (1989), the primary international legal hazardous waste management law, as well as the Basel Ban and the Technical Guidelines for the Environmentally Sound Management of the Full and Partial Dismantling of Ships govern the transboundary movement of ships for recycling (Puthucherril 2010).

Elements of the 1982 United Nations Convention on the Law of the Sea (LOS Convention) are also relevant to shipbreaking. The International Labour Organization (ILO) has established standards to protect workers for occupational safety and health. Several IMO conventions on shipping and maritime safety are also apposite to ship recycling. These include the Convention on the Prevention of Marine Pollution by Dumping of Wastes and Other Matter (1972) (London Convention) and its 1996 Protocol, the Nairobi International Convention on the Removal of Wrecks (2007), the International Convention on the Control of Harmful Anti-fouling Systems on Ships (2001), the International Convention for the Control and Management of Ships' Ballast Water and Sediments (2004), and the International Convention for the Prevention of Pollution from Ships (1973) and its Protocol of 1978.

The most important among the IMO instruments having a direct bearing on ship recycling prior is the Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships, 2009. The Convention embraces the "cradle to grave" concept for the purpose of addressing all environmental and safety aspects relating to ship recycling. It takes into account all aspects, from the ship's design stage onward to the end of the ship's life. It also includes the responsible management and disposal of associated waste streams in a safe and environmentally sound manner.

In December 30, 2013, the new EU new shipbreaking act came into effect. The bill applies to all existing ships, new ships, supporting products,

non-EU flag vessels moored at EU ports, and the dismantling of EU vessels. Many of its provisions are higher than the Hong Kong Convention. As per this regulation, all ships entering European Union (EU) ports as well as EU-flagged ships should have an inventory of hazardous materials (IHM). The requirements for an IHM are detailed than for the Hong Kong Convention's IHM, specifically concerning accuracy and comprehensiveness (The European Parliament and Council of the European Union 2013).

Ship Recycling Process

The entire process of ship recycling can be divided into three main phases or sub-processes, i.e., pre-cutting phase involving various surveys and other preparations for gas cutting, cutting phase where actual cutting of steel hull and machinery takes place, and finally the post-cutting phase involving sorting and segregation of materials (Du et al. 2018).

The pre-cutting sub-process comprises all the activities of the ship recycling process that take place before the actual cutting of an end-of-life ship starts. The pre-cutting sub-process can be opened up into various sub-processes such as removal of loose items; removal of liquids; removal of hazardous materials; removal of insulation, tiling, and flooring; and removal of cables and electrical equipment. Though vessel survey is also a part of pre-cutting process, it is not shown in the process diagrams because they are focused at showing the material flow through the ship recycling process.

The cutting sub-process is divided into primary cutting and secondary cutting. The primary cutting is the process where ship's hull is cut into ferrous blocks and nonferrous items such as zinc anodes, copper pipes, valves, etc. are taken apart. During primary cutting, ship's machinery is cut from the base either to be sold in the secondhand market or to be fed as an input for secondary cutting, if it is of no value to the secondhand market. The process of primary cutting is executed using mainly gas cutting torches.

The post-cutting sub-process comprise "pick-up and storage," "separation," and "segregation and transport" sub-processes.

The specific steps can be divided into seven steps.

Step One: Vessel Survey

Diagrams of all compartments, tanks, and storage areas are used (or prepared, if not available) to identify areas that may contain hazardous materials such as fuels, oils, asbestos, PCBS, lead, and other hazardous wastes. Sampling is carried out to enable recycling facility to decide disposal options of certain items containing hazardous materials.

Step Two: Removal of Fuel, Oil, and Other Liquids and Loose Items

Next step is the removal of loose items and fuels, oils, and other liquids (e.g., bilge and ballast water) from the ship. During the vessel scrapping process, water may accumulate due to rain, firefighting activity, or use of hot work cooling water and will have to be properly removed. Loose items include furniture, lifesaving equipment, etc.

Step Three: Removal of Asbestos, PCBS, and Other Hazardous Materials

Both asbestos-containing materials (ACMS) and PCBS are usually removed in two stages. First prior to cutting away a section of the vessel, from areas that are to be cut as and when accessible and then after the vessel section has been moved to shore, remaining ACMS and PCBS can be removed.

Step Four: Removal of Insulation, Cables, and Equipment

Fixtures, anchors, chains, and small equipment are removed first. Large reusable components (e.g., engine parts and other machineries) are removed as they become accessible. Propellers also may have to be removed so that the hull can be pulled into shallow water insulation and cable is also removed before cutting commences.

Step Five: Surface Preparation for Cutting

Before cutting the metal, the surfaces require specific cutting preparation. The paint or preservative coatings are stripped from surfaces by grit blasting or flame removal, in order to prevent exposing

workers to toxic metals and volatile components of paint.

Step Six: Metal Cutting

During the cutting phase, the upper decks, superstructure, and systems are cut first, followed by the main deck and lower decks. Metal cutting is usually done manually using oxygen-fuel cutting torches but may be done with shears or saws for nonferrous metals. Typically, as large parts of the vessel are cut away, they are lifted by crane to the ground where they are then cut into specific shapes and sizes required by the foundry or smelter to which the scrap is shipped.

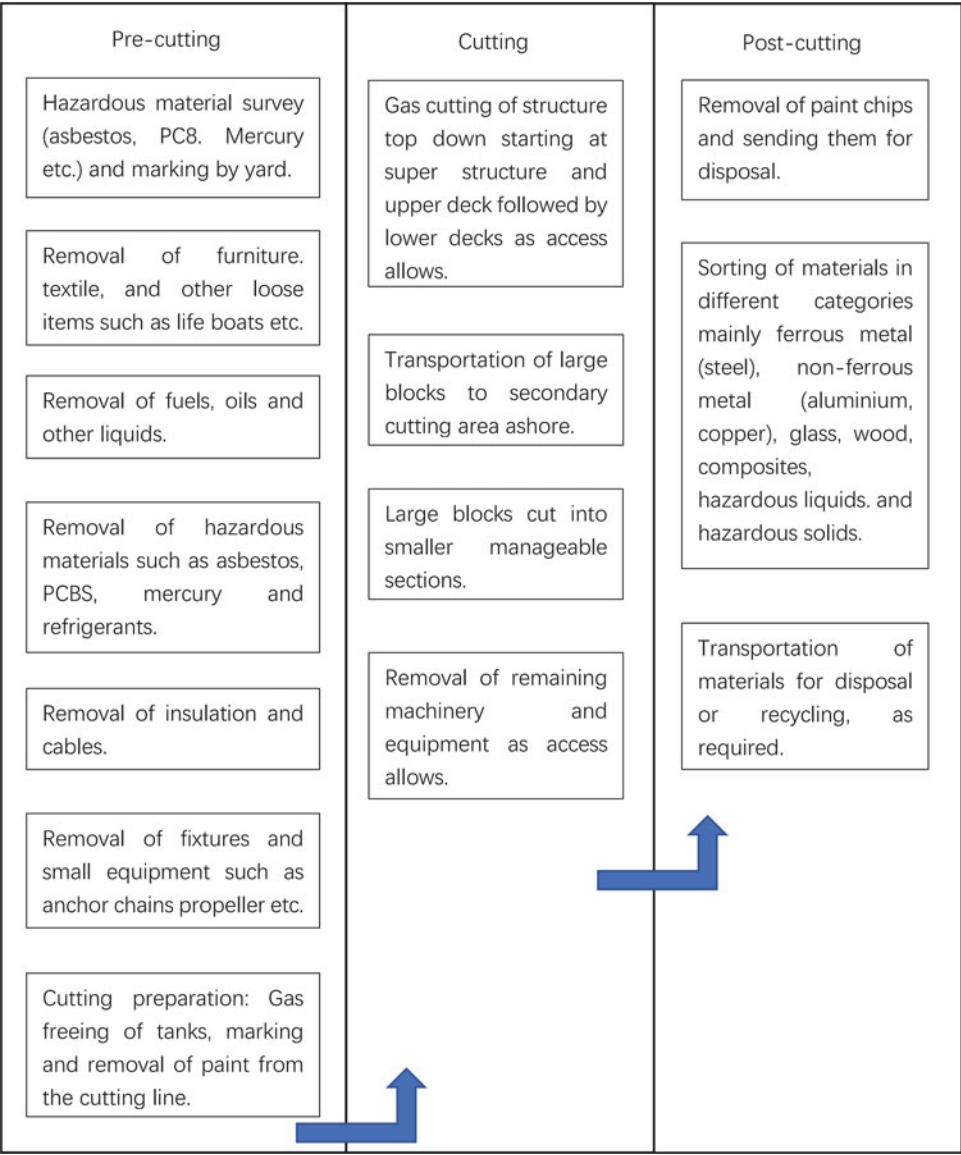
Step Seven: Resale, Recycle, or Disposal

During the post-cutting phase, various material streams from the ship are segregated for resale, recycle, or disposal.

In a word, the entire process of ship recycling can be represented by the following figure (Fig. 1).

Problems of Ship Recycling

The process of ship recycling is essentially the recycling of ship materials. The recycling process may lead to inevitable problems threatening the safety and health of the workers, as well as polluting the ambient environment. Especially in some developing countries, the working conditions on many beaching yards are very poor. Workers hardly wear shoes or safety clothes; they are unskilled and untrained and have to face dangerous situations like fire, explosions, exposure to toxic materials, heights, and falling debris. Accidents causing injuries, permanent disabilities, and even fatalities are a common phenomenon. In short, these workers risk their lives for a low salary. The environment is also affected due to the way these ships are dismantled. Because the ships are scrapped on the beach, all sorts of harmful liquids and heavy metals enter the soil and the sea. The effect on the environment is huge on some of the recycling yards. These issues are collectively called as health, safety, and environmental (HSE) problems.



Ship Recycling, Fig. 1 The entire process of ship recycling

Improvement of Ship Recycling

In order to deal with these problems of ship recycling, design for recycling can be a great way. Instead of only looking at altering the process of recycling a ship, design for recycling addresses the ship’s entire lifetime. The idea is to implement changes to the design of a ship, which make recycling easier and safer.

The three most promising ideas are “reverse production”, “accommodation modularization”, and “resigning the fuel tanks”.

The first concept is “reverse production.” The idea of this concept is to reverse the section building process in ship recycling. The ship must be divided and dismantled in blocks. In the design and engineering phase, the dismantling blocks in

which the ship must be divided during the recycling phase must be selected. The concept also introduces lifting supports which must be fixed to each dismantling block during the building phase itself. The dismantling blocks must be divided in such a way that they do not exceed a weight of more than 500 tons. This is the minimum crane capacity assumed to be available at the shipbreaking yards.

Recycling of the accommodation has always been problematic. In the past, asbestos was commonly used; this is prohibited nowadays. Though, toxic heavy metals are still used which endangers the health of workers. Another problem is that only few materials from the accommodation are recycled, affecting the environment. The concept of modular accommodation builds upon the existing research done by Fikkert, Otheguy, and Durmisevic on modular accommodation on off-shore platforms. In this case, the accommodation is designed to be recycled and takes the recyclability of the materials into account.

Redesigning the fuel tanks is the third and last concept. One of the most dangerous jobs during ship recycling is cleaning and scrapping the tanks. The heavy fuel sludge has to be manually removed by the yard workers. Because the fuel tanks are confined spaces, with a lot of small and sharp corners, this is a difficult process. This concept focuses on creating an easier and safer cleaning process. A fuel tank is redesigned with corrugated bulkheads and trapezoid stiffeners to reduce the amount of difficult-to-clean places and to improve the ease of cleaning. An addition to the physical redesign of the fuel tank is an automatic cleaning system. The tank is designed with appropriate manholes, to enable the use of such a system. This machine works best in wide spaces, which are created in the new tank design. Because of this automatic cleaning system, no workers have to clean the tank manually, which reduces some health and safety issues significantly.

Key Applications

An activity as old as shipbuilding, ship recycling is an essential and integral component of the

shipping business. Once they have outlived their use, ships have been subjected to the axe to recover useable materials. However, with the advent of large modern ships with diverse structural complexity, the ship recycling industry became complicated and dangerous. Till the 1970s, large shipyards in the USA and in Europe performed ship recycling. With mounting economic costs for safe demolition of ships, industrialized countries found a convenient way to dispose of obsolete ships – pack them off to the developing world where there is a great hunger for steel for development activities.

Initially, the industry relocated to Korea and Taiwan Province of China. By the early 1990s, realizing the dangers which the activity entailed, they followed the West in discouraging ship recycling. Thereafter, business gravitated primarily to the beachheads of Third World countries, namely, India, Bangladesh, and Pakistan.

A ship can be recycled by employing different types of ship recycling methods in different parts of world. The methods are similar in most aspects wherein ships are cut apart to retrieve materials and components. The major difference between various methods is firstly by the way ships are docked and secondly by the level of mechanization employed. There are four general methods to dock ships for recycling, i.e., beaching, slipway, alongside, and dry dock. The level of mechanization can be characterized as the non-mechanized process, the highly mechanized process, and the intermediate process. The non-mechanized process is a labor-intensive process which uses a strong labor force and is generally found in the Indian subcontinent recycling yards. The highly mechanized process uses a small amount of labor force and is found solely in Western countries. The intermediate process uses some equipment but still maintains a significant amount of labor force and is generally found in Turkey, China, or even at some dismantling facilities in the USA.

General Methods to Dock Ships for Recycling

Beaching

Beaching is the term generally used for dismantling ships at grounded conditions on the intertidal

zone. Ships are run ashore, as far up the beach as possible, at high tide to leave them grounded at low tide. Ships often don't make it to the beach and are stranded on the mudflats. They are then pulled higher onto the beach using chains or heavy steel wires attached to large winches on the beach. As steel blocks and other equipment of a ship are progressively cut in the intertidal zone using cutting torches, ships become lighter which are then pulled up the beach by winches. Large blocks may also be cut from the ship, released onto the mudflats, and dragged individually by the winches on to the shore. Once on shore, everything is cut into smaller pieces as required by end buyers.

The beaching method is used to dismantle about two-thirds of the world's end-of-life ship capacity, i.e., 66% in terms of gross tonnage. The main locations include Chittagong in Bangladesh, Alang in India, and Gadani in Pakistan. These areas have a very large tidal difference and extensive mudflats which are utilized to drive ships up the beach. The beach is generally divided into plots of about 50 m wide and up to 100 or 150 m deep. A major issue with dismantling ships on tidal mudflats is that any spills of oil or cargo remaining on board are likely to be swept out to sea by the next tide.

Slipway

This method is a modification to the beaching method and is also called as nontidal beaching. The major difference between beaching and the slipway method is that of tide. It is practiced in areas with a low tide difference, especially in Turkey. Other than Aliaga in Turkey, slipway recycling is practiced in many small-scale historical recycling locations such as Inverkeithing in the UK and other locations in Europe and USA today. In the USA slips are generally 400–700 ft long and 100–120 ft wide at the entrance.

The ship is still beached against the shore or, preferably, a concrete slipway extending to the sea. In either case, the lack of racing tides provides an element of control which means that any accidental spillages can reasonably be contained. It is generally acknowledged that the low tidal difference and improved access to the hull and the

working area offer advantages for safe and environmentally sound operations compared to the beaching method. About 4% of the world's recycling capacity uses nontidal beaching method for ship recycling at Turkey.

Normally pieces are removed from the ship by mobile crane working from the shore. In this method also, the ship is dragged up the shore as it is lightened but because the tide is constant and a permanent, predictable, and stable waterfront exists where the lifting and access operations take place.

Alongside

The alongside method, also referred to as quayside, pier side, or floating method, is a method to dismantle ships that are afloat and moored along wharfs, jetties, or quays and/or moored offshore. Cranes and either automated cutting gear such as mechanical shears or gas cutting torches are used to reduce the ship in a planned and structured manner. The process is top down: the superstructure and upper pieces are removed first, and then the work continues along the ship into the engine room until only the double bottom is left. This last part of the ship, an empty floating hull called the canoe, is reduced to the extent possible while afloat and then either taken out as a whole or further cut into pieces in a dry dock. This method is mainly practiced in China, the USA, and Belgium. China provides about 28% of the world's recycling capacity in terms of end-of-life ships' gross tonnage.

During alongside recycling, local impact of any pollution is likely to be increased since there is no tidal dispersal effect. However, this means that concentrations can be properly monitored, contained, and cleaned if necessary.

Dry Dock

In this method ships are dismantled at a dry dock, floating dock, or a slipway which has a lock gate and an impermeable floor structure. This method is the safest and cleanest way of recycling a ship because chances of polluting surrounding waters by accident are virtually nil as everything is contained within the dock. The dock is cleaned before it is flooded for dismantling the next ship in order to avoid accumulations of contaminants.

The only downside with this method is that it is the most costly method of recycling a ship which makes it most scarcely used method. One of the main dry-dock recycling locations is Leavesley International's facility in Liverpool in the UK.

Cross-References

- [Beaching](#)
- [Complete and Partial Dismantling](#)
- [Inventory of Hazardous Materials](#)
- [Recycling Regulations](#)
- [Ship Recycling Facility Plan](#)
- [Ship Recycling Plan](#)

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Definition

The Ship Recycling Facility Plan (SRFP) is the main document relied upon by Competent Authorities (or an organization recognized by them) when authorizing a Ship Recycling Facility. The convention and subsequent regulations for the SRFP are in place to ensure that recycled ships pose no threat to either human life or the environment during or after they have been recycled.

Scientific Fundamentals

In accordance with the 2011 Guidelines for the Development of the Ship Recycling Plan under the Hong Kong Convention, and the EU Ship Recycling Regulations, ships that are to be recycled require a Ship Recycling Plan, provided by the Ship Recycling Facility, who must also evidence a plan for how they safely dispose of the ships under their Ship Recycling Facility Plan (SRFP).

The plan should fully describe the operations and procedures that are in place at the Ship Recycling Facility to ensure compliance with the Hong Kong Convention and the EU Ship Recycling Regulations.

The SRFP should also demonstrate knowledge and understanding of

- All applicable statutory and regulatory requirements.
- A strong commitment to worker health and safety and protection of the environment.
- The operational processes and procedures involved in ship recycling.
- How the requirements of the Convention will be met.

An SRFP should also provide information regarding the organizational structure and management policies of the Recycling Company, an overview of the Ship Recycling Facility and their methodologies relating to shipping recycling.

Ship Recycling Facility Plan

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Synonyms

[Ship Recycling Facility Plan \(SRFP\)](#)

Requirements for Recycling Facilities

Recycling facilities authorized to recycle ships will be expected to produce a ship-specific Ship Recycling Plan on the basis of information provided by the shipowner (IHM, ship's plans and manuals), taking into account the guidelines of the HKC.

Recycling facilities will also be required to produce and operate in accordance to a Ship Recycling Facility Plan that is developed considering the relevant guidelines of the HKC.

To be authorized, recycling facilities will have to comply with the provisions of the HKC:

1. "Operate from build structures"
2. Demonstrate "the control of any leakage, in particular in intertidal zones".
3. Ensure "the handling of hazardous materials, and of waste generated during the ship recycling process, only on impermeable floors with effective drainage systems".

These three requirements are products of the negotiations between the European Council and the Parliament. A key policy of the Parliament was to achieve an outright ban to beaching. The second and third of the above three provisions prescribe requirements that will have to be met regardless of the recycling method used, while the first requirement for operating "from build structures" is imprecise and ambiguous and is a compromise devised to accommodate political sensitivities in the negotiations.

Since the conclusion of the negotiations, the Rapporteur of the Parliament has chosen to treat the outcome as confirmation of a ban to beaching. He issued in July 2013 a celebratory press release, stating: "These standards effectively mean the end of beaching where ships are simply taken apart on a beach, with scant regard for human health or the environment. . . . The new law will make it compulsory for ships to be recycled from built structures only and in such a way that all hazardous materials are fully contained. As this is impossible on a beach, the practice of beaching is de facto forbidden." More recently, on the occasion of the formal adoption of the ER by the Parliament on

22/10/2013, a further press release stressed that: "Plans agreed with EU ministers to end the scrapping of old EU-registered ships on third-country beaches . . .", and also: "During the negotiations, Parliament strengthened the proposed requirements, inter alia by obliging ship-recycling businesses to operate in built structures . . .".

Reportedly, the European Council seems to have understood that it would be an absolute blunder to exclude the South Asian ship recycling market from the ER and from the options available to European shipowners. Moreover, ship recycling experts are very clear that there is no contradiction between good quality beaching, safety, and protection of the environment. Apparently, it was for these reasons that the Council refused to agree to ban beaching. Such a ban would lead to a new massive evasion of the ER by owners and probably to a total disinterest by South Asian countries to consider acceding to the HKC, thus torpedoing its prospects of entry into force. Ironically, the ban for which the Greens are fighting, would only serve the commercial interests of substandard recyclers and of the Shipbreaking Platform, who, in this way, will continue to have something to do.

The European Commission will implement the ER and will be required to put to practice its requirements. The ER provides the option to the Commission to issue technical guidance notes so as to facilitate the certification of recycling facilities (Art. 15(4)). The Commission is expected to develop such guidance in 2014 and in so doing to interpret the meaning of "from build structures." The ER, like the HKC, requires the recycling facility to formally report its readiness for the commencement and also the completion of recycling of each ship under ER's scope. Interestingly, whereas the HKC requires the facility to report to its competent authority, ER requires the facility to report to the administration of the flag State, as Europe has no jurisdiction over competent authorities of recycling States outside the European Union.

There is one further important, but understandable, difference between ER and the HKC. Whereas according to the HKC, the authorization of recycling facilities is a matter for the competent

EXAMPLE FORMAT OF FACILITY INFORMATION IN SRFP

Facility name and contact information			
Facility name			
Registered address			
Facility address			
Representative and communication address			
Number of employees			
Tel. No.		Fax No.	
E-mail address		URL	
Working language			

Capacity of Facility	
Maximum capacity of ship to be recycled	Length Breadth Width Depth DWT GT LDT
Types of ship to be accepted	
Annual recycling capacity (in LDT)	

Waste management capacity	
Asbestos	removal storage process
Ozone-depleting substances	removal storage process
Polychlorinated biphenyls (PCB)	removal storage process
Anti-fouling compounds and system	removal storage process

Ship Recycling Facility Plan, Fig. 1 (continued)

Cadmium and Cadmium Compounds	removal storage process
Hexavalent Chromium and Hexavalent Chromium Compounds	removal storage process
Lead and Lead Compounds	removal storage process
Mercury and Mercury Compounds	removal storage treatment process
Polybrominated Biphenyl (PBBs)	removal storage treatment process
Polybrominated Diphenyl Ethers (PBDEs)	removal storage treatment process
Polychlorinated Naphthalenes (more than 3 chlorine atoms)	removal storage treatment process
Radioactive substances	removal storage treatment process
Certain Shortchain Chlorinated Paraffins (Alkanes, C10-C13, chloro)	removal storage treatment process
Hazardous liquids, residues and sediments	removal storage treatment process
Paints and coatings that are highly flammable and/or lead to toxic release	removal storage treatment process
Other Hazardous Materials not listed above and that are not a part of the ship structure (specify)	removal storage treatment process

Ship Recycling Facility Plan, Fig. 1 Example Format Of Facility Information in SRFP

authorities of the recycling State, the ER could not expect the authorities in countries outside the European Union to enforce the ER, and for this reason it has developed the new mechanism of the “European List of ship recycling facilities”, Fig. 1 is example format of facility information in SRFP.

Facilities located in a European Member State will be authorized by the competent authorities of that Member State who will then notify the Commission of the facilities that it has authorized. The Commission will then include these facilities in the European List that will establish and publish in the Official Journal and on its website no later than 36 months after the ER’s entry into force. On the other hand, facilities located in third countries will have to apply individually to the Commission for their inclusion in the European List. Together with their application, they will provide evidence showing that they meet the requirements of the ER. The compliance of individual facilities “shall

be certified following a site inspection by an independent verifier with appropriate qualifications.” Furthermore, the facilities will have to “accept the possibility of being subject to a site inspection by the Commission or agents acting on its behalf.” The renewal of the certification and inclusion in the European List will take place every 5 years, subject to a mid-term review confirming compliance.

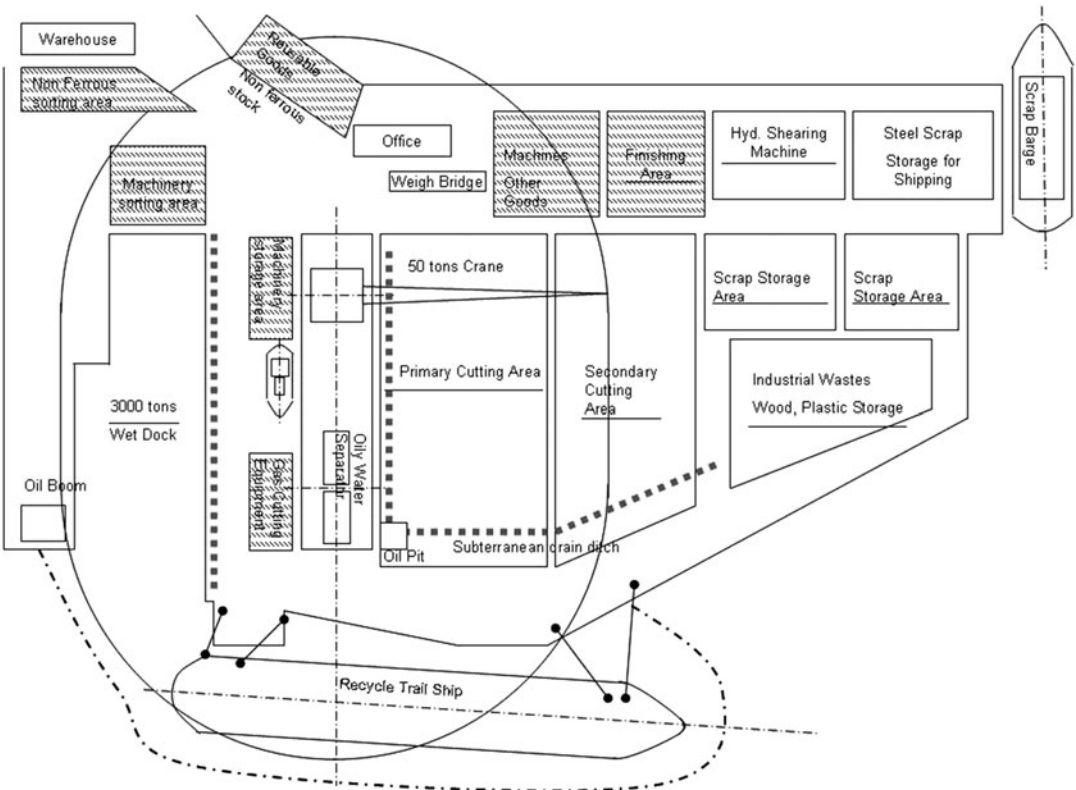
Key Applications

Example Format of Facility Information in SRFP

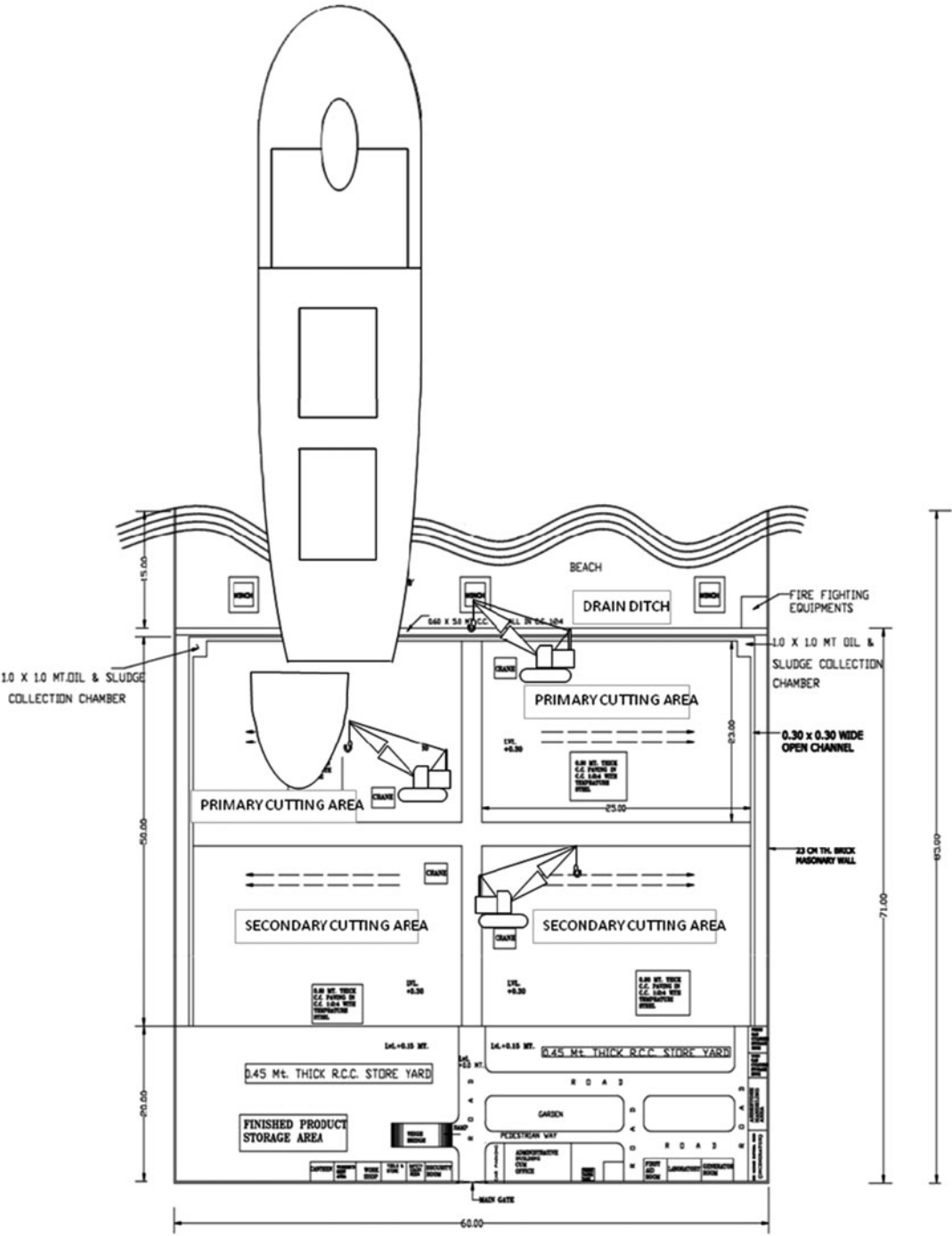
Location Map

Yard plan (examples)

Yard plan should be included in Facility information, such as Figs. 2 and 3.



Ship Recycling Facility Plan, Fig. 2 Site Layout Example



Ship Recycling Facility Plan, Fig. 3 Site Layout Example

Cross-References

- [Recycling Regulations](#)
- [Ship Recycling](#)
- [Ship Recycling Plan](#)

References

- ISO 30000:2009 Ships and marine technology – ship recycling management systems – specifications for management systems for safe and environmentally sound ship recycling facilities
- ISO 30002:2010 Ships and marine technology – ship recycling management systems – guidelines for selection of ship recyclers (and pro forma contract)
- ISO 30003:2009 Ships and marine technology – ship recycling management systems – requirements for bodies providing audit and certification of ship recycling management
- ISO 30006:2010 Ship recycling management systems – diagrams to show the location of hazardous materials onboard ships
- ISO 30007:2010 Ships and marine technology – measures to prevent asbestos emission and exposure during ship recycling
- ISO/AWI 30008 Ship recycling management systems – yachts recycling
- ISO/AWI PAS 30001 Ship recycling management systems – best practice for ship recycling facilities – assessment and plans
- ISO/CD 30004 Ship recycling management systems – guidelines for implementing ISO 30000
- ISO/PAS 30005:2010 Ships and marine technology – ship recycling management systems – information control for hazardous materials in the manufacturing chain of shipbuilding and ship operations
- Yu J, Jin W, Tan Z et al (2017) Development and experiments of the Sea-Wing 7000 underwater glider. Oceans–Anchorage, Anchorage

Ship Recycling Facility Plan (SRPP)

- [Ship Recycling Facility Plan](#)

Ship Recycling Plan

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Synonyms

[Ship Recycling Plan \(SRP\)](#)

Definition

A Ship Recycling Plan is a living document detailing the specific recycling process information relating to a specific marine vessel.

In accordance with the 2011 Guidelines for the Development of the Ship Recycling Plan under the Hong Kong Convention, and the EU Ship Recycling Regulations, ships that are to be recycled require a Ship Recycling Plan. Additionally to the regulations, the benefits of ship recycling in reducing human and environmental impacts include:

- Recycling reduces the need for mining – an environmentally damaging practice.
- Some materials can be recycled infinitely.
- Contributes to the sustainable development of the shipping industry whilst minimizing and reducing the current environmental impact on oceans and seas.
- Deters the practice of “beaching” and protects workers from dismantling without using PPE.
- Control, monitor, and reduce the impact of IHMs on local environments that disrupt biodiversity and can affect the surrounding environments impacting on wildlife, farming, and communities.
- To support our clients in meeting the regulatory requirements our team produce internationally recognized Ship Recycling Plans and

support you through the process with expert advice and guidance.

Scientific Fundamentals

Ship recycling means the activity of complete or partial dismantling of a ship at a ship recycling facility in order to recover components and materials for reprocessing and re-use, while taking care of hazardous and other materials, and includes associated operations such as storage and treatment of components and materials on site, but not their further processing or disposal in separate facilities. A Ship Recycling Plan shall be developed by the Ship Recycling Facility(ies) prior to any recycling of a ship, considering the guidelines developed by the Organization. The Ship Recycling Plan shall:

1. Be developed considering information provided by the shipowner.
2. Be developed in the language accepted by the Party authorizing the Ship Recycling Facility.
3. Include information concerning inter alia, the establishment, maintenance, and monitoring of safe-for-entry and safe-for-hot work conditions and how the type and amount of materials including those identified in the Inventory of Hazardous Materials will be managed.
4. Should be either explicitly or tacitly approved by the Competent Authority authorizing the Ship Recycling Facility. The Competent Authority shall send written acknowledgment of receipt of the Ship Recycling Plan to the Ship Recycling Facility, Ship Owner, and Administration within 3 working days of its receipt.
5. Once approved in accordance with paragraph 4, be made available for inspection by the Administration, or any nominated surveyors or organization recognized by it.
6. Where more than one Ship Recycling Facility is used, identify the Ship Recycling Facilities to be used and specify the recycling activities and the order in which they occur at each authorized Ship Recycling Facility.

Framework of SRP

The responsibility for developing a comprehensive SRP rests with the Ship Recycling Facility,

although development of the SRP is a cooperative effort between the Ship Recycling Facility and the shipowner. The Ship Recycling Facility is best placed to understand and describes the methods and procedures that it uses in its recycling operations and it has knowledge of the available facilities and capabilities for the safe and environmentally sound management of all hazardous materials and wastes generated during recycling, of the skills and capabilities of its workforce and the availability of local support services, and of the relevant national laws and regulations that apply to the facility and its activities, including the activities which it is approved to perform under its DASR. A sample cover page for the SRP is included in the appendix. The body of the SRP should include a more detailed narrative of the ship-specific recycling elements.

The SRP should describe how the Ship Recycling Facility will recycle the specific ship in a safe and environmentally sound manner, covering the recycling process steps and their sequence over the entire process. Any processes or procedures that deviate from the SRFP and are specific to the ship should be described in detail in the SRP.

Where more than one Ship Recycling Facility is used, SRPs should be prepared separately, in principle, by each of the Facilities involved, according to their respective duties and indicate the order in which the activities will occur.

1. Prearrival Elements

The SRP should include a description of any specific preparatory work that should be carried out. The SRP should clarify whether and to what extent any preparatory work – such as pretreatment, identification of potential hazards, and removal of stores – will take place at allocation other than the Ship Recycling Facility identified in the SRP. The extent to which such preparatory work will be covered in the SRP will depend upon the capability of the authorized Ship Recycling Facility and the scope of the agreement with the shipowner. In the case of a tanker, the ship should arrive at the Ship Recycling Facility with cargo tank and pump room(s) in a condition that is ready for certification as Safe-for-entry, or Safe-for-hot work, or both.

The Ship Recycling Facility should plan appropriately for the ship's arrival. The SRP should include the location where the ship will be placed during recycling operations and a concise plan for the arrival and safe placement of the specific ship to be recycled.

2. Arrival of Ship

The SRP should describe the procedures that the Ship Recycling Facility will follow to conduct a walk-through (on-board check) of the vessel to identify any potential environmental or safety issues. The Ship Recycling Facility should verify whether safe access and egress have been provided for and that the SRP is in place throughout the ship recycling process. It is recommended that the Ship Recycling Facility should mark the location of the known hazardous materials. Any specific items or locations on board whose hazardous characteristics are uncertain should be marked for additional sampling as necessary.

3. Management of Hazardous Materials

The SRP should include information on how the type and amount of hazardous materials will be managed, as required by regulation 9.3 of the Convention and specify the facility's approach for managing each hazardous material. Special attention should be paid to the types and quantities of hazardous materials on the ship. If ship-specific conditions require deviation from normal practices for managing hazardous materials, the appropriate ship-specific measures should be described in detail in the SRP. In order to avoid confusion, it is recommended that the SRP should use the same nomenclature and identification scheme as those included in the IHM.

The SRP should also contain additional information on the management of hazardous materials as required in Appendix 5 of the Convention (also known as the DASR). Specifically, the SRP should describe where the hazardous materials are to be processed or disposed of if the operation is not being conducted at the Ship Recycling Facility. The SRP should state that the removal of hazardous materials will be undertaken by responsible personnel who are trained and authorized to do so.

4. Safe-for-Entry and Safe-for-Hot-Work Procedures

Regulation 9 of the Convention requires the SRP to include information concerning the establishment, maintenance, and monitoring of safe-for-entry and safe-for-hot-work procedures. The Ship Recycling Facility is encouraged to review the Facility Guidelines, as they contain specific technical recommendations to address these important safety issues.

While the SRFP will describe general procedures on how the Ship Recycling Facility will achieve safe atmospheric conditions during the ship recycling process, the SRP should describe in detail how safe-for-entry and safe-for-hot-work procedures will be implemented on the specific ship, taking account of such features as its structure, configuration, and previous cargo.

5. Dismantling Sequence

An important component of the dismantling sequence is the removal of hazardous materials to the maximum extent practicable prior to and during cutting activities. Depending on a number of factors, including the age of the ship and the quantity of hazardous materials present, it may be impossible to remove all hazardous materials prior to the start of cutting activities. The SRP should include a dismantling sequence that is ship-specific and takes into account the cutting operations and locations of hazardous materials.

6. Other Necessary Elements

In addition to the elements described above, the SRP should include any ship specific processes and/or procedures that will be necessary to recycle the ship and that are not fully covered in the SRFP. For example, a Ship Recycling Facility may need to use additional workers or subcontractors, or they may need additional equipment to deal with unique aspects of the ship. Such ship-specific processes/procedures may take into account the technical guidance manual to be developed by the Organization.

7. Attaching a Copy of DASR

The Ship Recycling Facility should attach a copy of the DASR (Document of Authorization to conduct Ship Recycling) to the SRP (See Fig. 1).

Ship Recycling Plan Summary of information on ship and Ship Recycling Facility

This Ship Recycling Plan was developed in accordance with the Hong Kong International Convention for the Safe and Environmentally Sound Recycling of Ships, 2009 (the Convention).

Ship information

Name of ship	
Distinctive number or letters	
Port of registry	
Gross tonnage	
IMO number	
Name and address of shipowner	
IMO-registered owner identification number	
IMO company identification number	
Telephone number	
E-mail address	

Ship Recycling Facility information

Name of Ship Recycling Facility	
Distinctive Recycling Company identity No.	
Full address of Ship Recycling Facility	
Primary contact person	
Telephone number	
E-mail address	
Name, address and contact information of ownership company	
Working language(s)	

Projected schedule for ship recycling

Date of ship arrival at Ship Recycling Facility	
Date of commencement of ship recycling	
Date of Completion of ship recycling	
Date of completion of sale/disposal of all components	

.....
(Date)

.....
(Signature of Ship Recycling Facility owner/operator)

Cross-References

- Recycling Regulations
- Ship Recycling
- Ship Recycling Facility Plan

References

- Basel Convention technical guidelines on waste oils from petroleum origins and sources. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/old%20docs/tech-y8.pdf>
- Basel Convention technical guidelines on used oil re-refining or other re-uses of previously used oil. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/old%20docs/tech-r9.pdf>
- Basel Convention technical guidelines on specially engineered landfill. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/old%20docs/tech-d5.pdf>
- Basel Convention technical guidelines on incineration on land. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/old%20docs/tech-d10.pdf>
- Basel Convention technical guidelines on hazardous waste – physio-chemical treatment – biological treatment. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/old%20docs/tech-d8d9.pdf>
- Resolution MEPC.196(62). 2011 Guidelines for the development of the Ship Recycling Plan (SRP)
- Technical guidelines for the environmentally sound management of the full and partial dismantling of ships. <http://www.basel.int/Portals/4/Basel%20Convention/docs/meetings/sbc/workdoc/techships-e.pdf>
- Technical guidelines for the environmentally sound management of wastes consisting of elemental mercury and wastes containing or contaminated with mercury. <http://www.basel.int/Implementation/TechnicalMatters/DevelopmentofTechnicalGuidelines/AdoptedTechnicalGuidelines/tabid/2376/Default.aspx>
- Technical guidelines for the environmentally sound management of waste lead-acid batteries. <http://www.basel.int/Portals/4/Basel%20Convention/docs/pub/techguid/tech-wasteacid.pdf>
- Technical guidelines on the environmentally sound recycling/reclamation of metals and metal compounds. <http://www.basel.int/Portals/4/Basel%20Convention/docs/pub/techguid/r4-e.pdf>
- Technical guidelines on the environmentally sound management of biomedical and healthcare wastes. <http://www.basel.int/Portals/4/Basel%20Convention/docs/pub/techguid/tech-biomedical.pdf>
- Training resource pack for hazardous waste management in developing countries. <http://www.basel.int/pub/pub.html>
- United Nations recommendations on the transport of dangerous goods. <http://www.unece.org/trans/danger/publi/unrec/English/Recommend.pdf>
- United Nations Globally Harmonized System for the classification and labelling of chemicals (GHS). http://www.unece.org/trans/danger/publi/ghs/ghs_rev03/03files_e.html
- Updated general technical guidelines for the environmentally sound management of wastes consisting of, containing or contaminated with persistent organic pollutants (POPs). <http://www.basel.int/Portals/4/Basel%20Convention/docs/pub/techguid/tg-POPs.pdf>
- Updated technical guidelines for the environmentally sound management of wastes consisting of, containing or contaminated with polychlorinated biphenyls (PCBs), polychlorinated terphenyls (PCTs) or polybrominated biphenyls (PBBs). <http://www.basel.int/Portals/4/Basel%20Convention/docs/pub/techguid/tg-PCBs.pdf>

Ship Recycling Plan (SRP)

- Ship Recycling Plan

Ship Structural Design

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Synonyms

Hull structure design

Definition

Ship structure design is to select the appropriate structural materials and structural forms and determine the size and connection of all components after the overall design (determining the main dimensions of the ship, the hull profile, and the overall layout of the ship) is completed. Ship structure design is to ensure that the ship has

enough strength, stiffness, stability safety, and other requirements, so that the structure has the best technical and economic performance.

Scientific Fundamentals

Ship Structural Design System

Ship structural design is generally carried out in three stages with the design process of the whole ship: preliminary design, detail design, and production design. Complete different tasks at different stages:

- Preliminary design: According to the approved technical specifications, the overall structural design principles (structural arrangements, structural forms, etc.) are analyzed and compared. Theoretical estimation of the arrangement and size of major components, the typical midship section drawings, and hull steel weight estimates are given.

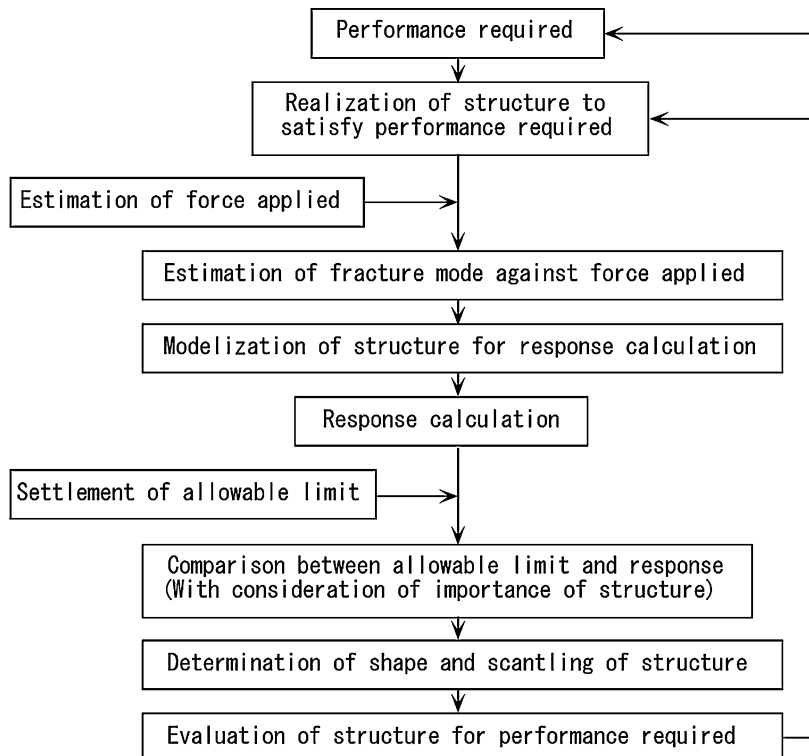
- Detail design: The design is based on the decisions made during the preliminary design. At this stage, the technical problems in the structural design are fully solved, including determining the arrangement, size, and connection of the components. Design drawings and documents to be reviewed are submitted to the classification society. These designs take into account the fitting arrangement and hull block assembly process.
- Production design: According to the detailed design drawings approved by the classification society, all technical preparations for the construction of the hull structure, such as construction details, process measures, and assembly sequence of each component and component connection, are determined.

Design Procedure of Structures

Not only the ship structural design but also general structural design is carried out in accordance with the procedure shown in Fig. 1 (Yasuhisa Okumoto et al. 2009).

Ship Structural Design,

Fig. 1 Sequence of structure design (Yasuhisa Okumoto et al. 2009)



The design of the ship needs to meet the requirements for the type and quantity of cargo loaded, navigation area, sailing speed, endurance, or new type of ship. Then a preliminary general arrangement is to be prepared based on these requirements.

The estimation of the forces applied to the structures shown in Fig. 1 means that the ship is placed on a wave and the establishment of static loads including buoyancy, weight of ship, cargo weight, and the estimation of dynamic loads such as wave load and exciting force. The wave load is a related to ship type, wave elements, and calculation status (relative position of ship and wave).

In Fig. 1 the design flow, shown in vertical arrows, is based on standards and research results that are supplied to the designers by outsiders. Since the setting of the forces acting on the ship structure affects the design very much, the ship structural designer should always pay attention to the research of these contents.

In the step of modelization of strength response calculation, the correct structural failure mode should be applied. Not only theoretical knowledge but also observations experienced on actual failures are necessary in estimating the failure modes correctly.

The next step is to calculate response calculation for the forces applied. The traditional calculation assumes that each structural member is a simple beam fixed at both ends. In recent decades, computers can be used to model the entire structure and analyze the estimated forces.

For the allowable limit, there are allowable stress, allowable deformation, allowable amplitudes of vibration, and allowable reliability (Yasuhisa Okumoto et al. 2009). In practical applications, these allowable limits are based on ship design, construction, and operational experience, as well as accumulated static load measurements and voyage test results of actual ships, determined on the base of safety and economy. These limits are usually stipulated by specialized agencies such as the classification society's rules. The ship structural designer should pay special attention to them. Any changes in the method of load estimation must be followed by modifying the allowable limit.

Due to the different effects of structural damage on the hull, different structures have different importance. Therefore, there are different safety factors for different structures.

If the calculated response according to the above steps is within the allowable limit and satisfies reality, the shape and scantling of the structure should be accepted and decided. If these values are not moderate, further factors must be considered and verified repeatedly to make it meet the requirements.

In addition, material that is not depicted in Fig. 1 plays an important role in the design of ship structures. Mild steel for ships that is mainly used in the construction of ships has its excellent properties, toughness, robustness, weldability, and low price. The ship structural designer can use it without any special attention. However, nowadays high tensile steel is becoming more common as a shipbuilding material. When the high tensile steel and other special materials are used as shipbuilding material, the hull structure designer must pay careful attention to these materials.

Structural Design Quality Indicators

At present, the production technology of the shipbuilding industry in the world is developing in the direction of mechanization, automation, and integration. Improving the structural design requirements of ships has become a very important issue in the shipbuilding industry. Ship structure design shall ensure that the ship meets the following requirements when applied:

- Safety: The structure should be able to withstand the various possible loads during normal use and still maintain the necessary overall stability when accidental events occur.
- Operational suitability.
- Overall compatibility: In the design of the ship, the structural design must be coordinated with the design of the general, turbine, electrical, and other aspects to ensure that the ship has good performance in all aspects.
- Durability: Durability is the ability of a structure to maintain its original structural efficiency level. In other words, the structure can

be used for a predetermined number of years under normal maintenance conditions.

- **Manufacturability:** The designed structure must be easy to construct and ensure construction quality while minimizing the amount of construction work to reduce manufacturing costs.
- **Economy:** For civil ships, the general goal of the design is to make the ship the most economical benefit throughout the life of the ship.

Key Technology

Ships are often operated under high environmental conditions such as high speed, strong water flow, and strong air flow. The ship structure design must consider not only the longitudinal strength, local strength, and structural stability of the ship but also vibration, shock, and noise. It can be seen that the ship structure design is a subject area with high technical content and difficult design. In order to properly design the ship structure, it is necessary to understand the characteristics of the hull structure and the loads acting on it. Therefore, determining the ship's load, strength calculation, selection of suitable materials, and connection methods is the key to the ship structure design.

Structural Design Load

When considering the load features where the load is transmitted gradually and continuously from a local structural member to an adjacent bigger supporting member, the best way to categorize loads on the hull structure is as follows (Yasuhisa Okumoto et al. 2009).

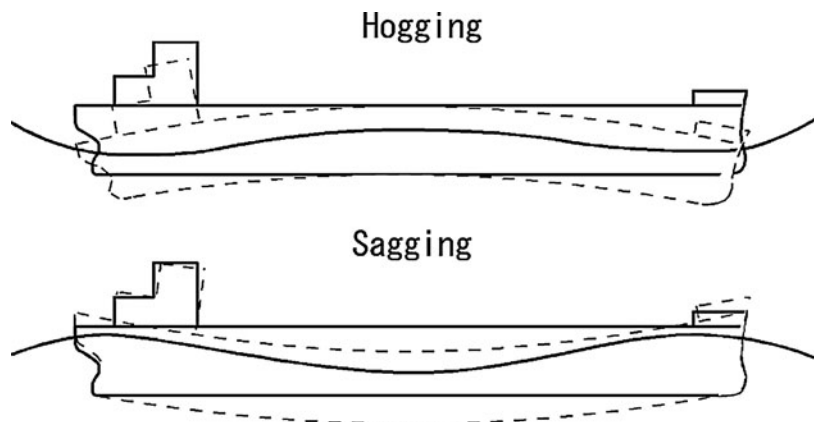
- **Longitudinal strength loads:** Longitudinal strength load means the load concerning the overall strength of the ship's hull, such as the bending moment, shear force, and torsional moment acting on a hull girder. If the longitudinal strength loads exceed the upper limit of longitudinal strength for the hull, the hull will be bent or twisted. Therefore, the longitudinal strength load is one of the most important loads when calculating the overall strength of a hull structure.

Longitudinal strength loads may be divided into two categories: static longitudinal loads and dynamic longitudinal loads. Static longitudinal loads are induced by the local inequalities of weight and buoyancy in the still water condition. For instance, differences between weight and buoyancy in longitudinal direction cause a static bending moment and a static shear force, and asymmetrical cargo loading causes a static torsional moment. Dynamic longitudinal loads are induced varying loads and impact loads by waves. When the ship is on top of a wave crest in head sea condition, it causes a "hogging" bending moment and a shear force. When in a wave trough, a "sagging" bending moment and shear force are experienced, as indicated in Fig. 2.

Usually the "slicing method" which is a linear method for slender hulls is used to calculate the wave-induced shear and bending moments. In the slicing method, the hull is divided into "sheets", and the loads of each of the sheets are calculated, such as additional

Ship Structural Design,

Fig. 2 Vertical bending due to waves (Yasuhisa Okumoto et al. 2009)



inertial forces, wave forces, and Froude forces. The comparison between the result of slice method calculation and the test result shows that the accuracy of the wave load acting on the hull girder can meet the engineering requirements. So far, many scholars have done a lot of research and improvement work on the sectioning method, including nonlinear effects (Jensen and Pedersen 1978), three-dimensional effects of the ship's mast, and so on. Due to the action of waves, structures near the static side of the ship should be submerged or exposed to water, and necessary corrections must be made. Recently, the three-dimensional Green function method and the Rankine source method have been widely used to calculate wave loads in hulls. Newman (1990), Noblesse (1982), and Chen et al. (1995) have done a lot of work in this area and proposed an efficient numerical solution of the Green function. The Rankine source, which is the surface element method that began in aviation (Hess and Smith et al. 1962), is used as the Green function to solve the numerical calculations around the three-dimensional object potential flow.

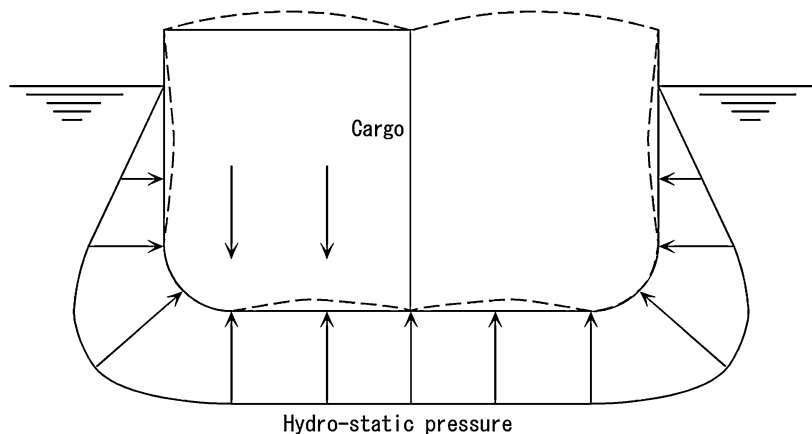
The above methods are to calculate the actual load acting on the hull structure. However, in the normal hull design, the hull structural strength calculation and analysis are carried out according to the loading method specified by the classification society and in

accordance with the loading method prescribed by the classification society.

Wave impact loads can be divided into two loads: (a) impact force induced by slamming and (b) impact force of green seas on the deck. With rough sea conditions, linear theory cannot be applied, since the calculated results of ship motion and dynamic loads are influenced by the wave impact force and/or by the nonlinear effect of the vertical change of ship breadth at each cross section. These impact loads can contribute to the increase of longitudinal loads, for they may generate both an additional bending moment and shear force in the hull girder. Recently, many scholars have published articles using finite element, boundary element method, and finite difference method to calculate impact load, but the actual application is far from enough.

- Transverse strength loads: The transverse strength loads represent the loads which act on transverse members and cause structural distortion of a cross section. Transverse strength loads include hydrostatic pressure on the outer shell, weight of cargo load working on the bottom structure, ballast water pressure inducing the deformation of the ballast tank, etc. For instance, let's imagine a transverse section of a ship floating in still water as illustrated in Fig. 3. This section is subjected to (a) hydrostatic pressure due to surrounding water and (b) internal loading due to self-weight and cargo weight. These loads are not

Ship Structural Design,
Fig. 3 Example of deformation due to transverse strength loads (Yasuhisa Okumoto et al. 2009)



always equal to each other at every point, consequently loads working on transverse members will produce transverse distortion as shown by the broken line in Fig. 3.

- Local strength loads: The local strength loads, which include loads that affect the local strength members such as shell panels, stiffeners, and connecting constructions between stiffeners, is mainly the pressure acting on the shell plating and the load transmitted to the internal structure. The other important load is the wave slamming load, especially for the high-speed ship's crotch structure.

Strength Evaluation

The ship structure has been planned originally by the designer in such a way that the structure can effectively resist the presumed maximum load, which is estimated from previous experience of failure modes of damage. When the designer starts a ship structure design, the very first step is to establish the magnitude, direction, probability of occurrence of the load, and the structural failure modes experienced in the past.

- Determine an initial system of structural members.
- Presume a magnitude, direction, and probability of load.
- Assume a failure mode of structure due to the load.
- Select an appropriate analysis method.
- Choose acceptable strength criteria for a particular failure mode.
- Analyze structural response.
- Evaluate the response for given criteria.

Materials

Selection of materials is very important in structure design, because it directly affects ship cost and strength. Hence, the structure designer should know well the characteristics of materials, especially newly developed ones. Reasonable material selection should ensure the strength of the ship structure and reduce the weight of the ship as much as possible. For different materials, choose the appropriate welding method to ensure the service life. For special parts of ships and special

ships, materials with suitable properties should be selected.

Key Applications

Application History

With the growth in demand for ships and an increase in their complexity, ship structural design and calculation procedures have advanced considerably. Earlier, ships were designed and dimensioned solely on the basis of prescriptive rules from classification societies, which were themselves largely based on experience and feedback from ships in service; in the final quarter of the last century, rational analysis and design methods were introduced (Hughes et al. 2010).

The former Soviet Union first used the strength calculation design method in shipbuilding. In the 1950s, it issued the strength criteria including strength calculation conditions (external force, stress) and allowable stress standards, but the method still belongs to the deterministic design method of "analysis and verification."

Since the 1960s, the shipbuilding industry has undergone profound changes. The ship is required to be safer and more reliable, and at the same time, it must have higher efficiency and economy. Therefore, the old normative design method cannot adapt to the new situation, and it is urgent to establish a more scientific and versatile structural design method. In addition, due to the emergence of high-speed electronic computers, the finite element method of structural analysis and the optimization technology of mathematical programming have developed rapidly, which makes the structure design workers not only have a powerful fast calculation and analysis tool but also have a systematic method to improve juice design and optimize design. Just like the changes in the structural design fields such as aviation and civil architecture, the basic principles and methods of ship structural design are gradually undergoing profound changes.

Firstly, the structural design gradually transitions from a deterministic design principle to a probabilistic design principle. The so-called probability design method is reliability design. It is

based on the probability model of load and strength, which is to use various parameters that affect structural safety and performance during the life of the structure as random variables, using probability theory and combining statistical methods to analyze the probability that the structure meets the basic functional requirements during the service period. Many countries and international organizations have adopted new structural design standards and norms in some areas using structural reliability theory. In the field of ship engineering, the probability-based ship structure design is still in its infancy due to the complexity of the problems faced, but this far-reaching change is gradually deepening. According to the development stage and the degree of precision, the current international probabilistic design method is divided into three levels:

- **Full Probability Design Method:** This method is a design method based entirely on probability theory. It requires an accurate probability description of the joint occurrence of various basic variables that contribute to the structural response, as well as the accuracy of the failure range, in order to obtain a “precise” failure probability of the structural or structural component as a direct measure of reliability. This method is quite difficult in both theory and application, so it is still in the research stage.
- **Approximate Probability Design Method:** This method involves some approximate iterative computational processes, using probabilistic and mathematical statistics to approximate the failure probability of a structure or component. It is generally desirable to make the failure range ideal and usually to simplify the expression of the joint probability distribution of the variable. Today, this method has entered a practical stage and is gradually becoming the basis for the development of structural norms in many countries.
- **Semi-probability Design Method.** This method uses mathematical statistics analysis for certain quantities that affect structural safety and introduce some empirical coefficients in combination with experience, so it is also

called semi-probability semiempirical design method. This method does not estimate the reliability of the structure quantitatively. The appropriateness of structural reliability is provided on the basis of structural components (and sometimes on the basis of structures) by applying a number of subitem safety factors which are related to predetermined eigenvalues or standard values of the major variables of the structure and load.

In addition, with the wide application of electronic computers in structural analysis, the optimal design method characterized by “comprehensive and optimized structure” has rapidly developed. The structural optimization design not only reduces material consumption and cost under conditions of strength and stiffness but also enhances the function of the structure when needed. However, it is essentially a tool only, and it is impossible to solve the problems that are not solved by the structural design principle.

Finally, it is about the load state of the design and the form of damage to be prevented. The limit state calculation method is a major development in structural design. The former Soviet Union applied this method to some structural design specifications in the 1950s. Japan also applied this method in some ship construction specifications in the 1960s and 1970s. Today, the calculation of structural reliability and safety is inseparable from the limit state analysis.

Computer-Aided Design

Computer-aided design (CAD) is the process of creation and manipulation of design models on a computer to assist the engineer in the design and analysis process. In the modern environment, CAD can be used to design, simulate, develop, and optimize engineering structures.

Today, in ship structure design and analysis, it is possible to use the CAD model not only for visualization purposes but also for other analytical purposes such as basic structural form design, configurational design, design of final dimensions, simulation of motion characteristics, design of required plate sizes and thickness for

production, design of heating pattern for welding and joining, etc.

The complex beam structure is a 2D/3D structure subjected to simple or combined loading, e.g., superstructure comprising of side frames, deck frames, longitudinal girders, and compression posts that is subjected to deck pressure and sea pressure simultaneously, and this is analyzed for stress, deflection, and vibration. In ship structural analysis, normally three methods of analysis are employed, and they are (Sharma et al. 2012):

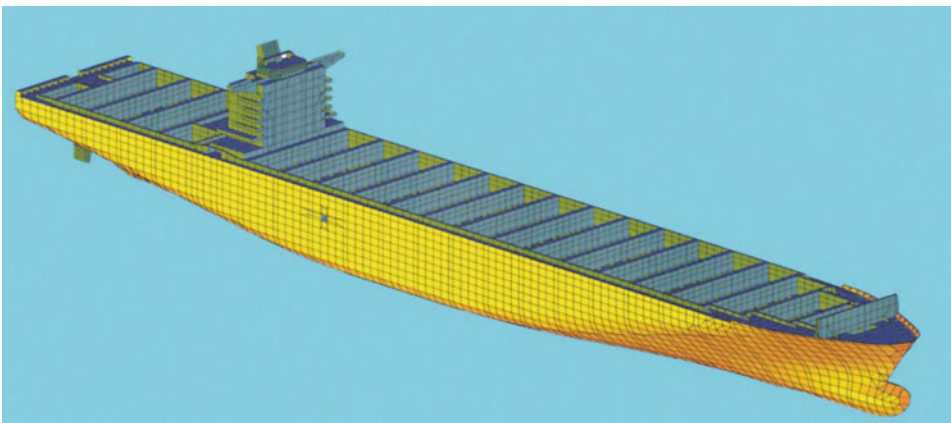
- Spreadsheet computation: The spreadsheet is used for the first set of local calculations to the class society rules. They can also be used for simple one-dimensional beam calculations as the formulas are quite straightforward and easy to write in a spreadsheet format.
- Beam modeling: A 1D beam such as a fixed-ended beam with a uniformly distributed load can be analyzed with a simple spreadsheet. However, 2D/3D models or 1D model with a complex load condition is analyzed with beam modeling software, e.g., deck beam of the mid-ship section.
- Finite element analysis: The beams in a ship are normally welded directly to the shell plate. This means that the shell plate is included in the beam properties. In this formulation, the results give an indication of the stress in the plate; however, due to the mathematical formulation of a beam element, the analysis does

not consider the connection of adjacent plates. Therefore to compute shell plate stresses and deflections, other methods are needed, e.g., finite element analysis (FEA).

Finite Element Method

FEM is a numerical solution method to discretize the continuum based on variation principle. It can disperse the hull structure into a finite number of units that can accurately simulate its load-bearing mode and deformation. The details of the hull structure can be described in detail. To express the coordination relationship and changes among various components, it is possible to determine the actual deformation and stress of each component or area. This method is currently the most accurate and perfect method for hull strength analysis, and it is also the structural analysis method that can accurately forecast the response of structures to loads in rational structural design. Figure 4 is an example of a finite element model of a medium-sized containership.

Our general-purpose finite element software includes ABAQUS, ANSYS, ADINA, SAP, MARC, NASTRAN, etc. The mainstream finite element software in the shipbuilding industry is NASTRAN. It has an open, fully modular organization structure that not only has strong analysis capabilities but also ensures good flexibility. Users can choose and combine the system requirements to get the best application system according to their own engineering problems. In addition to



Ship Structural Design, Fig. 4 Finite element model of a 9200 TEU containership (Hughes et al. 2010)

the universal finite element software, the world's major shipbuilding countries have developed ship-specific finite element software, such as the SHIPRIGHT system of the Lloyd's Register of Shipping, the SAFEHULL system of the American Classification Society, the NAUTICUS system of the Norwegian Classification Society and France, Classification Society's VERISTAR system, etc.

In general, depending on the requirements, the finite element analysis of the hull structure can be divided into the following three different levels:

- Analysis of the entire ship: The purpose is to obtain the overall situation of stress and deformation of the hull, which is mainly used for strength analysis of special ships such as large-opening vessels (such as container ships).
- Analysis of compartments: Evaluate longitudinal hull girder components, primary support members, and transverse bulkhead strength. The classification of the models is not exactly the same for each classification society based on their respective considerations.
- Subdivision mesh analysis: Evaluate the detailed stress levels of local structural details.

Example of New Ship Structure Design

In order to meet the needs of the market in terms of ship environmental protection, transportation economy, and maximum safety, Stena Bulk Shipping Company and other companies have jointly developed a product tanker E-MAXair with low emission, low energy consumption, and complete redundancy. The ship's design features include optimized hull structure (patented), the use of two large low-speed rotating propellers, and the use of LNG and special sails using wind to reduce fuel consumption. The ship uses airMAX technology to design a cavity that creates an "air cushion" on the bottom of the ship while using the newly developed ball nose to reduce hull resistance. Compared with ordinary oil tankers of the same scale, E-MAXair tanker significantly reduces energy consumption and emissions and improves transportation economy and ship safety. Optimization of ship structural design plays a key role in the development of new ship types.

In April 2010, the Norwegian Classification Society (DNV) launched the future environmentally friendly container concept ship "Quantum" (shown in Fig. 5). The main purpose is to reduce the negative impact on the environment by reducing fuel consumption while transporting more containers. When designing the "Quantum"



Ship Structural Design, Fig. 5 Future environmentally friendly container concept ship "Quantum" (<http://www.baidu.com/>)

Ship Structural Design,**Fig. 6** “River-to-SeaNo. 1” (<http://www.baidu.com/>)

container ship, DNV adopted a variety of techniques to improve the efficiency and environmental friendliness of the ship. In the hull design, the ship width is increased to improve stability and reduce the demand for ballast water. The ship uses an innovative wide deck design to increase the packing capacity and a narrow outboard side tank design to improve break stability and reduce steel weight. The ship obtains a smaller square factor (only 0.57) by optimizing the ship shape and the bulb nose and adopts an aerodynamic design of the wave breaker to reduce the wind resistance. For the main engine and propulsion, “Quantum” is equipped with four dual-fuel (marine diesel and LNG) generator sets to drive two pod-type electric propulsion units. Various improvements in the ship structure design have greatly improved the performance of this concept ship.

On April 10, 2018, “River-to-Sea No. 1” which is the first river-to-sea ship full of 20,000 t of iron ore sand in China (shown in Fig. 6) completed its first flight, marking the opening of a new era of China’s shipping industry. River-to-sea ship has the characteristics of both inland riverboats and seagoing vessels. Compared with traditional ships, river-to-sea ships have been innovative in terms of size, proportion, and configuration.

By improving the seagoing ship such as shortening the length and increasing the width of the seagoing ship, the river-to-sea can be obtained. Similarly, it is through the change of ship structure design that the river-to-sea ship structure can meet the requirements of sea and river transportation without intermediate transportation.

Cross-References

- Detailed Design
- Empirical Design
- Multidisciplinary Design Optimization (MDO)
- Optimal Design
- Preliminary Design
- Reliability Based Design (RBD)
- Ship Design Process
- Ship Overall Design
- Structural Design
- Welding Technology

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Shipboard Electrical Design

► Ship Electrical Design

Shipboard Electrical System Design

► Ship Electrical Design

Ship-Fitting Design

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Synonyms

Fit out a ship; Fitting-out; Outfit; Outfitting

Definition

Ship-fitting design is defined as the installation work of all devices, facilities, and equipments besides the ship hull structure, including hull fitting, machinery fitting, electric fitting, power plant, control device, pipeline, etc. In the process of modern shipbuilding, the workload of fitting is 50–60% of total workload of shipbuilding. For the complicated ship, this percent will be higher. So the fitting is the indispensable component of shipbuilding (Fig. 1) (Liang 2011).

Scientific Fundamentals

Fitting Equipment

See Fig. 2.

The facilities used for ship fitting could be classified as (Wang 2013) anchoring equipment, mooring equipment, rudder equipment, evading equipment, hoisting equipment, lashing equipment, lifesaving equipment, fire accessories, close device, ladder pole equipment, accommodations, kitchen equipment, marine ventilator, flexible unit, etc. Each fitting facility also contains many equipment and accessory, as shown in Table 1.

Fitting Classification

Ship fitting can be classified as construction process as unit fitting, section fitting, block fitting, berth fitting, and wharf fitting. Also ship fitting can be classified as content as hull fitting, engine room fitting, and electric fitting. And hull fitting can be classified as inner fitting and outer fitting. Inner fitting is also called accommodation fitting, while outer fitting is also called deck fitting.

Classification According to the Construction Process

Fitting unit is defined as the fitting assembly containing a group of fitting equipment. Unit fitting is the work to assemble the fitting equipment in the unit into the fitting assembly of the fitting platform in the plant. The unit fitting is carried out at three stages. The stages are unit

Ship-Fitting Design,**Fig. 1** Fitting locale
(FreeImages.com/szymon
szymon)

installation on section, unit installation on block, and unit installation on inboard. For example, some deck machines in outer fitting, such as anchor gear, gangway ladder, radar mast, etc., are combined into a whole by unit fitting and then hoisting to deck for positioning installation. The fitting equipment can be divided into fitting unit including equipment unit, bin unit, valves unit, panel unit, etc. (Wang 2011).

Section or block fitting is carried out at a proper stage of section or block construction and installs the fitting equipment or fitting unit on the corresponding section or block. Section or block fitting regards the section or block as fitting area and is carried out at platform in infield or outfield as the order of intermediate product assembly. Compared to the berth fitting and wharf fitting, section fitting or block fitting improves the working environment and working posture. The high-altitude operation is changed to ground operation, and the production efficiency, installation quality, and work safety are enhanced; meanwhile the number of scaffold and fitting period is reduced (Wang 2011).

Unit fitting, section fitting, and block fitting are defined as advanced fitting, as shown in Fig. 3, and are the technology concept appeared since the fitting and hull construction should be alternated

or parallel according to the IHOPS and modern shipbuilding mode (Li and Liang 2007).

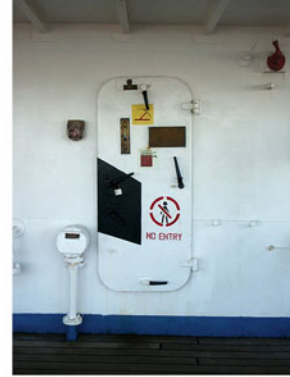
The berth fitting during berth final assembly and wharf fitting after launching all belong to inboard fitting. Ideal inboard fitting is limited to (1) the fitting equipment or fitting unit that is too large or too heavy to be installed on section or block; (2) the fitting equipment that is easy to be damaged before the compartment cover; (3) the fitting connecting piece between units and sections (Diao 2006). The fitting area of inboard fitting should be divided by the central position of fitting equipment, such as engine room area, rudder area, deck area, superstructure area, etc. Since the fitting workload is heavy and fitting period is long in engine room area, the inboard fitting should be started when the engine room open area is formed during ship assembly. By this way, the ventilation and lighting condition are well during the inboard fitting; meanwhile the installation into cabin for large equipment and fitting unit is more beneficial (Diao 2006; Wang 2011).

Classification According to Fitting Content

Engine room fitting is defined as the installation and trial run for all kinds of equipment in engine room. And the fitting work mainly contains the



Anchoring equipment
(FreeImages.com/Phillip Collier)



Close device
(FreeImages.com/rui lousao)



Hoisting equipment
(FreeImages.com/Vangelis Thomaidis)



Life-saving equipment
(FreeImages.com/Andy Muir)

Ship-Fitting Design, Fig. 2 Classical fitting equipment

installation and trial run for engine room equipment and erecting welding for fitting equipment. The former contains the installation for large mechanical equipment such as main engine, shafting device, boiler, dynamo, etc. And the latter contains the installation for piping system, substrate, and bin in engine room. The characteristics of fitting in engine room are intensive equipment, high accuracy requirement, poor work condition, and inefficiency (Fig. 4) (Xu 2015).

Electrical fitting is defined as the installation, wiring, examine and trial run for electrical equipment, and cable laying. Compared to the electrical equipment with the same function on land, the working environment of electrical equipment on ship is worse. And the shockproof performance, moisture resistance, and anti-corrosion property

must be considered. So the structure of electrical equipment is more complex, and installation technology is different. Especially the marine cable, since its laying must be carried out in narrow space, its structure and laying are special. Now, the degree of marine electrification and automation is increased day by day, and the workload of electrical installation is multiplied. To speed up the process of electrical fitting, the technologies like preassemble and electrical lofting are generalized. The electrical fitting is divided into infield preparation and outfield preparation. The condition of infield preparation is well, and the hull work will not affect the infield preparation. So the infield condition can be started ahead of time. The condition of outfield preparation is bad, since the work is disperse and high above

Ship-Fitting Design, Table 1 Fitting equipment classification (Wang 2013)

Fitting facility categories	Fitting equipment and accessory
Anchoring equipment	Anchor, anchor cable, hawse pipe, anchor recess, anchor rack, anchor stopper, abandon the anchor device, chain cable fairlead, anchor davit
Mooring equipment	Mooring rope, mooring pipe, fairlead, mooring bollard, cable reel
Rudder equipment	Rudder blade, rudderstock, tiller, rudder carrier, neck bearing, steering wheel
Evading equipment	Spring towing hook, spring free towing hook, clevis joint, towing post, pneumatic spring
Hoisting equipment	Derrick boom, derrick bearing, lifting hook, chain and swivel, lifting tackle, eye plates and rings, shackle, funiculus, jack rope winch
Lashing equipment	Container fasteners, container binding, tank fixing device, automotive aircraft fixtures, quick screw
Lifesaving equipment	Lifeboat, rescue boat, life jacket, cork hoop, inflatable life raft, davit gear, lifting frame
Fire accessories	Fire bucket, water cage with box, sand and sand box, withdrawal tool
Close device	Door, windows, hatch cover, manhole cover
Ladder pole equipment	Ladder, handrail, storm rail, backdrop
Accommodations	Room furniture, sanitary ware, ironware
Kitchen equipment	Steamer, oven, dishwashing machine, boiling water device, kitchen multipurpose machine, dirt crusher
Marine ventilator	Ventilation head, ventilation opening, ventilated brake
Flexible unit	Turnbuckle, shackle, eye plates and rings

the ground faceup. The section fitting can improve the work condition and balance operating load. So the electrical fitting should carry through the principle “more infield work and more section work” (Li 2010).

The hull fitting can be divided into inner fitting and outer fitting. The inner fitting is also called accommodation fitting containing the laying of

insulation and dressing, non-steel wall, ceiling, door, windows, furniture, toilet, kitchen equipment, refrigeration house equipment, air condition equipment, etc. The outer fitting is also called deck fitting containing rudder and steering equipment, ground tackle, mooring equipment, towing equipment, life-saving equipment, cargo lift equipment, hatch cover, roll-on/roll-off facilities, firefighting accessory, natural ventilation accessory, pipeline, etc. The outer fitting is throughout the whole ship and covers the area except the engine room area and accommodation area. And the fitting equipment of the outer fitting to be assembled is numerous (Figs. 5 and 6) (Diao 2006).

Features of Fitting (Ying 2013)

In the whole productive process of fitting, it has the features including many aerial works, interchange works, over waterworks, and special works. It covers almost all kinds of shipbuilding. The main features are shown as below.

Many Aerial Works

With the development of shipbuilding industry, the size of ship hull is increased. At the inboard fitting stage, the whole ship has folded and undocked. So a large number of workplaces are far from the ground, so aerial works are everywhere in fitting (Fig. 7).

Many Interchange Works

At the fitting stage, the cross-operation of each department and occupation is taken place at the high, middle, and low place of the ship's hull and cabin. This situation will bring about the difficulty of interconnection and coordination for each occupation. And the accident happens at a high risk.

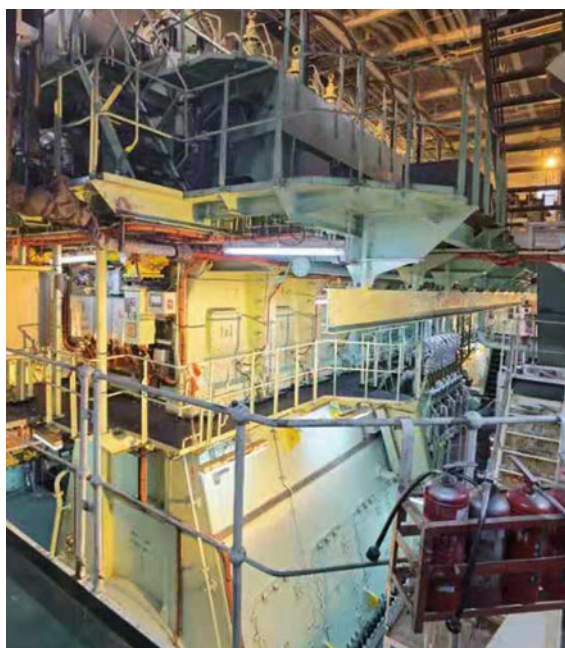
Many Restricted Space Works

For all kinds of ship, although they vary in size and their forms are different, they all have a set number of hermetic and narrow cabin. In the restricted space, the work is inconvenient, and high concentration of inflammable, explosive, and toxic substances may occur because of the poor ventilation condition. So explosions and poisoning accidents may happen at a high risk (Fig. 8).



Ship-Fitting Design, Fig. 3 Advanced fitting (Fang 2014)

**Ship-Fitting Design,
Fig. 4** Energy room fitting



S

Many Fire Operations

The fire operation including electrowelding and gas cutting is the main occupation of shipbuilding. According to the statistics, welding work always

accounts for more than half of the total time of ship hull construction. At the fitting stage, many fire operations often conduct simultaneously on the same ship (Fig. 9).

Ship-Fitting Design,

Fig. 5 Outer fitting (deck fitting)(FreeImages.com/
Tomas B.)

**Ship-Fitting Design,**

Fig. 6 Inner fitting (accommodation fitting)
(FreeImages.com/
NotImportant
NotImportant)

**Ship-Fitting Design,**

Fig. 7 Aerial works ([http://
info.cm.hc360.com/2017/
03/030906662482.shtml](http://info.cm.hc360.com/2017/03/030906662482.shtml))



Ship-Fitting Design,**Fig. 8** Most energy rooms have restricted space works**Ship-Fitting Design,****Fig. 9** Fire operations
(FreeImages.com/Daniel Vazquez)**Many Dangerous Goods**

In the fitting, there are many dangerous goods, such as oxygen, propylene, and acetylene used for gas welding and cutting, marine paints, etc. These goods are all inflammable and explosive substances.

Many Occupations

Fitting needs numerous operating personnel, including productive work and service work. The productive work contains electrician, bench worker, burner, welder, assembler, plumber,

cabinetworker, painter, etc. The service work contains ventilate and illumination worker, lifting worker, scaffold worker, etc.

Key Applications

Since the development of manufacturing technology and design technology, the development stage of shipbuilding technology can be divided into (Wang 2013):

First stage, shipbuilding schema that is system oriented. In this stage, the main technologies are in bulk at slipway, wharf fitting, and coating the whole ship.

Second stage, shipbuilding schema that is system and region oriented. In this stage, the main technologies are section building, advanced fitting, and advanced coating.

Third stage, shipbuilding schema that is region, type, and stage oriented. In this stage, the main technologies are hull process lane construction, regional fitting, and regional coating.

Fourth stage, intermediate production-oriented shipbuilding schema. In this stage, the main fitting technology follows the hull construction fitting and painting integration schema (Okayama et al. 2006).

Fifth stage, production-oriented shipbuilding schema. In this stage, the fitting technology follows the integration schema of design and manufacturing.

The first stage and second stage belong to the traditional shipbuilding technology. The third stage and fourth stage belong to modern shipbuilding technology. The fifth stage is future shipbuilding technology (Wang 2013). As shown from the evolution of shipbuilding technology, the development process of fitting can be divided into the stages as follows: slipway and wharf fitting stage, advanced fitting stage, region fitting stage, hull construction fitting and painting integration stage, and hull and fitting separation stage. The hull and fitting separation technology is the development direction of modern shipbuilding schema and is the new construction technology improved from the fitting technology according to the modular technology (Calvo 1987; Baade et al. 1998; Jolliff 2010). The separate construction of hull and fitting can realize the fitting unitization and function modularization (Liu 2012). Besides the modularization, the fitting technology also shows tendency of the premiumising, environmental protection, and energy-saving (Wang 2013).

Modularization (Zhou 2008; Liu 2012)

The main advantages of the fitting modular design are forward production hour, less shipbuilding

cost, and improved work method and work environment (Zhou 2008).

The core of the modern shipbuilding is to realize the production rhythm with balance and continuity. With the shifting forward of production hour increasing, the period of dock and wharf will be reduced, and the use of radio as core resource is enhanced. As the analysis of statistics, if the same works are arranged at different construction stages, the engaged production hours of C stage, B stage, P stage, and D stage are 1, 3, 6, and 9, respectively (where C, B, P, and D mean the craft stage of little assemblage, large assemblage, slipway, and wharf, respectively). Thus it can be seen, since the bad work environment of the later stages with more aerial work and more preparation time of auxiliary facilities, the engaged production hour is increased obviously.

The engaged production hour of advanced shipyard is balanced. The engaged production hour of later stages is reduced; thus the total production hour is also decreased obviously. The fitting modularization can advance drastically the work at each stage mentioned above, and the construction hour is reduced obviously.

The fitting modularization design and manufacture can reduce the time and complexity of running and assurance in total life cycle including the design, construction, procurement, and so on, through the modularization, standardization, and process simplification of electromechanical devices. Thus the total cost in shipbuilding period and service period is also reduced. Meanwhile, the modular design and manufacture can advance the work at slipway to infield or outfield. It also reduces the cost, since the expense of unit or module installing in infield or outfield is five to seven times lower than that at slipway or dock.

The traditional shipbuilding industry is called 3K industry, i.e., dirty, laborious, and dangerous, and is labor-intensive industries. With the improvement of working method, the work environment can be improved as well. And the labor-intensive industries can be turned into equipment-intensive industries and knowledge-intensive industries. Through the modular shipbuilding including modular fitting, the spare parts have to

be installed in section, block, and dock before but now can be installed at open flat ground. It will improve the work environment obviously.

The shipbuilding modular can be divided into self-maintaining modular and attachment modular (Zhou 2008).

The self-maintaining modular is the advanced fitting unit on the section or block. It has independent substrate or support structure and can be prefabricated at infield and installed on the ship subsequently, such as CO₂ fire extinguishing device, remote control valve set for infusion system, seawater and freshwater pressure tank, ordinary crew accommodation, etc.

Attachment modular indicates that the design modular usually, since the wide arrangement range of its equipment and high requirement of fixation, is dependent on the structure of section or ship in the actual construction. The prefabricate of the attachment modular will be limited by the schedule of hull construction, but the optionality of equipment and universality of unit are the same with self-maintaining modular, so the attachment modular is also called as drawing shipbuilding modular.

In addition to the two modules mentioned above, there are many distributional fitting equipments on the ship. The distributional fitting equipment module is the particular case of attachment module, and it mainly contains the pipeline, cable, and its accessories.

The features of these modules are having independent function, impeccable port, well interchangeability, applicability, advancement, and commodity.

In the future, the fitting module will be developed in that direction of functionality, regionality, and integrity (Liu 2012).

Functionality module treats the equipment with strong independence as principal part, and has independent function, connecting with pipeline, accessory, electric appliance, iron fitting equipment, etc., to be a entirety. This type of module has well technology performance, convenient operation, and reliable quality, such as module of fuel oil, slide oil, and oil purifier, fuel feed module, cold and hot water pressure cylinder module, various kinds of individual oil tank, etc.

Regional module is the overall module with large scale and completeness and is an advanced fitting form with high production efficiency, quick returns, and good benefits, for example, the overall fitting module in dynamo region, in inner bottom of engine room region, in chimney region, etc.

Integrity module contains the radar mast, light mast and inert gas discharge column module, material shelf module, transport module, ventilating device module, system device module, etc.

Premiumising

Premiumising is mainly embodied at the appearance of fitting equipment, such as esthetic measure, symmetry, paint coat, design and manufacturing accuracy, etc., and the performance of fitting equipment, such as running smoothness, interface fit degree, interchangeability, etc. Although premiumising is the target of fitting development, excessive and one-sided pursuing of premiumising will increase the production costs too much. So the enterprise shall consider the actual situation and comprehensively consider the product competitiveness and economic benefits objectively (Wang 2013).

Environmental Protection and Energy Saving

Because of frequent human activities, the pollution of the sea is increasing day by day. So the environmental protection and energy-saving design for fitting equipment must be considered. At first, the material of fitting equipment must be nonhazardous and environmental, to reduce the pollution. Secondly, the energy of ship should be saved as much as possible. On the one hand, more manual operations can be added and, on the other hand, minimize the weight of the equipment to save the material and energy at the same time (Wang 2013).

Cross-References

- [Forming Technology of Profiles](#)
- [Mooring System](#)
- [Welding Technology](#)
- [Winch](#)

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Ship-Iceberg Interactions

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Synonyms

Confinement; High-pressure zone; Iceberg;
Pressure-area relationship; Ship

Definition

Iceberg is formed from land-based glaciers by falling off from the land. It is made from

freshwater rather than sea water. The ship-iceberg collision is a potential hazard for those ships navigating in ice-covered areas. However, due to the increased technology of detection system on board, the probability of large iceberg colliding with ship might be marginal. **Ship-iceberg interactions** shall include the global and local effects of both ship and iceberg. Due to the existence of free surface, the complicated ice mechanic properties, this phenomenon is a complicated topic on ocean engineering.

Scientific Fundamentals

Iceberg Types

The iceberg can be categorized according to its main dimensions, the height over sea level (H) and the overall length at sea level (L), see Fig. 1. The draft of iceberg (D) is normally not known (Table 1). There are studies performing the statistic relationship between H , L , and D , see Fig. 1.

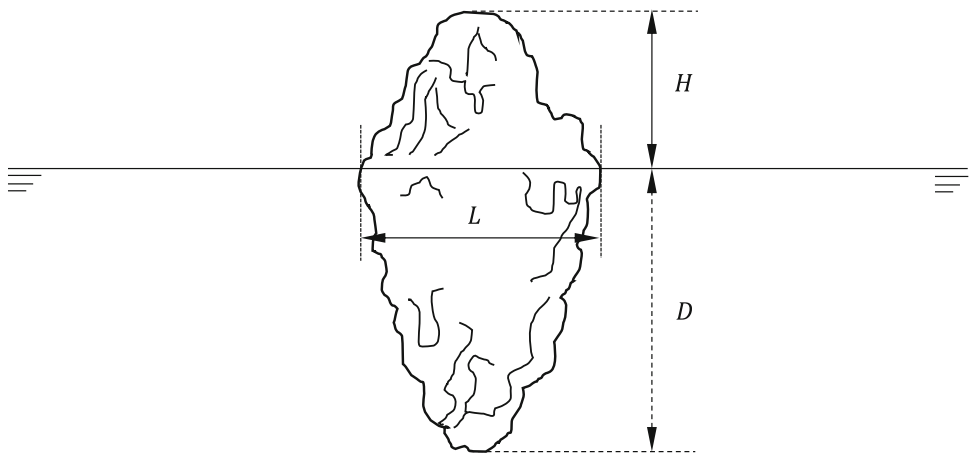
According to the shapes of iceberg above sea level, following six types are defined:

- (a) Tabular iceberg: a flat-topped iceberg, width is greater than five times height
- (b) Blocky iceberg: a flat-topped iceberg with steep sides
- (c) Domed iceberg: an iceberg with smooth and round top
- (d) Pinnacled iceberg: an iceberg has a central spires
- (e) Wedged iceberg: an iceberg with wedge shaped top
- (f) Drydocked iceberg: an iceberg which is eroded to a U-shaped slot

Type (a) and (b) are the main types for those new icebergs. The rest of the types are for those icebergs drifting in the ocean for a certain time (Fig. 2).

Global Ship-Iceberg Interaction

The study of ship and iceberg interaction is a complicated subject. One way to simplify the study is the split of external and internal mechanics as illustrated in Fig. 3. The external mechanics is about the global impact study. The hydrodynamic



Ship-Iceberg Interactions, Fig. 1 Main dimensions of iceberg

Ship-Iceberg Interactions, Table 1 Iceberg classification by size, based on Canada (2016)

Name	Height above sea level H (m)	Length L (m)	Weight W (Mt)
Growler	$H < 1$	$L < 5$	$W \approx 1.0 \text{ e}^{-3}$
Bergy bit	$1 \leq H < 5$	$5 \leq L < 15$	$W > 0.01$
Small berg	$5 \leq H \leq 15$	$15 \leq L \leq 60$	$W > 0.1$
Medium berg	$16 \leq H \leq 45$	$61 \leq L \leq 120$	$W \approx 2.0$
Large berg	$46 \leq H \leq 75$	$121 \leq L \leq 200$	$W \approx 10.0$
Very large berg	$H > 75$	$L > 200$	$W > 10.0$

effects, the eccentricity of both objects shall be considered in the assessment. The internal mechanics is the detailed study of the interaction between ship and iceberg at the impact area.

Liu and Amdahl (2010) further apply Stronge’s impact theory to study the ship and iceberg interaction. The governing equation is listed as follows:

$$dv_i = m_{ij}^{-1} dp_j \tag{1}$$

where dv_i is the relative velocity change on i direction, m_{ij}^{-1} is the inverse of inertia matrix, see Stronge (2004), dp_j is the impact impulse on i direction. The inverse inertia matrix is symmetric, i.e., $m_{ij}^{-1} = m_{ji}^{-1}$. The following are representative elements:

$$m_{11}^{-1} = (M^{-1} + r_2^2 I_{33}^{-1} - 2r_2 r_3 I_{23}^{-1} + r_3^2 I_{22}^{-1}) + (M'^{-1} + r_2'^2 I_{33}'^{-1} - 2r_2' r_3' I_{23}'^{-1} + r_3'^2 I_{22}'^{-1}) \tag{2}$$

$$m_{12}^{-1} = (M^{-1} + r_1 r_3 I_{23}^{-1} - r_3^2 I_{21}^{-1} - r_1 r_2 I_{33}^{-1} + r_2 r_3 I_{31}^{-1}) + (M'^{-1} + r_1' r_3' I_{23}'^{-1} - r_3'^2 I_{21}'^{-1} - r_1' r_2' I_{33}'^{-1} + r_2' r_3' I_{31}'^{-1}) \tag{3}$$

$$m_{13}^{-1} = (M^{-1} + r_1 r_2 I_{32}^{-1} - r_2^2 I_{31}^{-1} - r_1 r_3 I_{22}^{-1} + r_2 r_3 I_{21}^{-1}) + (M'^{-1} + r_1' r_2' I_{32}'^{-1} - r_2'^2 I_{31}'^{-1} - r_1' r_3' I_{22}'^{-1} + r_2' r_3' I_{21}'^{-1}) \tag{4}$$

where M^{-1} , M'^{-1} are the inverse of mass matrix for ship and iceberg, respectively. I_{ij}^{-1} , $I_{ij}'^{-1}$ are the inverse of inertia matrix for ship and iceberg, respectively. r_i , r_i' are the vectors from the center of gravity to the collision point for ship and iceberg, respectively. In these expressions, it should be noted that the matrix I_{ij} of moments and products of inertia has an inverse, which is denoted by I_{ij}^{-1} , e.g., $I_{21}^{-1} = (\hat{I}_{13}\hat{I}_{23} - \hat{I}_{12}\hat{I}_{33})/\det(\hat{I}_{ij})$. Recently, Liu and Amdahl (2019) presented the detailed



a), tabular iceberg



b), blocky iceberg



c), domed iceberg



d), pinnacled iceberg



e), wedged iceberg



f), drydocked iceberg

Ship-Iceberg Interactions, Fig. 2 Iceberg types based on the shape over sea level, photos provided by David Stanley, Lavivm, Andreas Weith, Magnus Manske, Natalie Lucier, and Kim Hansen, respectively

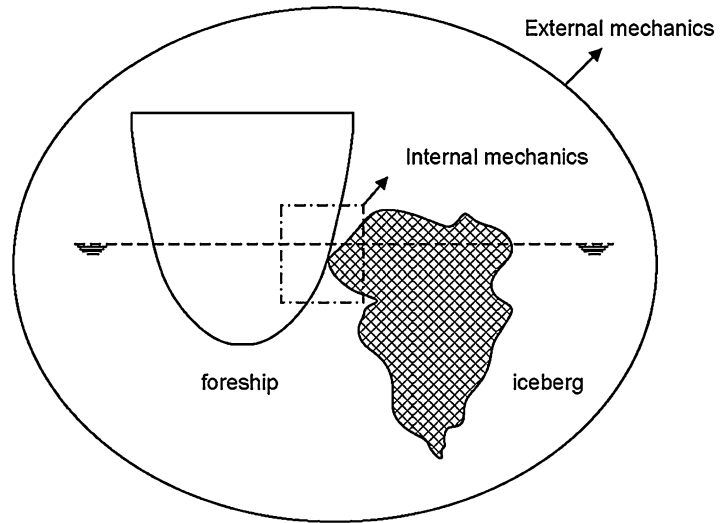
procedures on deriving the equation of motions for ship collisions. The formulation was presented under the ship's body coordinate system, rather than the local coordinate system at the impact point. By doing this, the items in the inertia matrix could be approached with the so-called reduction factors, which was first proposed by Popov et al. (1969) by considering the normal impact force only. It unifies existing ship impact theories, see Liu and Amdahl (2019) for more details.

As shown in Fig. 4, a local coordinate system is established at the collision point and the equations of motion are transformed to the local coordinate system. A transformation matrix is defined based on the ship's hull shapes, see Eq. (5).

$$T_{lg} = \begin{bmatrix} \cos(\alpha) & -\sin(\alpha) & 0 \\ -\sin(\alpha)\sin(\beta') & -\cos(\alpha)\sin(\beta') & -\cos(\beta') \\ \sin(\alpha)\cos(\beta') & \cos(\alpha)\cos(\beta') & -\sin(\beta') \end{bmatrix} \quad (5)$$

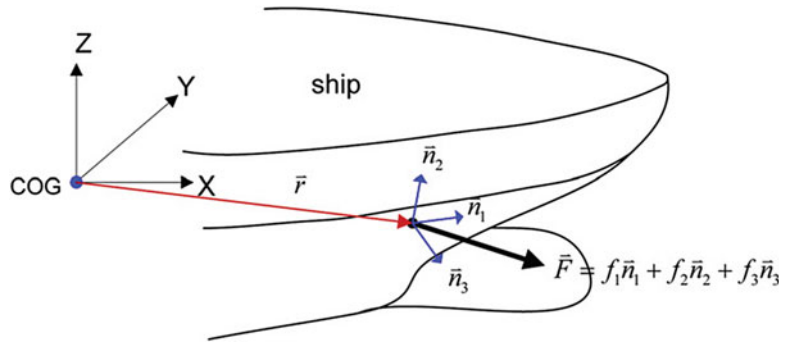
Ship-Iceberg Interactions,

Fig. 3 Illustration of external and internal mechanics (Liu et al. 2011)



Ship-Iceberg Interactions,

Fig. 4 Collision model by Liu and Amdahl (2010), with permission of Elsevier



where α , β' are the waterline angle and normal frame angle of ship, respectively at the collision point. The definition of angles can be found in DNV (2009).

The impulse in each direction, $\vec{n}_i$ ($i = 1, 2, 3$), is given by:

$$dp_i = f_i dt \quad (6)$$

where f_i is the i th component of the interaction force acting on the infinitesimally small deforming element in the $\vec{n}_i$ direction at the collision point.

The relative acceleration $\ddot{s}_i$ in each direction is:

$$\ddot{s}_i = \frac{dv_i}{dt} \quad (7)$$

By substitution of Eqs. (6) and (7) into Eq. (1), we obtain:

$$\ddot{s}_i = m_{ij}^{-1} f_j \quad (8)$$

The acceleration can be written as:

$$\ddot{s}_i = \frac{d\dot{s}_i}{ds_i} \frac{ds_i}{dt} \quad (9)$$

Substituting Eq. (9) into Eq. (8) and integrating over the impact duration t , it can be seen that:

$$\begin{aligned} \int_0^t m_{ij}^{-1} f_j ds_i &= \int_0^t \dot{s}_i d\dot{s}_i \\ &= \frac{1}{2} \left((v_i^t)^2 - (v_i^0)^2 \right) = \frac{1}{2} \Delta v_i^2 \end{aligned} \quad (10)$$

where v_i^t, v_i^0 are the relative velocities before and after the collision in the $\vec{n}_i$ -direction, and Δv_i^2 is the change of the squared relative velocities v_i^t and v_i^0 . Note that it is not always positive.

$$\Delta v_i^2 = (v_i^t)^2 - (v_i^0)^2 = dv_i(dv_i + 2v_i^0) \quad (11)$$

where dv_i is the relative velocity change in the $\vec{n}_i$ -direction:

$$dv_i = v_i^t - v_i^0 \quad (12)$$

Further,

$$\begin{aligned} \int_0^t m_{ij}^{-1} f_j ds_i &= m_{ij}^{-1} \frac{f_j}{f_i} \cdot \int_0^t f_i ds_i \\ &= E_i \cdot m_{ij}^{-1} \frac{f_j}{f_i} \end{aligned} \quad (13)$$

Substitution of Eq. (13) into Eq. (10) yields:

$$\begin{aligned} E_i &= abs \left(\int_0^t f_i ds_i \right) \\ &= \frac{1}{2} \cdot abs \left(\frac{1}{m_{ij}^{-1} \frac{f_j}{f_i}} \cdot \Delta v_i^2 \right) \end{aligned} \quad (14)$$

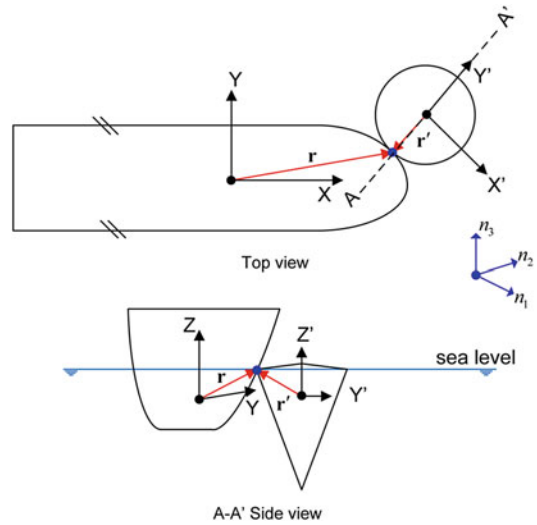
where “abs” indicates to calculate the absolute value because the force f_i and ds_i may be not at the same direction. If we define an equivalent mass variable $\bar{m}_i$ as follows:

$$\frac{1}{\bar{m}_i} = m_{ij}^{-1} \frac{f_j}{f_i} \quad (15)$$

Then, the dissipated energy has the following form:

$$E_i = \frac{1}{2} abs(\bar{m}_i \Delta v_i^2) \quad (16)$$

Equation (16) shows that the complicate *IDOF* can be simplified to a *IDOF* problem in each direction if the equivalent mass $\bar{m}_i$ and change of the squared relative velocity Δv_i^2 can be derived. These can only be done by introducing proper



Ship-Iceberg Interactions, Fig. 5 An example of ship and iceberg impact by Liu and Amdahl (2010), with permission of Elsevier

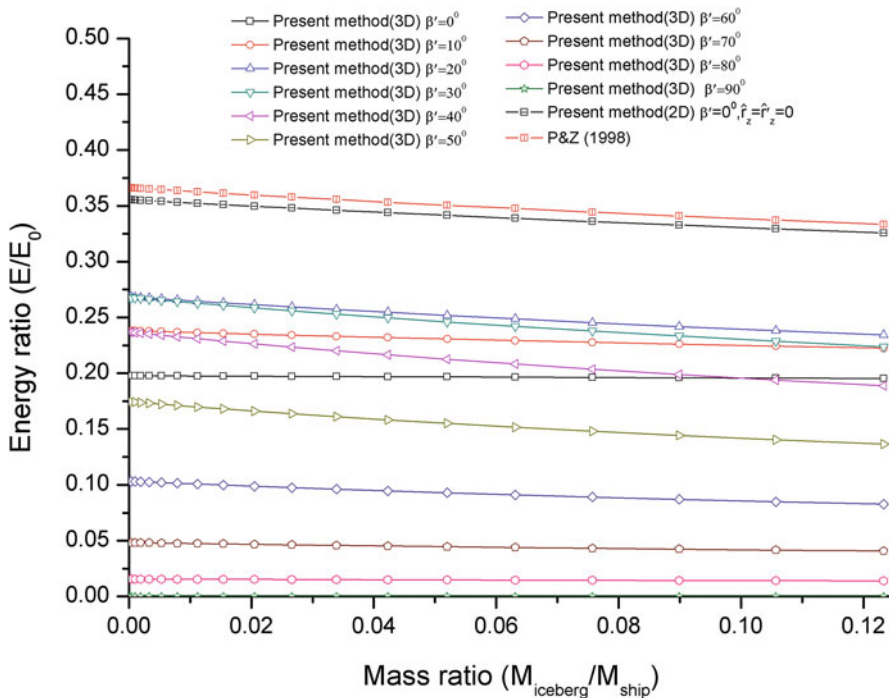
boundary conditions, such as the friction definition and relative velocities.

The final output of this method is the impact energy between ship and iceberg. An example case has been shown in Figs. 5 and 6. An iceberg with idealized shape collides with a tanker in its front area. As seen from Fig. 6, the energy dissipation ratio largely depends on the normal frame angle of ship. Comparing to the 2D method approach, the 3D approach is less conservative. The maximum energy dissipation happens for the normal frame angle between 20° and 30° in the studied case.

Local Ship-Iceberg Interaction

Mechanism

Iceberg ice falls within the category of a multi-year ice, which has survived at least two summer seasons. It is composed of freshwater ice from land-based glaciers flowing off the land into the sea. Iceberg ice can be idealized as an isotropic material, as it shows no significantly different mechanical properties in each direction (Sanderson 1988). In this sense, it is comparatively simpler than sea ice. However, iceberg ice is also influenced by temperature, porosity, grain size, strain rate, and confinement. In general, the



Ship-Iceberg Interactions, Fig. 6 Energy ratio and mass ratio relationship (Liu and Amdahl 2010), with permission of Elsevier

ice becomes weaker and softer with increasing temperature, porosity, and grain size. The effect of the strain rate (loading rate) was discussed by Schulson and Duval (2010). Strength increases with increasing strain rate until brittle failure takes over after which strength decreases. There exists a transition from ductile to brittle, with a strain rate of 10^{-3} . In iceberg collisions, it is believed that the strain rate is greater than the transition value, as discussed by Liu et al. (2011).

During a collision with structures, ice within the center contact area undergoes large confinement and compression (see Fig. 7). High-pressure zones (HPZ) exist in these areas, which may be as high as 70 MPa. For ice features next to the edge, spall and extrusion are more likely. A phase change from ice to water is possible in these small areas. Thus, for high hydrostatic pressures, the ability of ice to resist deviatoric stresses is reduced.

Numerical Simulation

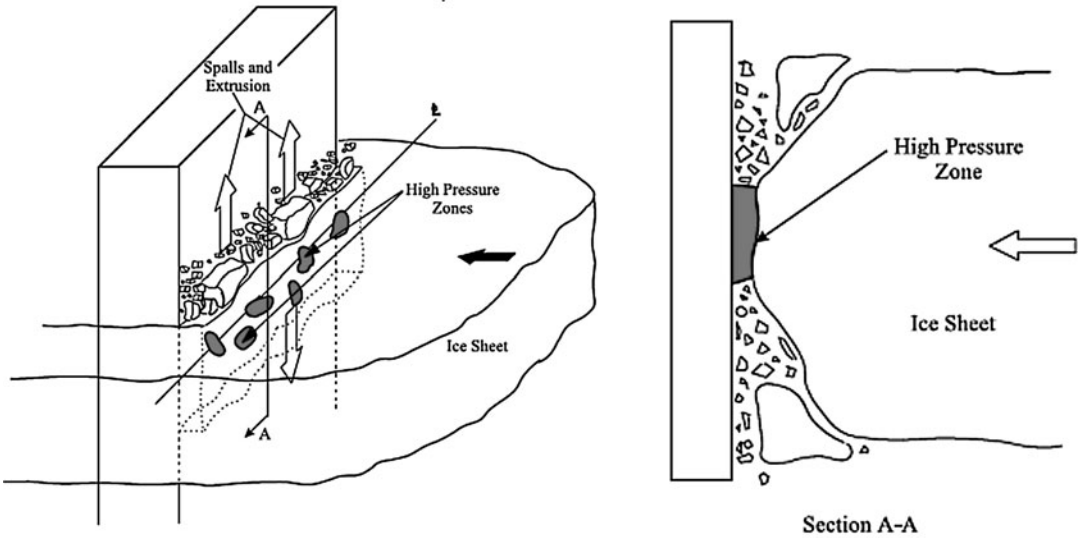
A key technique in the numerical simulation of the ice-structure interactions is the choice of failure

criterion for ice, either plastic yielding or fracture initialization/propagation. Due to the large uncertainties on the ice properties, the failure criterion might be not straightforward to obtain. The material models for steel and soil are not suitable due to the independence of hydrostatic pressure on the material strength. This section presents an approach proposed by the authors, in which a shear-cap yield surface, so-called “Tsai-Wu” (e.g., Tsai and Wu 1971), is used to simulate iceberg impact behaviors.

The Tsai-Wu yield surface for isotropic material is usually written in the following form:

$$f(p, q) = q - \sqrt{a_0 + a_1 p + a_2 p^2} \quad (17)$$

where p is the hydrostatic pressure, $p = \frac{\sigma_{kk}}{3} = \frac{I_1}{3}$ and q is the deviatoric stress, $q = \frac{3}{2} \sqrt{s_{ij} : s_{ij}}$, s_{ij} are the deviatoric stress components, and a_0, a_1, a_2 are constants which should be fitted to triaxial experimental data. In order to make the implementation



Ship-Iceberg Interactions, Fig. 7 Schematic illustration of the ice and structure interaction by Jordaan (2001), with permission of Elsevier

convenient, the Tsai-Wu yield surface is rewritten in $p - J_2$ space as follows:

$$f(p, J_2) = J_2 - (a_0 + a_1 p + a_2 p^2) = 0 \quad (18)$$

where J_2 is the second invariant variable for deviatoric stress tensor, $J_2 = \frac{1}{2} s_{ij} : s_{ij}$. Figure 8 shows a comparison of the recommended inputs based on various data sources in $p - J_2$ space. It is seen that there are big differences existing which are due to the different data sources and the fitting methods used to approach the experimental data sets.

An empirical failure criterion, as follows, was proposed in the present doctoral thesis; see Eqs. (19) and (20).

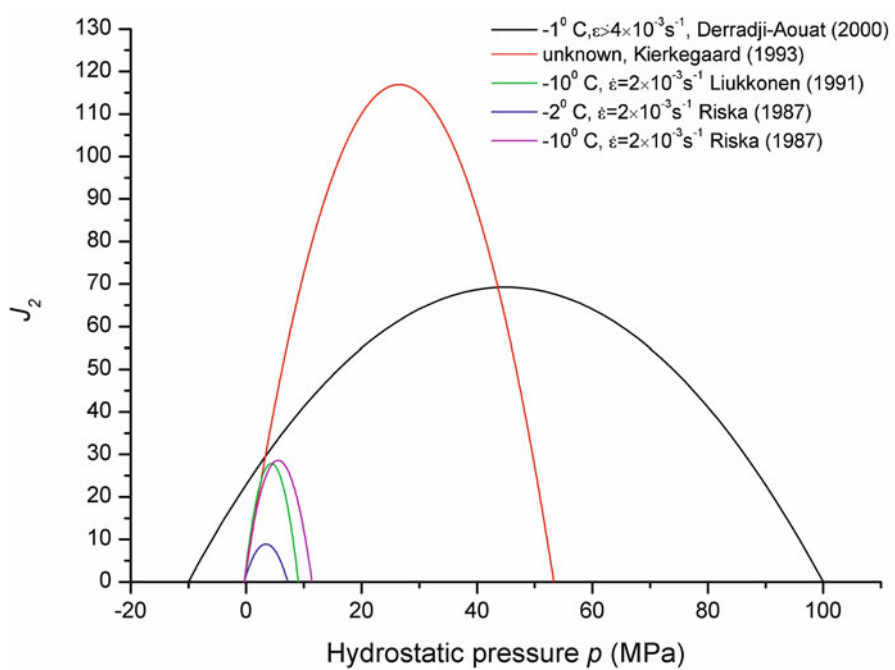
$$\varepsilon_{eq}^p = \sqrt{\frac{2}{3} \varepsilon_{ij}^p : \varepsilon_{ij}^p} \quad (19)$$

$$\varepsilon_f = \varepsilon_0 + \left(\frac{p}{p_2} - 0.5 \right)^2 \quad (20)$$

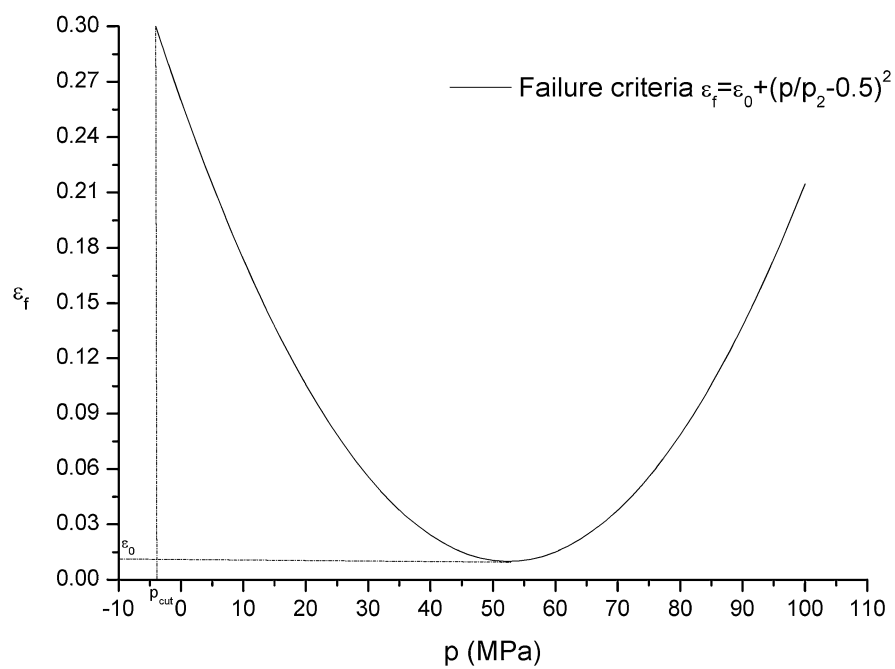
where ε_{eq}^p is the equivalent plastic strain; ε_{ij}^p is the plastic strain tensor; ε_f is the failure strain; ε_0 is the initial failure strain, which should be adjusted

according to experimental data; and p_2 is the larger root of the yield function. If $\varepsilon_{eq}^p > \varepsilon_f$ or if the pressure is less than the cut-off pressure p_{cut} , erosion is activated. This failure criterion is based on trial-and-error and is purely empirical. Only one input parameter (ε_0) is needed (Fig. 9).

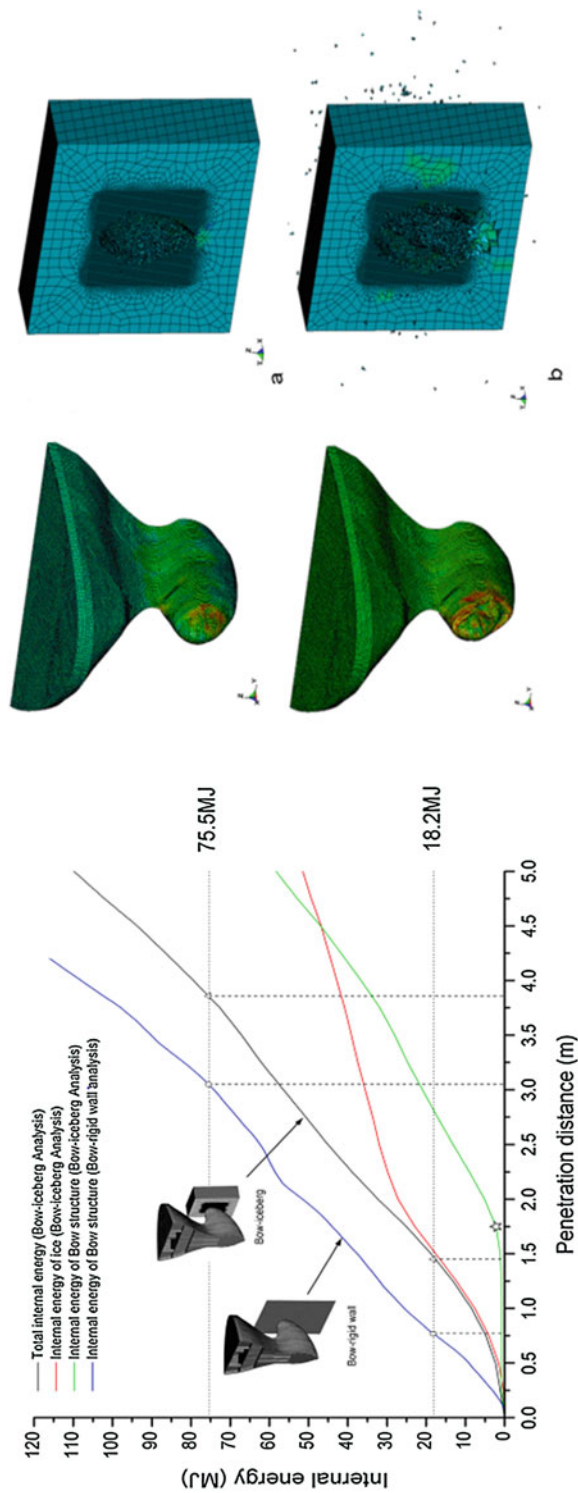
Present iceberg material model has been successfully applied to simulate the head-on collision between iceberg and ship. The simulation was carried out in the same environment as that used in the previous validations. Figure 10 shows that the strain energy was dissipated in both the iceberg and the ship structure for the bow collision with iceberg and rigid wall. In scenario A, the penetration distance corresponding to an energy dissipation of 18.2 MJ is 1.5 m; the ship structure undergoes minor plastic deformations while the iceberg dissipates most of the energy (Fig. 10a). In other words, the ship bow structure is strong enough to crush the iceberg. For scenario B, 75.5 MJ is dissipated for a penetration distance of 3.9 m. Both the ship bow and the iceberg experience large plastic deformations. Ice particles have been eroded extensively (Fig. 10b), and the bow is subjected to significant crushing behavior. It is interesting to see that there is a “maximum crushing distance” for the ship bow corresponding



Ship-Iceberg Interactions, Fig. 8 Illustration of Tsai-Wu yield surface in $p - J_2$ space from different sources, see Liu et al. (2011)



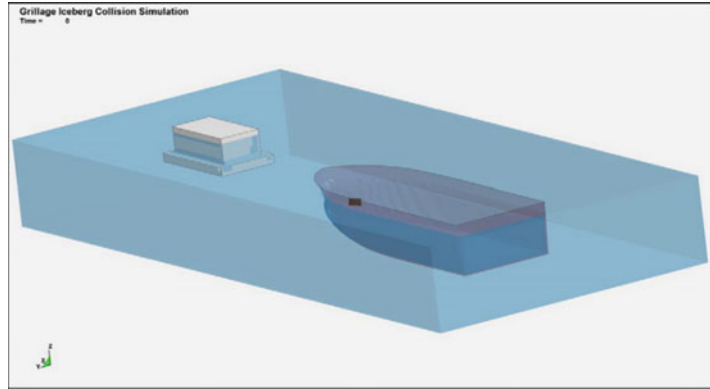
Ship-Iceberg Interactions, Fig. 9 Illustration of the failure criteria curve ($p_2 = 100 \text{ MPa}$), from Liu et al. (2011)



Ship-Iceberg Interactions, Fig. 10 Internal energy vs. penetration distance (left) and local structure/ice deformation in the simulation (right), from Liu et al. (2011), with permission of Elsevier

Ship-Iceberg

Interactions, Fig. 11 A full view of the numerical simulation model by Gagnon and Wang (2012), with permission of Elsevier



to the present ice material model. From this point, the bow structure begins to undergo significant permanent deformation (see the turning point symbol (☆) in Figs. 3, 4, 5, 6, 7, 8, 9, and 10). This takes place for a penetration of 1.7 m, which corresponds to an internal energy dissipation of 25 MJ. Using the results of the external mechanics calculations, the associated iceberg mass is estimated to be 1750 t. In other word, if the iceberg is heavier than 1750 t in the present scenario, the bow will be crushed. In future studies, a failure criterion will be included for the steel material so that tearing of the shell plating can be accounted for as well.

Complete Ship-Iceberg Interaction Simulation

Comparing to the methods mentioned above, an advanced approach could be also done by integrating the hydrodynamic, ice failure, and the ship structure failure together in the assessment. Figure 11 shows a complete model of a numerical simulation of ship and iceberg (Gagnon and Wang 2012). The drawback is that such approach is very time consuming and special knowledge to the numerical simulation tool is required.

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Short Baseline (SBL)

- Integrated Navigation
- Long Baseline Underwater Acoustic Location Technology
- Ultra-short Baseline Underwater Acoustic Location Technology

Short Pile and Uplifting Capacity

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Introduction

Offshore floating infrastructures, such as the traditional FPSOs and the emerging floating wind turbines, must be held in position to resist winds, waves, and currents in harsh environments. Anchor piles are a widely used type of foundations in offshore engineering to provide the required tensile load. For the new generation of renewable facilities which have relatively smaller dimension, short piles are often selected to resist the load passing through a catenary or taut wire mooring chain. Normally, uplifting load is imposed on the short piles.

These anchor piles have relatively larger stiffness with smaller embedment depth compared to the traditional piled foundation subjected to compressive load. Thus, short piles mostly behave as a rigid foundation and its deflection is usually negligible.

Various approaches are suggested to identify the short piles. Generally, there are two traditional categorizations. Bierschwale et al. (1981) pointed out that piles with L/d (embedded length/pile diameter) less than 6 could be considered as short (rigid) pile. Poulos and Davis (1980) proposed a criterion based on the elastic theory

analysis in the form of flexibility factor, K_r , which is given by

$$K_r = E_p I_p / E_s L^4 \quad (1)$$

where E_p is pile elastic modulus; I_p is inertia moment of pile cross section; E_s is elastic modulus of the surrounding soil. They suggested that if the flexibility factor is less than 10^{-5} , flexible pile behavior is expected, whereas if it is over 0.01, the pile will be fairly rigid.

Ultimate Capacity

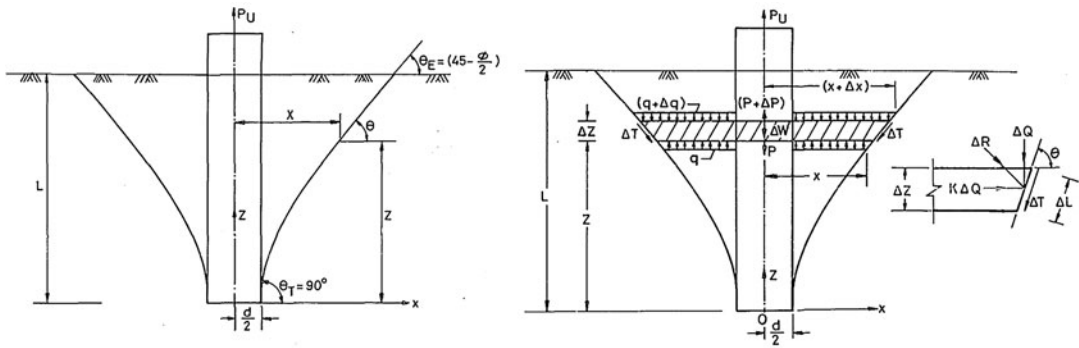
Uplift Resistance

The shaft friction capacity in compression, Q_s , is calculated as local shaft friction at failure along the shaft of a pile, τ_f , and the depth, z , as:

$$Q_s = \pi d \cdot \int \tau_f dz \quad (2)$$

where τ_f is local shaft friction at failure along the shaft of a pile; z is depth. The uplifting capacity of piles consists of shaft friction and the weight of the pile. The pile capacity in tension, Q_u , is less than or equal to, but shall not exceed Q_s . In the previous design code, it is mostly considered that Q_u is equal to Q_s for the piles in clay. But for the piles in sand, a factor which is less than 1.0, e.g., 0.75 in UWA-05 (Lehane et al. 2007), is suggested. Furthermore, in computing the tensile uplifting capacity of the pile, the weight of the pile and the soil plug shall be considered. However, no difference is considered between long piles and short piles, basically. In the following content, some design formula for investigating shaft friction, τ_f , for piles in clay and sand are presented.

Chattopadhyay and Pise (1986a) provided a theoretical method for calculating the uplifting capacity of a circular vertical pile embedded in sand using limiting equilibrium. The failure surface is assumed curved and passing through the surrounding soil mass. The lateral horizontal extent of the failure surface is dependent on the friction angle of the surrounding soil, ϕ , interface friction angle, δ , and the slenderness ratio $\lambda = L/d$.



Short Pile and Uplifting Capacity, Fig. 1 Pile and failure surface (Chattopadhyay and Pise 1986a)

The axis system and configuration of the pile are shown in Fig. 1. During uplift loading, an axisymmetric solid body of revolution of soil along with the pile is assumed to be initiated to move up along the resulting surface. The movement is resisted by the mobilized shear strength of the soil along the failure surface and the weight of the soil and the pile. Some assumptions have been made. The lateral horizontal extent of the failure surface from the axis of the pile is maximum for $\delta = \phi$, and it gradually increases and keeps constant when δ increases to a particular value. For $\delta = 0^\circ$, the inclination of the failure surface with the horizontal at the ground surface approaches 90° .

With the pile and the proposed failure surface as shown in Fig. 1, it is assumed that the ultimate uplift capacity of the pile is attained in the limiting equilibrium condition, when the mobilized shear strength of the soil along the failure surface and the weights of the body of the soil and pile balance the applied uplift forces. A circular-disc wedge contained between the axisymmetric failure surfaces is considered, which is of thickness Δz at a height z above pile tip. The failure surfaces for different slenderness ratios are evaluated from the following equation, which was proposed by

$$\frac{X_G}{d} = \frac{1}{2} + \frac{\lambda \exp(-\beta)}{\beta^2 \tan\left(45 - \frac{\phi}{2}\right)} \left(1 - \frac{1}{\beta}\right) \quad (3)$$

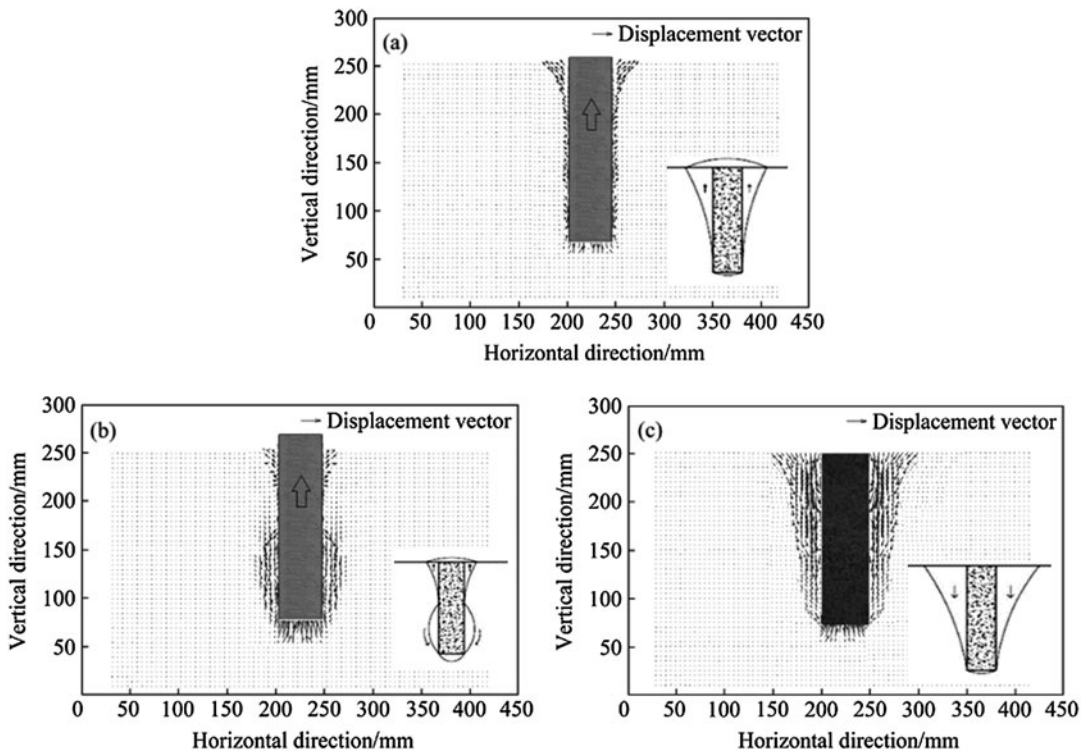
where β is $(50 - \phi)/(2\delta)$. Eventually, net uplift capacity P_u (NET) is given by ϕ

$$P_u = A\gamma\pi dL^2 - \frac{\pi d^2}{4}\gamma L \quad (4)$$

where uplift capacity factor $A = \frac{1}{L} \cdot \int_0^L \times \left\{ \frac{2x}{d} \left(1 - \frac{z}{L}\right) (\cot \theta + (\cos \theta + K \sin \theta) \tan \phi) \right\}$; coefficient of lateral earth pressure, $K = (1 - \sin \phi) (\tan \delta / \tan \phi)$.

A similar study was presented by Faizi et al. (2015). A series of small-scale physical modelling tests were conducted to investigate the uplift resistance of piles in loose sand, and PIV was utilized to observe the bearing mechanism. Fig. 2 shows the displacement field derived from the PIV analysis in loose sand when the pile was at peak resistance, post-peak, and fully removed, and a complete deformation mechanism related to three bearing stages was presented. Peak resistance was associated with the formation of a sliding surface bounded by a pair of distributed dilating shear zones. Normality was not obeyed, so a limiting equilibrium method could better represent the failure mechanism than an upper bound strict plasticity. The infilling occurs with large downward displacement of the sand particles near the piles. As the soil closed the pile fell, the uplifting zone got narrower, and the uplifting zone was steeper sided. This flow-around mechanism with a maximum wide curve around the pile was mobilized at a displacement of almost the same embedment pile length.

An extended failure surface is widely accepted for the cases of pipes and flat anchors under uplifting (White et al. 2008). At present, this



Short Pile and Uplifting Capacity, Fig. 2 failure mechanism for piles by PIV method by particle movement at: (a) peak resistance; (b) flow around stage; (c) fully removing of pile. (Faizi et al. 2015)

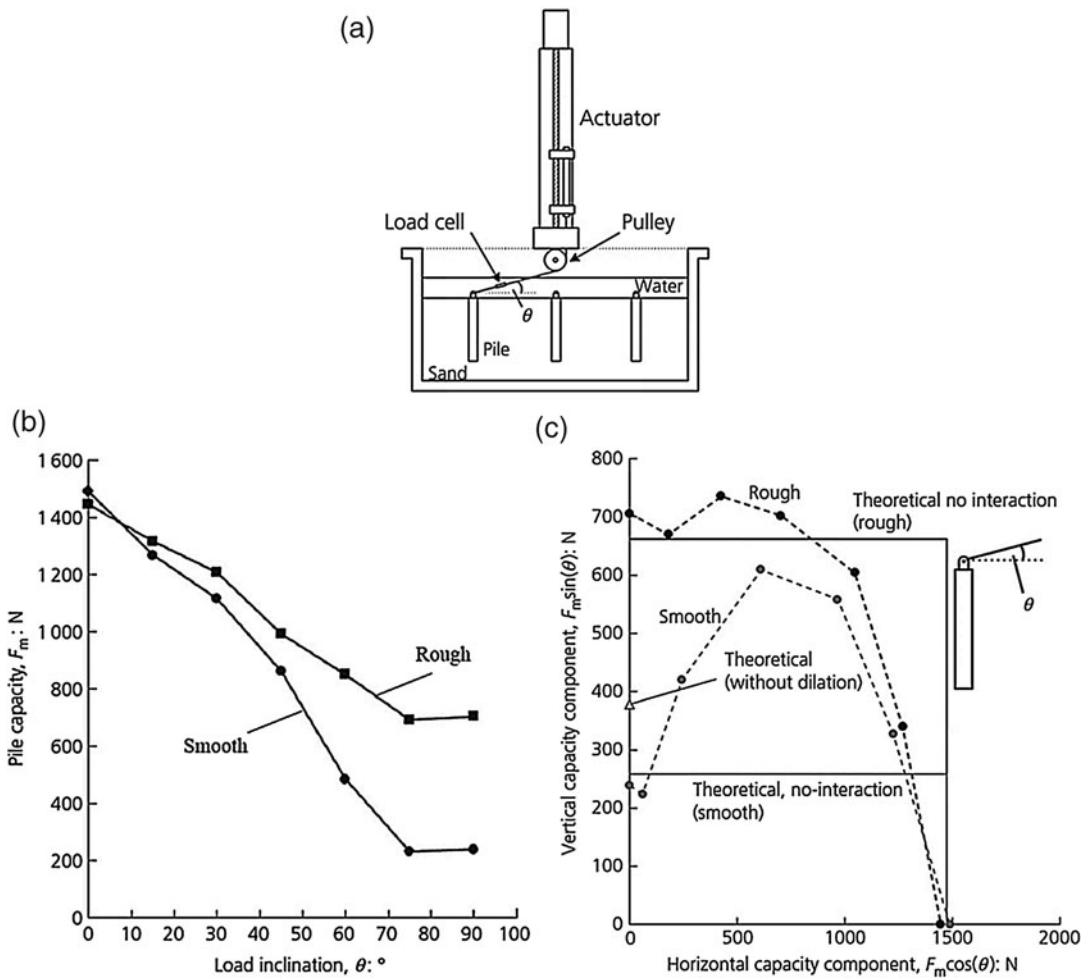
failure mechanism is not popular for long piles yet. But for the short piles in sand, rough pile shaft with large interface friction angle, δ , can lead to a failure surface which forms inside the surrounding soil. Additionally, dilation induced by upward loading increases radial effective stress on the shaft and failure tends to happen in the distant soil with relatively low effective stress. In general, the bearing mechanism considering the extent failure surface is more applicable for short pile while calculation may be significantly conservative using traditional methods ignoring the slenderness ratio $\lambda = L/d$.

Inclined Tensile Resistance

Theoretically, the failure mechanism of the short pile under inclined tensile loading is more complex than a long pile. The anchor pile resists pullout as a combination of bending plus passive resistance and skin friction shear. A simplification of piles compared to shallow foundations is that

the interaction between combined loads is less significant: the application of a horizontal load does not significantly affect the vertical capacity and vice versa. This is because horizontal load is resisted over the upper few diameters of the pile (3–6 d), whereas vertical load is resisted on the lower part of the pile, where the stress and soil strength is generally higher. This conclusion is available for long piles. However, for short piles both lateral and axial pullout load components will interact. Skin friction will be reduced in the upper part of the pile due to the gap formation surrounding the pile during loading.

Additional insights into the interaction of short piles can be obtained from test results in the literature (Patra and Pise 2006; Kong et al. 2015). Fig. 3 shows the results of short piles in centrifuge tests (Huang et al. 2019). The experiments were conducted in a geotechnical centrifuge at 100 g using aluminum closed-end piles with length, $L = 155$ mm, diameter, $d = 22$ mm, wall



Short Pile and Uplifting Capacity, Fig. 3 Centrifuge tests for short piles: (a) Pile loading arrangement; (b) Variation of pile capacity; (c) Interaction of vertical and horizontal components (Huang et al. 2019)

thickness, $t = 4$ mm, and mass, $m_p = 136$ g. The surface of each pile was either anodized or bead blasted such that the piles could be categorized as either (almost) smooth or rough. A fine-to-medium subangular silica sand was used. The samples were first prepared by air pluviation with the piles “pre-located” in the container, before saturation from the sample base. An electrical actuator was used to load the piles, either in displacement control for monotonic loading. Loading was achieved using a steel wire that was connected to a point 12 mm above the pile head and the vertical axis of the actuator. Variations in the load inclination, θ (the angle of inclined load with the horizontal plane), were

achieved by presetting the vertical and horizontal distance between a pulley located at the base of the actuator and the pile head. Pile capacity was measured using a load cell located in series with the mooring line. A displacement rate of 0.1 mm/s was adopted for the monotonic tests. Capacity reduces as the load inclination becomes more vertical, reflecting the transition in failure mechanism from a combination of passive and active soil resistance along the pile under pure horizontal loading, to pure shearing at the pile-soil interface or inside the soil under vertical loading (Fig. 3b). The effect of interface roughness is most pronounced for load inclinations between $\theta = 90^\circ$ and 45° ; below 45° the difference in capacity

for the rough and smooth piles reduces from 15% at $\theta = 45^\circ$ to 5% at $\theta = 0^\circ$. Fig. 3c shows that as the load inclination reduces from $\theta = 90^\circ$ to 0° , the vertical capacity component initially reduces (at $\theta = 75^\circ$) and then increases up to $\theta = 60^\circ$ (rough pile) and $\theta = 45^\circ$ (smooth pile). When the load inclination further reduces, passive resistance is mobilized, leading to pile capacity increases that are more significant for the smooth pile than the rough pile. Fig. 3c also shows the theoretical interactions for the rough and smooth pile, assuming no coupling between the vertical and horizontal capacity components – that is, the shape of the interaction diagram is rectangular.

Experience of how piles perform under tensile, inclined loading from inclined taut moorings is much less. Previous experimental results indicate that the ultimate oblique pulling resistance is a continuous function of θ . Available approaches of inclined tensile capacity are dependent on its vertical and horizontal capacity. A semi-empirical method has been proposed by Chattopadhyay and Pise (1986b) to predict the ultimate inclined pulling resistance, P_θ , in terms of $m = P_u/P_l$ and θ .

$$P_\theta = P_u \cos^2 \theta \exp \left(- \left(\frac{1-m}{1+m} \right) \frac{90-\theta}{90} \right) + P_l \sin \theta \exp \left(- \left(\frac{1-m}{1+m} \right) \frac{\theta}{45} \right) \quad (5)$$

where $P_u = P_{\theta=90}$ is net ultimate uplift load for axial pulling; $P_l = P_{\theta=0}$ is ultimate resistance for horizontal pulling.

An alternative method was suggested by Meyerhof (1973), which applied an approximate parabolic interaction relationship between the limits of vertical and horizontal pulling resistance, P_u , and P_l , respectively.

$$\frac{P_\theta \sin \theta}{P_u} + \left(\frac{P_\theta \cos \theta}{P_l} \right)^2 = 1 \quad (6)$$

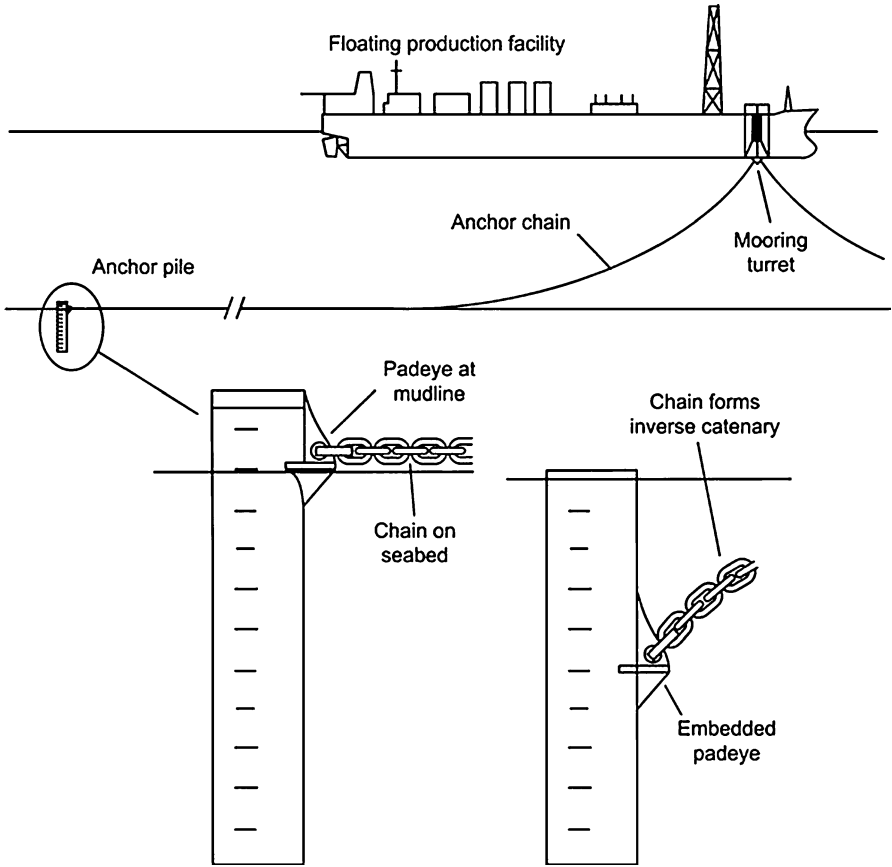
Key Applications

The main types of floating offshore platforms are tension leg platform, SPAR platform, semi-submersible platform, and Floating production,

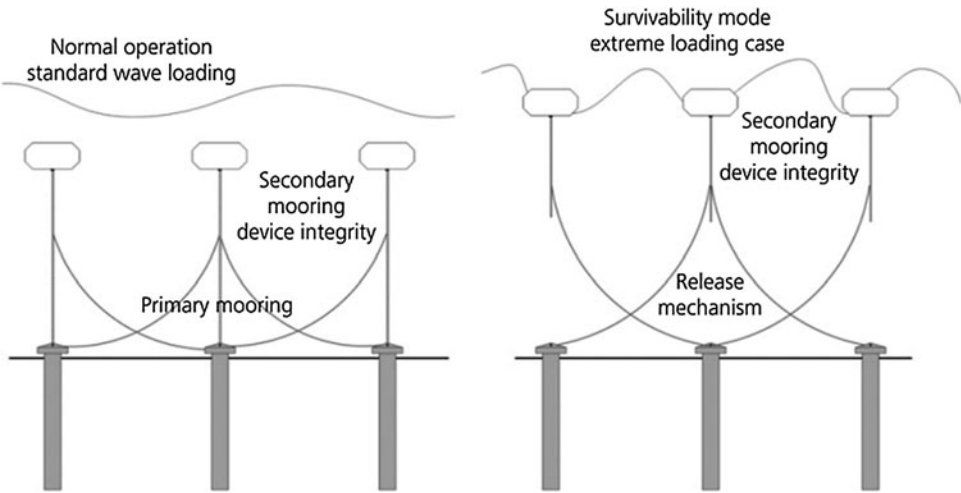
storage and offloading platforms (FPSOs), marine energy converters, etc. Short piles, which are short and stubby, with a large diameter, are more popular for the facilities like FPSOs, wind turbines, and wave energy converters (WECs). In shallow water, a length of the anchor chain will rest on the seabed during calm conditions, with the small mooring loads being resisted by friction against the seabed. In deep water, a taut line mooring is common, with the mooring line angled at 35° to the horizontal or steeper.

Fig. 4 shows the illustrations of typical anchor pile moorings. The chain may be connected to the anchor pile at the soil surface, or it may be connected via an embedded pad eye, which provides a more efficient arrangement and inclined tensile load. For suction caissons with lower L/d than short piles, the pad eye location is in the range 0.1–0.75 L indicated by Houlsby and Byrne 2005. The alternative method for pad eye depth is according to moment equilibrium assuming no rotation under lateral loading to achieve the maximum lateral load carrying capacity.

A novel arrangement of short piles for WECs (Gaudin et al. 2017) is shown in Fig. 5. The avoidance of the extreme load on floating body may be achieved by adopting different strategies. The extreme load can be by mechanically lowering the floating body in the water, or survived by incorporating a fuse in the primary mooring. The fuse disconnects the device from the primary mooring line on time, and then the extreme mooring line loads transfers to a number of secondary moorings. When the WEC is producing power under normal operating conditions, WEC motion produces extension of the power take off, creating a cyclic mooring line force that is transferred to the foundation. WEC foundations are also required to withstand the loading associated with extreme events, during which the WEC will not be producing power, but is in “survival mode.” One option for dealing with the extreme loading event for such a mooring configuration is the use of secondary slack moorings. When the primary mooring is disconnected during extreme loads, the extreme loads acting on the WEC are then shared across a number of secondary moorings that are connected to adjacent foundations. In



Short Pile and Uplifting Capacity, Fig. 4 Anchor piles in mooring systems (Randolph and Gourvenec 2011)



Short Pile and Uplifting Capacity, Fig. 5 Mooring configurations for point absorber WECs (Gaudin et al. 2017)

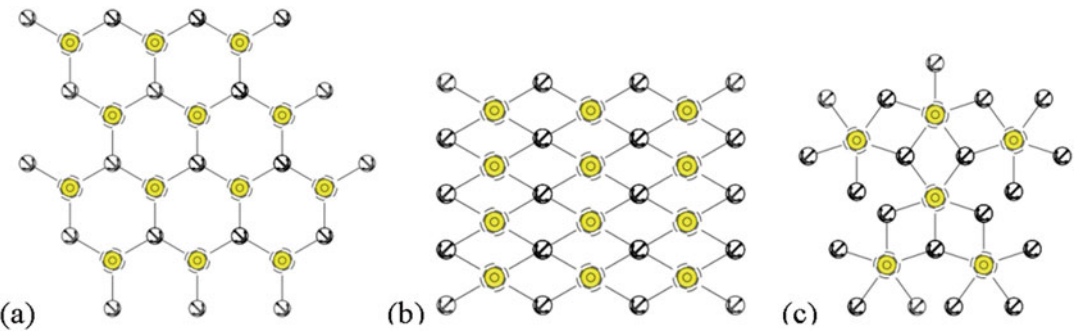
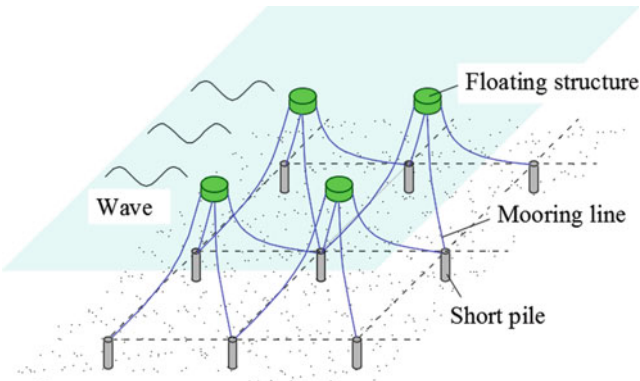
this arrangement, short piles are supposed to subject vertical or inclined tensile load.

Some similar ideas, such as anchor sharing, as shown in Fig. 6, were proposed to promote anchor efficiency for both floating wind turbines (Breton and Moe 2009) and wave energy converters (Gao and Moan 2009). Herduin et al. (2018) indicated that the main benefit was lesser number of the required anchors. Fig. 7 and Table 1 show several “anchor efficient” array configurations for 3 point, 4 point, and 5 point mooring systems. As evident from Table 1, the array (a) is using the least number of anchors and therefore provides the greatest potential for cost savings from a

geotechnical perspective. The arrangement represented in Fig. 7c with a 5-point mooring system makes the array arrangement less efficient in terms of anchor’s numbers. Each of the array arrangements implies that the anchors are connected and loaded by one to six mooring lines. The individual load passed by mooring is multidirectional. Unlike the drag anchors whose bearing planes are kept normal to the loading orientation, short piles show the better applicability for resisting multidirectional loading. However, the loading applied to the short piles is complex. Fig. 8 shows the variety of loading configurations. θ_F is the phase angle of the wave

Short Pile and Uplifting Capacity,

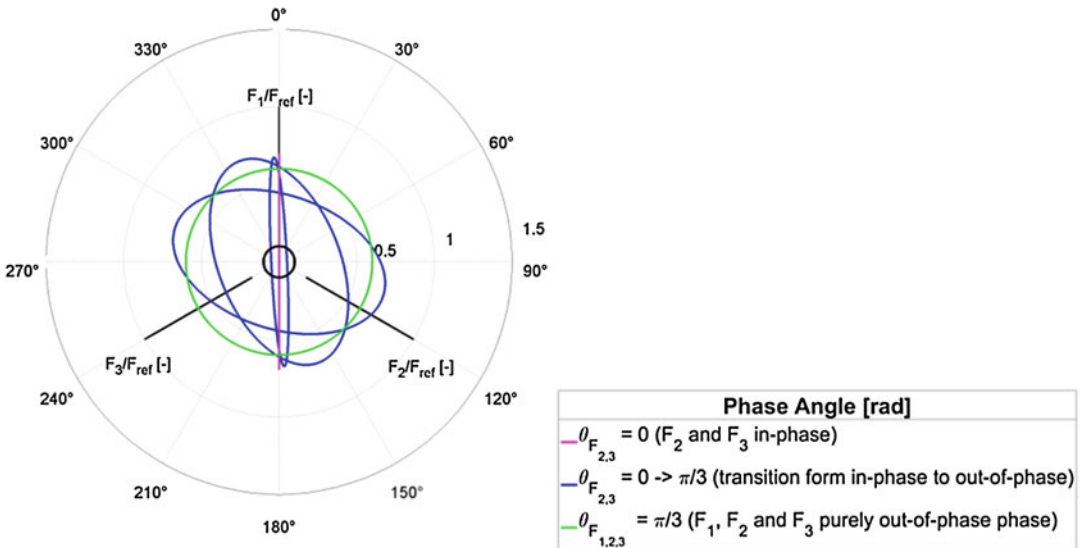
Fig. 6 Conception of anchor sharing



Short Pile and Uplifting Capacity, Fig. 7 Multiple “anchor efficient” array arrangements for: (a) 3-point; (b) 4-point; (c) 5-point (Herduin et al. 2018)

Short Pile and Uplifting Capacity, Table 1 Characteristics of “anchor efficient” arrays (Herduin et al. 2018)

Arrays	Fig. 7a	Fig. 7b	Fig. 7c
Mooring system	3a-point	4-point	5-point
Devices	13	12	6
Anchors	20	20	20
Anchors per device	1.5	1.7	3.3



Short Pile and Uplifting Capacity, Fig. 8 Variation of magnitude and direction of the load resultant from three harmonic loads of phase θ_F . (Gaudin et al. 2017)

arriving adjacent three floating body. As the phase difference between the three loads varies from 0 to π , resultant contours transform from a thin ellipse suggesting nearly bidirectional loading to a circle centered at the origin indicating constant amplitude and large variation in direction. This variety is much more complex when consideration is given to irregular waves and wave direction.

Cross-References

- [Monopile Foundations in Offshore Wind Farm](#)
- [Mooring System](#)
- [Offshore Wind Turbines](#)
- [Pile Capacity](#)
- [Wave Energy Converters](#)

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Side-Scan Sonar

- [Sonar Technology](#)

Signal-to-Noise Ratio (SNR)

- [Underwater Acoustic Communication](#)

Significant Wave Height

- [Ship Operational Environment](#)

Simple Shear

- [Shallow Foundations](#)

Simulation Tools

- [Underwater Acoustic Sensor Network](#)

Simultaneous Localization and Mapping

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Synonyms

[Localization and planning](#); [Simultaneous localization and mapping \(SLAM\)](#); [Underwater navigation and positioning](#)

Definition

Simultaneous localization and mapping (SLAM) is a technology of constructing or updating a map of an unknown environment through computational method while simultaneously keeping track of a robot's location within it. In recent years, SLAM technology has achieved great success in the field of underwater robot navigation and positioning. It enables underwater robots to explore and navigate in unknown marine environments; generate their own maps for the environment; be able to identify their location, landmarks, and obstacles; and make decisions based on the received new environmental data.

Scientific Fundamentals

Navigation and positioning are critical to the safety and effectiveness of underwater vehicle operations. The unavailability of the global satellite positioning system (GPS) has made underwater robotic navigation a challenging technical issue. In recent years, the performance of sensors applied to underwater vehicles has greatly improved, and the cost and size of these sensor devices have been greatly reduced. With the development of these technologies, advances in algorithms such as simultaneous positioning and mapping and collaborative navigation have greatly improved the navigation capabilities of

underwater vehicles. These improvements in underwater robotic navigation have contributed to the successful deployment of underwater vehicles in a variety of applications. Techniques have been introduced in finding a suitable technique to solve challenges faced by underwater vehicles as mentioned above. The aim is to improve and optimize the underwater vehicle performance in terms of landmark estimation, posterior and location estimation, effective path planning, and reduction of error made by the vehicle. When simultaneous localization and mapping (SLAM) are introduced, it became the most recognized technique by researchers (Leonard and Durrant-Whyte 1991; Paull et al. 2013).

Evolution of SLAM

In the decades since SLAM technology was introduced into the field of underwater robots, SLAM technology has been widely used. The development of SLAM has continued to this day to improve the efficiency and quality of SLAM algorithms in terms of accuracy, estimation, and reduction of error rates.

Durrant-Whyte (1986) had introduced the concept of implementing the estimation of spatial uncertainty at the IEEE Robotics and Automation Conference in San Francisco. Smith et al. (1987) had described a representation for spatial relationships which makes explicit their inherent uncertainty. Their researches are considered the starting point for the pre-development of SLAM technology. Leonard and Durrant-Whyte (1991) developed the SLAM technique based on the previous work using the probabilistic approach in solving the SLAM problem. That paper systematically introduces the implementation of extended Kalman filtering method.

In order to improve the efficiency of the SLAM algorithm, Montemerlo et al. (2002) introduced the famous FastSLAM algorithm. This SLAM algorithm is one of the most recognized SLAM methods after EKF-SLAM. The idea of FastSLAM is to use a hybrid method that integrates the particle filter and extended Kalman filter methods together. This implementation makes FastSLAM a high data accuracy and is therefore welcome. FastSLAM continued to

evolve and eventually formed the second version of FastSLAM, called FastSLAM 2.0. The modification made in FastSLAM 2.0 is that the distribution of proposals must rely on both the previous pose estimation and the actual measurement of the mobile robot, while the first version of FastSLAM only relied on the previous pose estimation of the mobile robot (Montemerlo et al. 2003).

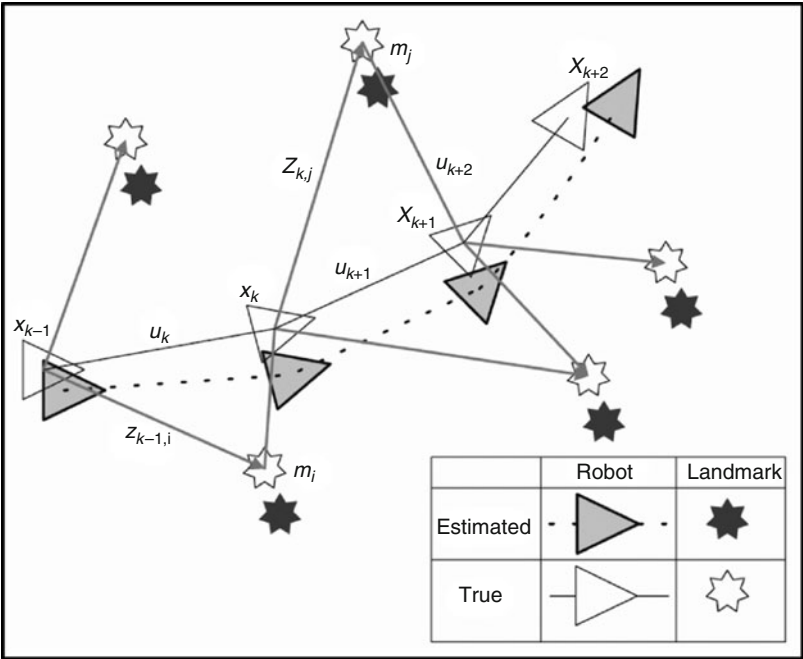
Dellaert and Kaess (2006) proposed the smoothing method called square-root smoothing and mapping (SAM) for the improvement of the mapping process of mobile robots. The technique uses the square-root information smoothing approach in solving the SLAM problem in improving mapping process efficiency. Kim et al. (2007) introduce a new technique that provides a robust algorithm based on a scale unscented transformation called UFastSLAM. The technique proposed to improve the FastSLAM method by using the scale unscented transformation algorithm approach.

SLAM algorithm keeps improving and enhanced its reliability from the first introduced SLAM algorithm until recent SLAM development. Different SLAM algorithms try to tackle and improvise different aspects of SLAM problems. The ultimate objective in the SLAM algorithm is to realize the challenge of optimum navigation, localization, and mapping, that is, the capability of underwater robots to self-explore with no human interference in dynamic environments given a time period. SLAM development shows that numerous implementations have been applied to SLAM to deal with the more dynamic environment and multiple vehicle navigation (Khairuddin et al. 2015).

Mathematical Model for SLAM

The goal of SLAM is to make vehicles generate a relatively consistent environment map and use the generated map to estimate landmarks and vehicle positions at the same time. An important fact of SLAM is that it does not require any prior knowledge of mobile robots and landmark locations, as platform trajectories and landmark locations are estimated and calculated online. Suppose the vehicle explores the entire unknown environment and uses sensor devices

Simultaneous Localization and Mapping, Fig. 1 The basic implementation of feature-based SLAM (Khairuddin et al. 2015)

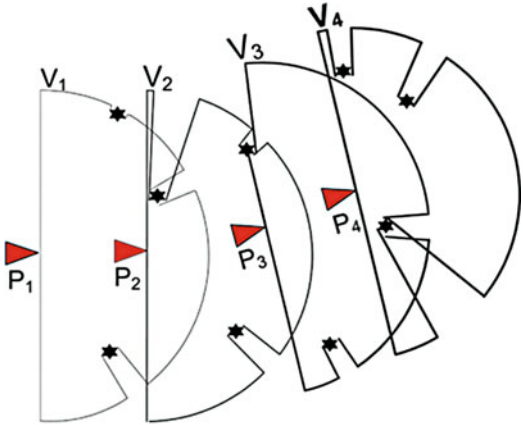


to observe and estimate the location of landmarks in the environment. Figures 1 and 2 show the basic implementation of feature-based SLAM.

In Fig. 1, both the robot and landmarks' location are estimated simultaneously, and the true robot and landmarks' location are unknown and not measured. Observations done are between true robot and landmark locations. The following variables are defined at time k .

- x_k Robot location
- X_k Sequence of robot location or path
- u_k Odometry between time $k-1$ and k
- U_k Sequence of robot odometry or relative motion
- m_i Map comprised of landmarks, objects, surfaces, and their respective locations
- Z_k Sequence of measurements between robot and landmarks assuming one measurement per time step

It is not appropriate to rely only on robot odometry U_k to determine robot location within a plane because, in the real application, it suffers the lack of precision required for accurate localization. Taking into consideration, the robot not



Simultaneous Localization and Mapping, Fig. 2 The basic implementation of view-based SLAM (Paull et al. 2013)

only can depend on odometry measurement alone but also relies on continuous sensory measurement Z_k for more accurate true location estimation. Once all the required information is obtained and defined, the next steps are building a map view of the environment and location prediction. SLAM technique is using the

probabilistic approach which applies a probability distribution to predict the robot and landmarks' location from the generated map. The probability distribution form, P , is defined as:

$$P(x_k, m | Z_k, U_k)$$

The above formula can be interpreted as the probability of the position at time k , and the map is drawn based on the history of the measurement and odometer data. The observation model of the robot can identify the relationship between the robot position x_k and the ranging u_k , which is defined as:

$$P(x_k | x_k, u_k)$$

Motion model of the robot identifies the relationship between sensory measurements z_k , map environment m , and robot position x_k which is defined as:

$$P(z_k | x_k, m)$$

In general application, the robot is able to detect landmark range, relative direction, and the unique identity in the typical environment. These understandings can be used as a basis to derive a measurement model. By applying probability distribution into the measurement model, it can be defined as:

$$P(z_k | x_k, m) \tilde{N}(h(x_k, m), Q_k)$$

where $h(x_k, m)$ is an arbitrary function that represents sensory equipment operation and Q_k is two-dimensional noise covariance.

For the motion model, a normal distribution is applied which focuses on kinematic motion model covariance. The motion model derivation is defined as:

$$P(x_k | x_k, u_k) = N(g(x_{k-1}, u_k), R_k)$$

where $g(x_{k-1}, u_k)$ is standard kinematic function and R_k is a three-dimensional noise covariance.

The robot will update its position and landmark measurement at the previous time step $k-1$ with

improved precision from the data it gathers from other landmark and previous robot position at time step k . The process is done continuously to update the robot measurement until the robot finishes its exploration in the environment (Khairuddin et al. 2015).

Filtering Approaches to SLAM

SLAM filtering methods use state estimation algorithms such as EKF-SLAM. Some methods use a combination of multiple filtering methods. The following sections describe some of the most popular SLAM method categories and their advantages and disadvantages.

EKF-SLAM

It uses Taylor expansion to linearize the system model. It applies recursive prediction-update cycles to estimate poses and maps. Its state vector includes poses and features. It applies to view-based SLAM and function-based SLAM. For large maps, since the calculation time is directly proportional to the number of features, EKF-SLAM is computationally intensive (Eustice et al. 2006).

SEIF-SLAM

Sparse extended information filter (SEIF) and precise sparse extended information filter (ESEIF) are two well-known methods of SLAM using information filters. They all maintain a sparse information matrix and therefore maintain the consistency of the Gaussian distribution. However, accessing the mean and covariance requires a large matrix inversion that is computationally intensive. Both methods need to actively “sparsify” the information matrix through a refinement strategy. ESEIF maintains an information matrix where most of the elements happen to be zero (Walter et al. 2007).

FastSLAM

It is based on a particle filter. The particle filtering method is a nonlinear filtering solution. Therefore, the system model is not approximate. In FastSLAM, poses and features are represented by particles (points) in the state space. FastSLAM is the only solution that can perform online SLAM and full SLAM simultaneously, which means that

it can estimate not only the current pose but also the entire trajectory. In FastSLAM, each particle contains poses and estimates for all features. However, each function is represented and updated by a separate EKF. Similar to other methods, it applies to view-based SLAM and feature-based SLAM (Montemerlo et al. 2002).

GraphSLAM

In the GraphSLAM method, the entire trajectory and map will be estimated. GraphSLAM also uses Taylor expansion for approximation, but it differs from EKF-SLAM in that it accumulates information and is therefore considered an offline algorithm. Generally, in GraphSLAM, the pose of a robot is represented as a node in the graph. Model the edges of connected nodes using motion and observation constraints. These constraints need to be optimized to calculate the spatial distribution of nodes and their uncertainty. For GraphSLAM, there are different solutions, such as slack on the grid, multilevel relaxation, iterative alignment, square-root smoothing and mapping, incremental smoothing and mapping, and hierarchical optimization of pose maps on the manifold. In principle, they are all similar, but optimization is implemented differently (Paull et al. 2013).

Artificial Intelligence SLAM

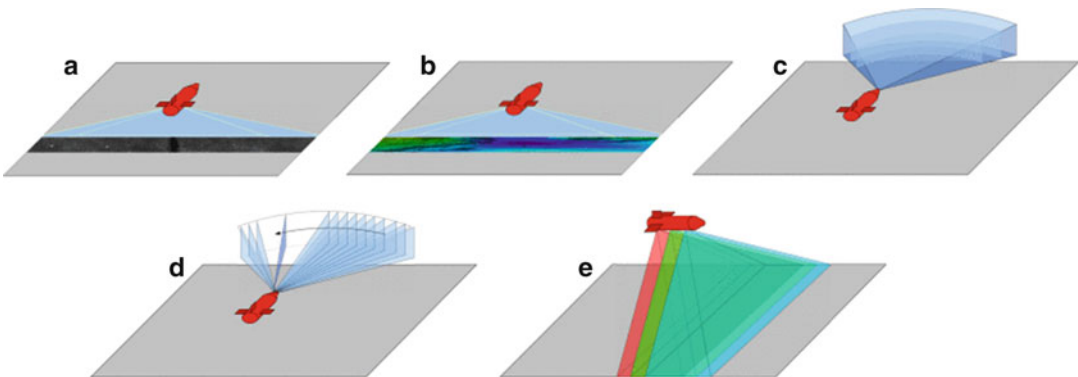
These methods of SLAM are based on fuzzy logic and neural networks. RatSLAM is a technique for modeling rodent brains using neural networks

(Wyeth and Milford 2009). In practice, this method uses a camera and an odometer for neural network-based data fusion. Saeedi et al. (2011) use self-organizing maps to perform SLAM with multiple robots. A self-organizing graph is a neural network that is trained without supervision.

Key Applications of SLAM in Underwater Vehicles

In traditional navigation methods, it is not possible to determine when the AUV will automatically locate without localization support for ships or sound transponders. With dead reckoning (DR) using inertial measurement units, the AUV increases its position based on knowledge of its direction and speed or acceleration vector. Traditional DR is not considered as the main navigation means, but modern navigation systems that rely on DR are widely used in AUVs. The disadvantage of DR is that errors are cumulative. Therefore, the AUV position error increases infinitely with the travel distance.

In recent years, through the development of underwater camera and sonar technology, underwater robots have gained more abundant marine environment information, as shown in Fig. 3. Therefore, underwater robot navigation technology based on sonar and visible light images has developed rapidly. Almost all methods in this category that implement boundary position errors use some form of SLAM.



Simultaneous Localization and Mapping, Fig. 3 Sonar sensor swaths: (a) side-scanning, (b) multi-beam, (c) forward-looking, (d) mechanical scanning and imaging, and (e) synthetic aperture (Paull et al. 2013)

Optical-Based SLAM

The visual ranging method is a process of determining the pose of the robot by analyzing subsequent camera images. This can be achieved by moving the optical flow or structure (SFM). Invariant extraction and representation of features is an important consideration. Many previous algorithms have been proposed and applied to ground and aerial robotics, such as scale-invariant feature transform (SIFT), accelerated robust features (SURF), etc. Images can be captured using a stereo or monocular camera. Another advantage of a stereo camera is that a full six-degree-of-freedom transform can be found between successive pairs of images. The main challenge is to close the loop in the trajectory by associating discontinuous images, which is necessary to limit the positioning error. Limitations of optical systems in underwater environments include reduced camera range, sensitivity to scattering, and insufficient light. Therefore, visible-wavelength cameras are usually mounted on hovering AUVs because they can approach objects of interest. In addition, visual odometers and feature extraction depend on the presence of features. Therefore, the underwater optical navigation method is particularly suitable for small-scale map drawing in a feature-rich environment.

Eustice et al. (2008) proposed the implementation of SLAM based on underwater vision, called visually enhanced navigation (VAN). The multi-sensor method combines the advantages of optical and inertial navigation methods and is robust to the low overlap of images. This method is a view-based version of EKF-SLAM, where camera-derived relative attitude measurements provide spatial constraints for visual ranging and closed loop. The VAN method is transformed into an information form and proves that view-based ESEIF-SLAM maintains a sparse information matrix without approximations or pruning (Eustice et al. 2008). This method has been applied to the deepwater measurement of Titanic. It also solves the problem of recovering the mean and covariance from the information table. Hover et al. (2012) use contour sonar and monocular camera functions. Hover the AUV for

hull inspection by using functions expressed using full sonar and monocular camera functions. Use the DVL locked on board to get the odometer relative to the hull.

Sonar-Based SLAM

By using acoustic detection technology, the underwater vehicle could identify and classify the features that can be used as navigation landmarks in the environment. With detection sonar, features can be extracted almost directly from the combined echo. The sonars used on underwater robots mainly include (a) side-scanning, (b) multi-beam, (c) forward-looking, (d) mechanical scanning and imaging, and (e) synthetic aperture. The following section uses the side-scanning sonar and ranging sonar as examples to introduce the related SLAM methods.

Side-Scanning Sonar

The image of a side-scan sonar is shown in Fig. 3 (a). The intensity of the sound waves returned by the fan beam from the ocean floor also depends on the type of the ocean floor and is recorded in a series of cross-track slices. When subdivided along the direction of travel, these assembled slices form an image of the seafloor within the beam. Hard objects protruding from the seafloor echo strongly and are represented by dark images. Shadows and soft areas (such as dust and sand) have weak echoes, represented by bright images.

Ruiz et al. (2004) have presented the SLAM method with the side-scanning sonar using an augmented EKF and the Rauch-Tung-Striebel smoother. The value of smoothing in the pose estimation is emphasized since all previous poses should be updated when a loop closure event is detected. It is noted that automated feature detection (which is not trivial) and data association are necessary to achieve autonomy for side-scanning sonar SLAM. Aulinas et al. (2011) have approached the SLAM problem by using a submap joining algorithm called the selective submap joining SLAM. Their work used a cascaded Haar classifier for object detection from side-scanning sonar imagery and is demonstrated offline on a gathered data set.

Ranging Sonar

Figure 3 (b) shows the multi-beam sensor image. Instead of using multiple beams to point a single transducer down, multiple beams in a transducer array are aligned on the AUV hull in a precise pattern. The sound bounced off the seafloor at different angles and was received by the AUV at slightly different times. The signal is then processed on the AUV, converted to water depth, and arranged as a sounding map. The multi-beam resolution achieved depends on its sensor quality, operating frequency, and height from the seafloor. Multi-beam sounding systems are often used to map large areas of the seafloor. Each measurement line traversed by the AUV collects a data channel called a strip. Multi-beam sonars produce 2.5-dimensional detection features (elevation maps), while side-scanning sonars produce 2-dimensional images.

Barkby et al. (2012) have proposed a bathymetric SLAM algorithm called BPSLAM that is based on a featureless FastSLAM implementation. In their approach, the need for feature extraction is removed, and each particle maintains an estimate of the current vehicle state and the 2-D bathymetric map. An important issue with employing a particle filter-based system with such a large state space is the computational expense of copying particles' maps during the resampling process. This problem is solved by "distributed particle mapping" where a particle ancestry tree is maintained. Copying of particle maps is avoided by having new particles generated during the resampling process point to their parents' maps rather than copying them. Maps of leaf nodes in the tree are reconstructed by recombining the maps of all ancestors. The need to store each particle's map is removed completely by storing just the particle's trajectory and linking poses to an entry in a log of bathymetry observations. Maps are then reconstructed as needed using Gaussian process regression, and, as a result, loop closures can be achieved even in the case where there is little or no overlap between sonar images since the regression process is able to make predictions about areas of seabed that have not been directly observed (Paull et al. 2013).

Cross-References

- [Localization and Planning](#)
- [Simultaneous Localization and Mapping](#)
- [Underwater Navigation and Positioning](#)

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Simultaneous Localization and Mapping (SLAM)

► Simultaneous Localization and Mapping

Single Anchor Leg Mooring

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Synonyms

Catenary anchor leg mooring; Catenary mooring; Mooring system; Single point mooring; Taut mooring

Definition

The single anchor leg mooring (SALM) is a specific type of single point mooring (SPM) which is anchored by a tensioned leg to a base on the sea floor. A hose system and a fluid swivel concentric with the anchor leg permit cargo to

flow between the moored tanker and an underwater pipeline, while the tanker swings around the mooring (Flory et al. 1972; Luo 2015) (Fig. 1).

Scientific Fundamentals

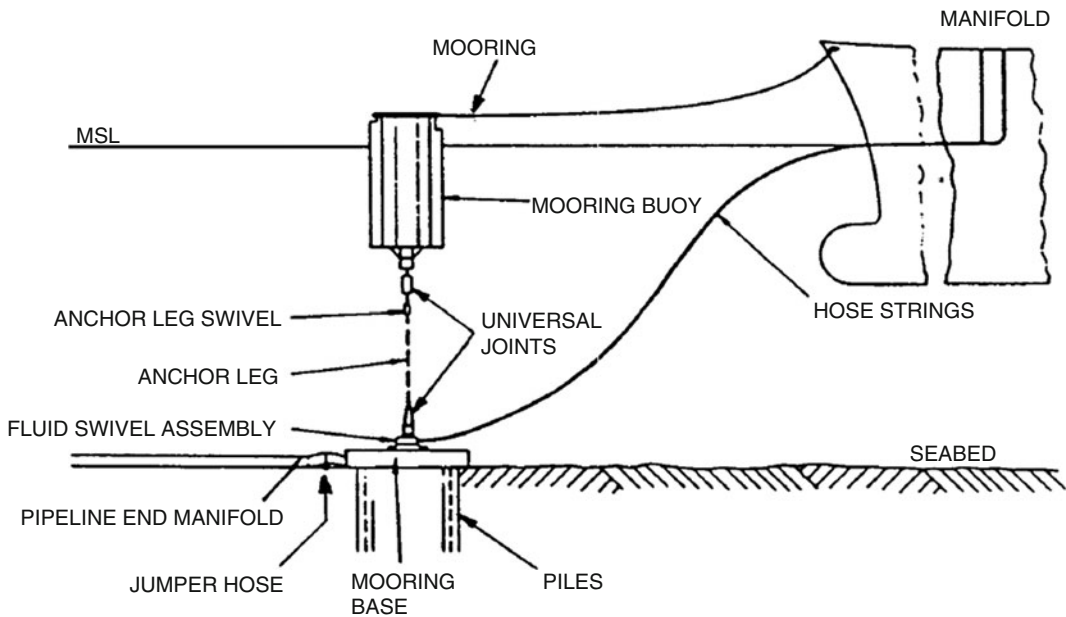
System Composition

The mooring buoy is normally steel and has typical dimensions of 4 m diameter and 12 m height. Vessels are moored to the SALM by one or more hawsers connected to the mooring buoy (ABS 2014). Hawsers are normally made from synthetic fibers with dimension depending on tanker size and local operating conditions (SBM 2012). The SALM anchoring system has two main sections, the base and the anchor leg or riser (API RP 2SK 2005). The base is the principal structural member which secures the SALM in position. The single anchor leg is usually a large chain with a swivel incorporated to allow buoy rotation and prevent chain torquing. A fluid swivel is mounted concentrically about the anchor leg either on top of the base or on top of a riser. The fluid swivel permits the passage of fluid while rotating in response to the moored vessels movement. The hose strings for the SALM extended the fluid swivel unit to the moored tanker manifold and are composed of submarine and floating hoses. (Baritrop 1998).

SALM prevents collision damage to the swivels by placing them underwater and below the keel level of the tanker. Any damage should then only affect the simple surface buoy and be relatively cheap to repair (Gruy 1977).

Historical Development

The advent of very large oil tankers in the mid 1960s resulted in a need for new concepts in tanker terminals. Standard Oil of New Jersey affiliates, including Esso Standard Libya, were faced with requirements for expanding their facilities to accommodate tankers of 200,000 dwt and larger. Since the first set of SPM was used in the Arabian Gulf in 1964, various types of SPM have appeared one after another. In 1966, Esso Research and Engineering (ERE) began a research program to develop new single



Single Anchor Leg Mooring, Fig. 1 SALM (API RP 2SK)

point moorings. At about the same time, Esso Standard Libya requested that ERE investigate the feasibility of a new mooring concept, the single anchor leg mooring (SALM), for installation at their port at Brega, Libya. A second SALM was installed at Okinawa in 1971. Experience with these facilities has demonstrated the SALM to be a very successful design. Studies have shown the SALM offers considerable investment savings compared to conventional single point moorings in water depths greater than about 100 ft. Safety and reliability are other principal advantages (Vryhof Anchor Manual 2005).

Key Technology in the Development of SALM

After the concept of SALM system was proposed in 1966, the research on SALM system has been developing rapidly.

Butenin (1965) first proposed an approximate formula for the mooring lines in a SPM system. The shear strain effect based on the nonlinear elastic theory was added to the formula. At the same time, the anchor cables and pipes in the SPM system are considered as cantilever beam with very small stiffness coefficient. The results

show that when the nonlinear part is small, the results obtained by Butenin method are in good agreement with those obtained by other numerical methods.

Sarpkaya (1976) studied the influences of Re and roughness coefficient K on resistance coefficient CD (also known as drag coefficient) and mass coefficient CM systematically and drew the curves of the relationship between Re , K , CD , and CM of cylinders of different cross-section shapes in current. Labbe et al. (1983) analyzed the effects of different hydrodynamic parameters CD and CM on riser's response; proposed CD , CM , and Re are related to wave-period coefficient KC . The results show that the dynamic response of riser is insensitive to CD and CM in most cases, so the dynamic response of riser can be calculated by using CD and CM based on maximum Re and maximum KC .

Ansari and Khan (1996) did a systematic analysis of the influence of mooring line loads on ships. Halliwell (1988) uses line element to analyze the stress of anchor mooring cable in deepwater by incremental method and compiled

a geometric nonlinear finite element analysis program. Ahmad and Datta (1989) calculated and analyzed the dynamic response of riser components of offshore oil floating production platform under wave, current, and ship motion. Halliwell and Harris (1988) used model tests to study the effects of environmental and mooring system parameters on the mooring response. Hearn and Thomas (1991) compared four numerical integration methods for dynamic analysis of mooring systems – central difference method, Newmark method, Wilson- θ method, and Houbolt method, and it is found that Wilson- θ method has shorter calculation time and smoother numerical solution, which is more suitable for dynamic simulation of mooring system.

Ma Rujian and Li Guixi (1995) studied the nonlinear response of a SPM system in stochastic waves by using nonlinear spectrum analysis method.

Bin Pan et al. (1995) proposed a static calculation method for SALM system in a stable position under the action of external forces. Zhili Zou et al. (2002) studied the motion response, mooring force and fender collision force of mooring ship on oil platform under loads of wind, wave, and current. Based on the calculation results, the effects of different water levels and different angles between wind, wave, and current on mooring force and collision force were discussed (Fig. 2).

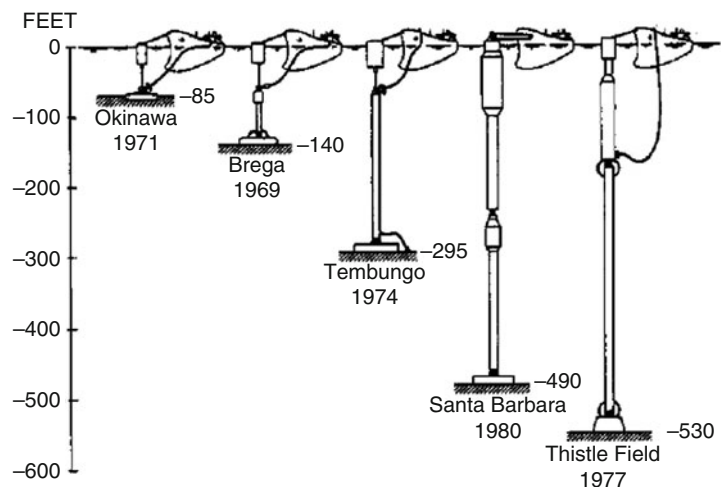
Key Applications

SALM system is usually used for shuttle tanker and FSO instead of FPSO. SALM can work in cooperation with semi-submersible drilling ships, and the prefabricated SALM can be used for supply ship transportation, installation, displacement, and positioning of vessels of various tonnage. It can be adapted to a wide range of weather conditions and water depths. It is quick and economical to construct and install, and it is suitable for retrofitted tankers (Luo 2015). The water depth of SALM reaches 500 m, deeper than CALM.

This system employs a vertical riser system that has a large amount of buoyancy near the surface, and sometimes on the surface, which is held by a pre-tensioned riser (GL 2014). The vertical buoyancy force acting on the top of the riser functions as an inverted pendulum. When the system is displaced to the side, the pendulum action tends to restore the riser to the vertical position. The tanker can be secured to the top of this SALM buoy with either a flexible hawser or a rigid yoke (Luo 2015). The base of the riser is usually attached through a U-joint to a piled or deadweight concrete or steel structure on the sea floor. In deepwater, the riser system usually has mid-span articulation.

There are two kinds of applications of SALM: one is SALM without riser and the other is SALM with riser.

Single Anchor Leg Mooring, Fig. 2 Single anchor leg mooring (SALM) with tubular riser and yoke



1. Single Anchor Leg Mooring (SALM) Without Riser

The pipe line does not pass through the buoy but combines the underwater oil hose with the surface floating hose into a hose. The system is anchored to the seabed and can withstand positive buoyancy and maximum mooring load. The installation depth of the system without riser is shallow, ranging from 20 m to 50 m, for example, the SALM system in Okinawa, Japan (Marcello et al. 1980).

2. SALM with Chain Riser

In order to increase the water depth, the number of SALM system installations with riser project has increased gradually. Different from the former, steel pipe called the riser is inserted between the anchor and the anchor chain. For instance, the installed SALM system with riser was installed in thistle oilfield in North Sea. It is 162 m deep; the riser's diameter is 2.5 m and the length is 101 m (Foolen and Smulders 1977).

As offshore activities continue to expand into the deep sea, the SALM system with riser has been put into use on a large scale. Figure 2 is the water depth and structure of several SALM systems, which was put into use from 1969 to 1981 (Marcello et al. 1980; Robert et al. 1980; Mack et al. 1981).

At present, the offshore oil industry is gradually developing into deeper waters in Gulf of Mexico, Brazil, western Africa, the north Atlantic, and the Norwegian Sea, with water depth in the range of 1000–3000 m. The operating depth of the production platform in Gulf of Mexico has exceeded 3000 m.

Cross-References

- ▶ [Catenary Anchor Leg Mooring](#)
- ▶ [Catenary Mooring](#)
- ▶ [Single Point Mooring](#)
- ▶ [Spread Mooring](#)
- ▶ [Taut Mooring](#)

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Single Anchor Leg Mooring (SALM)

- [Compliant Tower Platform](#)

Single Pile Foundation

- [Monopile Foundations in Offshore Wind Farm](#)

Single Point Mooring

- [Catenary Anchor Leg Mooring](#)
- [Mooring System](#)
- [Single Anchor Leg Mooring](#)

Single Point Mooring (SPM)

- [Compliant Tower Platform](#)

Single Point Mooring System

- [External Turret Single Point Mooring System](#)
- [Soft YOKE Single Point Mooring System](#)

Single-Input Single-Output (SISO)

- [Underwater Acoustic Communication](#)

Single-Point Mooring

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Synonyms

[Offshore Mooring System](#); [Spread Moorings](#); [Station Keeping System](#); [Turret Mooring](#)

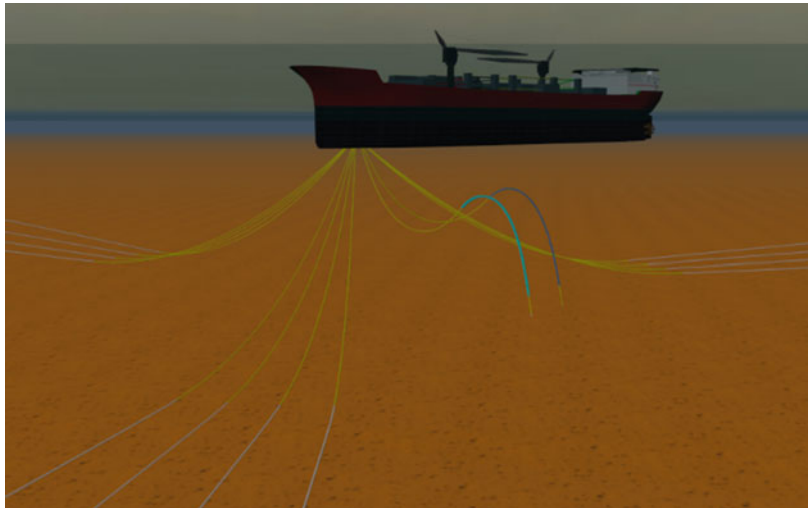
Definition

The passive mooring is the most commonly adapted station keeping method for offshore floating facilities. An offshore mooring system usually consists of one or multiple mooring lines that connect the floater to the seabed to retain floater's position and limit offset to enable designated operations in various environments and water depths. According to mooring system's control of floater heading, the mooring system can be divided into two main categories: **spread mooring system** (Fig. 1) and **Single-point mooring system** (Fig. 2). The spread mooring system utilizes spread mooring lines connected to the multiple locations of the floater to restraint its position as well as heading. The single-point mooring system has all mooring lines connecting to the point of rotation of the floater such that it only restrains the floater position while allowing floater weather

Single-Point Mooring,
Fig. 1 Spread mooring
 system (Courtesy of SBM
 Offshore)



Single-Point Mooring,
Fig. 2 Typical FPSO
 single-point mooring
 system (Courtesy of
 COTEC)



vanning. For both types of mooring systems, the mooring line is usually made up of chain, wire rope, polyester, anchor, and connectors. The mooring lines are pretensioned to form catenary or taut leg profile and spread out in all directions.

Historical Development

The mooring system can trace its origin to navigation. As ships sail around the world, the

anchor mooring hold vessel's position relative to a point on the seabed of a waterway without connecting the vessel to shore. As such, mooring lines have been used for station-keeping of floating vessels in various situations. For example, they are used to berth boats or ships at quaysides and often referred to as quayside moorings. In association with the development of offshore oil and gas exploration, the spread mooring system was further developed for station keeping of various offshore platforms for

operations of drilling, production, pipelaying, workover, etc.

When the ship shaped Floating Production Storage and Offloading system (FPSO) was first introduced, due to the challenges of spread mooring for ship-shaped FPSOs with excessive lateral load, the single-point mooring system was then developed. At present, most of the existing FPSOs are positioned with single-point mooring systems, and the spread moored FPSOs only exist in benign environment areas such as the West of Africa. A single-point mooring system allows the FPSO to freely rotate around a single point under the weather vane effect, thereby effectively reducing the forces of wind, waves, and current, so that the dimensions of the mooring systems are correspondingly reduced.

The world's first single-point mooring system was born in 1959 and was owned by the Royal Swedish Navy. The application of the single-point mooring system to the FPSO began in the 1970s. In fact, the vigorous development of FPSO and its mooring system were within the proceeding decade. At present, around 150 FPSOs are deployed in various sea areas of the North Sea, West Africa, South America, and Asia, most of which use single-point mooring systems (Nutter, 2014). The development of the global FPSO mooring system has generally gone through five phases, namely (Veldman, 1997; Ma et al., 2019):

- From 1976 to 1985, for the early stages of development, it became possible to deploy FPSOs in all areas for offshore oil and gas production through the single-point mooring system.
- From 1986 to 1994, during the FPSO growth period, the FPSO mooring technology was rapidly developed at this stage, associated with the increase of FPSO units by an average of two or more per year.
- From 1995 to 1998, during the expansion period of FPSOs, the number of FPSOs in this phase increased significantly, increasing at an average rate of more than 8 units per year.
- From 1999 to the present, the number of FPSOs has increased rapidly, and technological breakthroughs have been achieved. The

operating water depth increased from shallow water to 3000 m. The current record of FPSO operating water depth is Shell's Stone FPSO in the US Gulf of Mexico in a water depth of 2920 m.

- Especially after 2002, the FPSO concept has been extended to other types of operations such as in the form of LPGFPSO, LNGFPSO, Floating Storage and Regasification Unit (FSRU), Floating Production Drilling Storage and Offloading system (FPDSO).

The single-point mooring system is technically challenging and expensive to build. At present, companies that have the state of the art single-point mooring system technology are SBM, Bluewater in the Netherlands, SOFEC in the United States, and NOV (formally APL) in Norway and the United States.

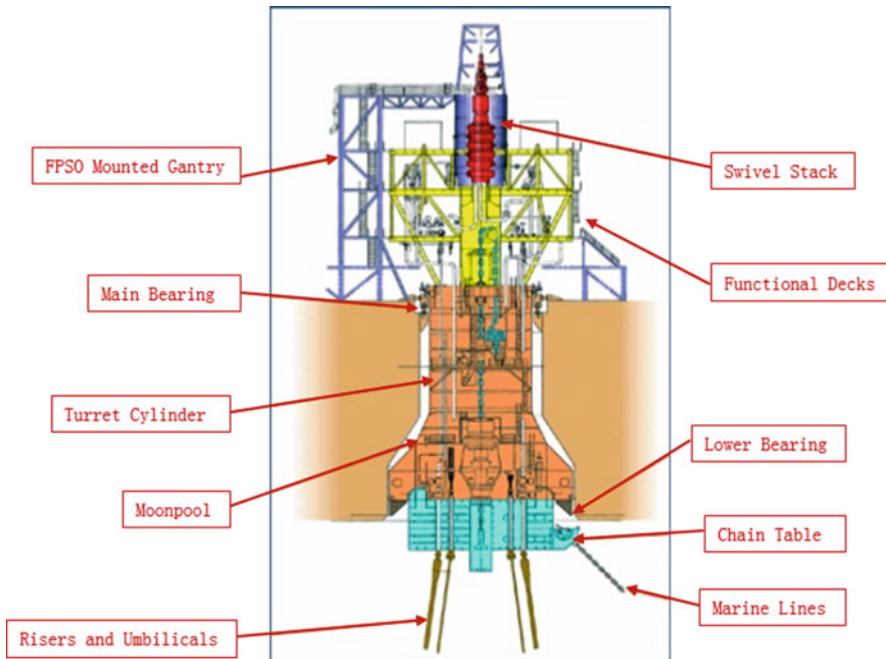
Principles of Single-Point Mooring

Working Principle

The magnitude and directionality of the prevailing environment of the wind, waves, and current are the driving parameters of mooring system configuration. They dictate the selection of mooring system type, i.e. the spread mooring or the single-point mooring. FPSOs have large size often with an overall length of longer than 300 m. The traditional spread mooring system fails to perform because of excessively large lateral force in harsh environment areas. For FPSOs in harsh environments, the single-point moorings have to be selected.

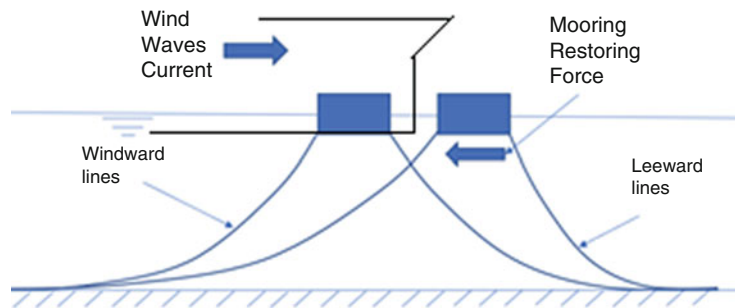
The beauty of the single-point mooring system allows the FPSO to freely rotate around a center of rotation such that the FPSO heads towards the prevailing environment. To achieve the feature of free weather vanning, the FPSO is fitted with a facility called turret at the forward part of the hull. The turret is geo-stationary connected by one or multiple mooring lines to the seabed, and the FPSO is connected to the turret via bearings which allows the free rotation (Fig. 3).

The FPSO single-point mooring system together with the turret has three functions:



Single-Point Mooring, Fig. 3 FPSO Single-Point Mooring Schematics (Courtesy of SOFEC)

Single-Point Mooring,
Fig. 4 Working principle
of spread mooring system



1. Stationkeeping – the FPSO is kept in position at the offshore oilfield operation site by the mooring restoring force generated by the single-point mooring.
2. Transfer of liquid and power – through the special liquid swivel, electric slip ring, etc., the produced liquid, liquid injection to wells, electrical power supply, and communication signals can be continuously transmitted from seabed to FPSO or FPSO to wells while the FPSO weather vanes around the single-point mooring.
3. Disconnectability – for specially designed single-point mooring, the mooring system can be disconnected in extreme condition for the safety of FPSO and onboard personnel.

Mooring Design

Like the spread mooring system, under the environmental loading of wind, waves, and current, the FPSO offsets from its mean equilibrium position in the horizontal plane. As the floater offset increases, larger tensions are being generated among the windward lines and tensions on leeward lines reduce (see Fig. 4 for illustration). The imbalance of mooring forces generates a so-called mooring restoring force that counterbalances the environmental forces. The larger the offset, the bigger the mooring restoring force until the acting environment force is balanced by the mooring restoring force. In this way, the offset of the floater is

also kept within the design limit (Barltrop, 1998; Luo, 2013).

The magnitude of the environmental loads also dictates the size of the mooring system components which are designed to withstand the environmental loading complying to applicable design codes.

Mooring Components

Like the spread mooring system, the mooring line is typically made up of synthetic fiber rope, wire and chain or a combination of the three. In some cases, buoyancy modules as well as clump weights can be deployed along the mooring line to achieve specific objectives such as seabed clearance, more desired mooring performance, etc. Environmental factors – wind, waves, and currents – drive the selection of mooring line materials (Figs. 5, 6, 7, 8, and 9).

Offloading

The single-point mooring system is beneficial to the crude offloading. The shuttle tanker can be connected with the FPSO either in tandem or side by side with the tandem mode being the most popular. The shuttle tanker and the FPSO can rotate freely around the single-point mooring, and the offloading operation is convenient, safe, and reliable (Fig. 10).

Key Applications

In a broad term, the single-point mooring system consists of the turret and mooring lines. The turret is typically a cylindrical steel structure that

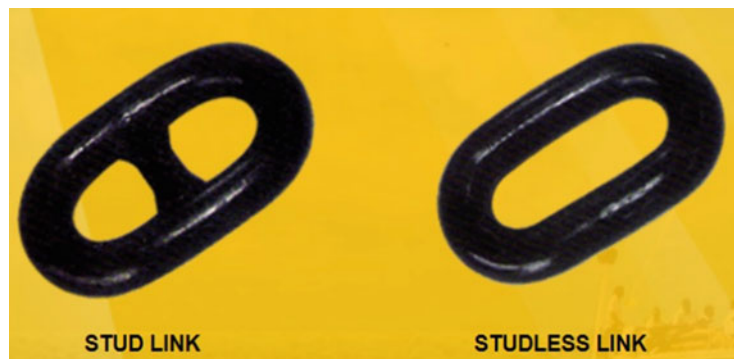
connects to the seabed via moorings and connects to the FPSO via rotation free bearings. The FPSO rotates around the inner bearings of the turret in countering the wind, waves, and current action. The mooring lines connect to the chain table which is at the bottom of the turret structure (Fig. 11) (Wichers, 2013).

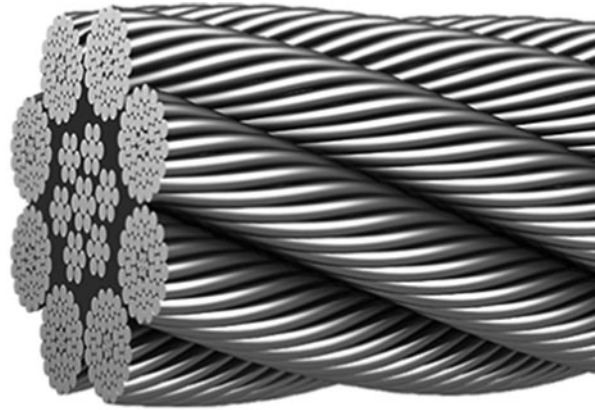
There are different types of single-point mooring systems. The exact structural design of the turret system varies according to project specific requirements, hull size, environmental conditions, etc. According to different working characteristics and the location of the turret, the turret system can usually be divided into two main types: (i) internal turret system and (ii) external turret system.



Single-Point Mooring, Fig. 6 Spiral strand wire rope (Courtesy of SIRTEF)

Single-Point Mooring, Fig. 5 Studlink and studless chain (Courtesy of AsAc)



Single-Point Mooring,**Fig. 7** Multistrand wire rope (Courtesy of Bridon)**Single-Point Mooring,****Fig. 8** Polyester rope (Courtesy Lankhost)**Internal Turret System**

Among the mooring systems for FPSOs, the internal turret system is most commonly used in the harsh environments with large number of risers and umbilicals (Fig. 12). The internal turret is generally located near the bow of the FPSO, and the turret sits inside a moonpool of the FPSO hull. The mooring lines is connected at the bottom of the turret via chain table, and the flexible risers from the seabed are connected with the rigid pipes inside the turret, or directly connected upwards to the upper part of the turret.

The main features of the internal turret system include:

1. The turret diameter can be larger than that of an external turret system, providing plenty of space for a large number of risers.

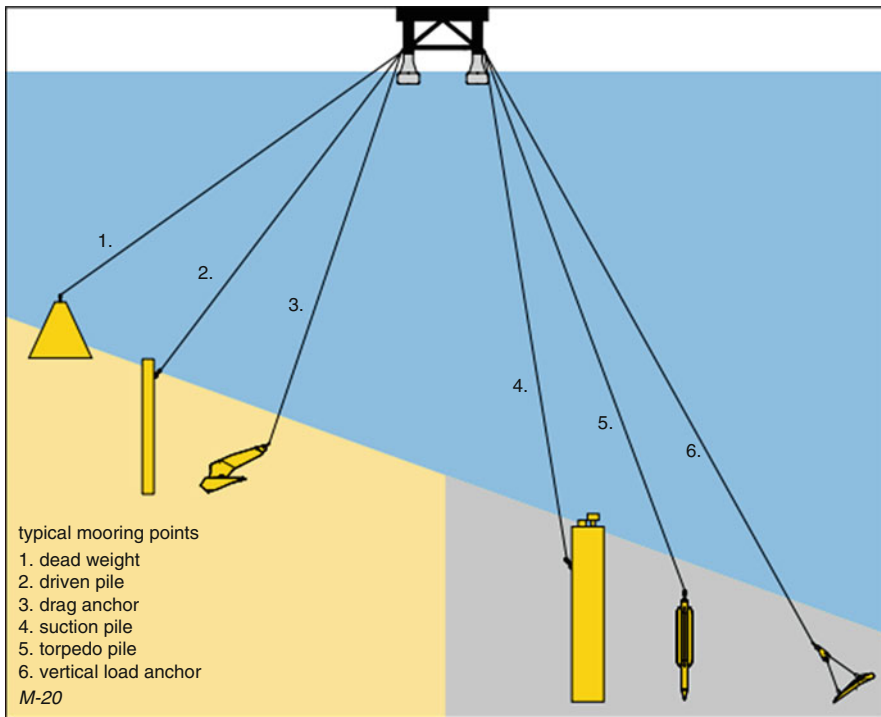
2. The inner turret is well protected by the hull.
3. The presence of the turret reduces the tank capacity and has an impact on the hull structure.

External Turret System

The external turret system (Fig. 13) is similar to the inner turret system except that the turret is located outside the FPSO hull. The external turret system is frequently used in shallow waters say below 50 m to increase the effective water depth because the external turret is positioned well above the waterline.

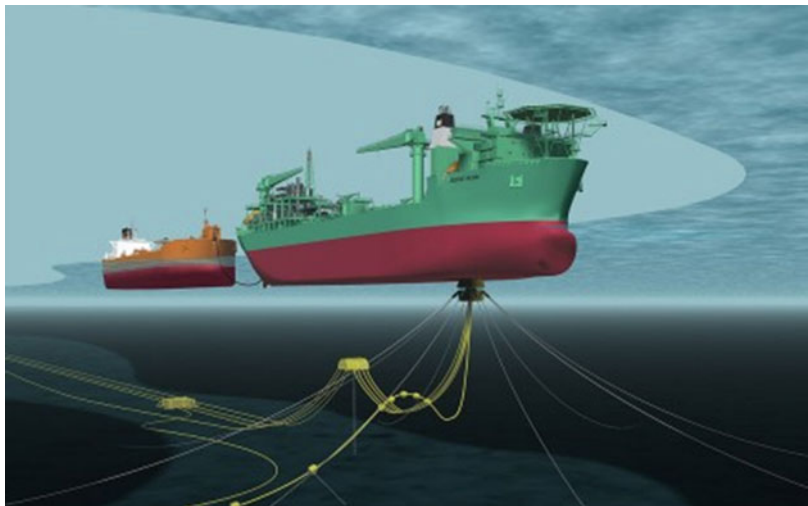
The main features of an external turret system include:

1. It allows stand-alone fabrication and installation along the quayside, while the internal turret system can only be installed in the dry dock.



Single-Point Mooring, Fig. 9 Various types of anchors (Courtesy of Vryhof Anchor)

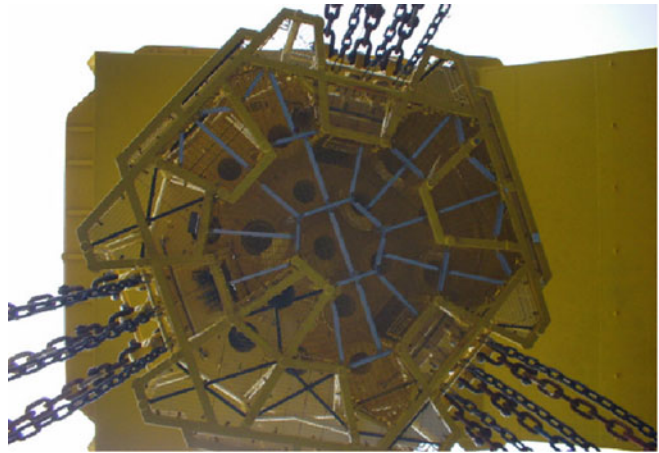
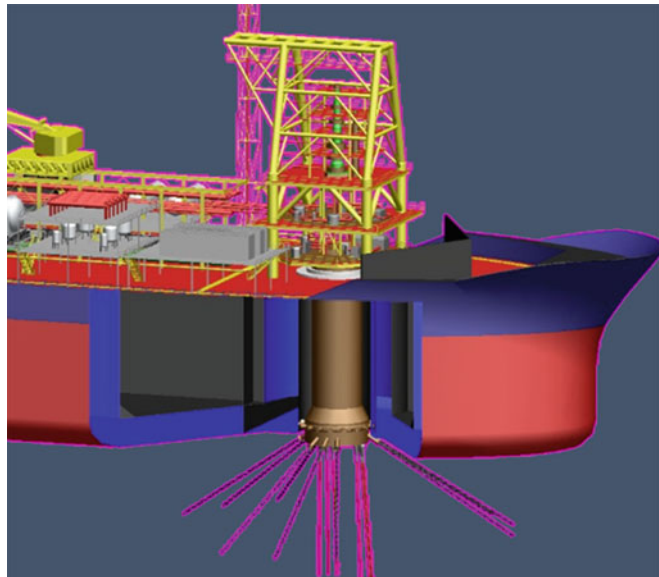
Single-Point Mooring, Fig. 10 FPSO in Offloading Operation (Courtesy of Bluewater)



2. It does not affect the storage capacity of the hull.
3. The mooring and riser connections are above water that eases the effort of inspection and maintenance.

Other Types of Single-Point Mooring Systems

To suit various types of operating environment and functionality requirements, other types of

Single-Point Mooring,**Fig. 11** Turret chain table
(Courtesy of SBM)**Single-Point Mooring,****Fig. 12** Internal turret mooring system in an FPSO
(Courtesy of SOFEC)

single-point mooring systems have been developed.

Tower Yoke Mooring System

The tower yoke mooring system uses a rigid jacket structure piled to the seabed as the anchor point and the FPSO is connected to the jacket via jointed steel members. The system was developed for shallow to ultra-shallow water (below 50 m) application where the conventional catenary mooring system fails to operate due to shallow water depth (Fig. 14).

Catenary Anchor Leg Mooring System (CALM)

With the growing crude oil trade, petroleum products are transported around the world. For coast area, the application of catenary anchor leg mooring system (CALM, Fig. 15) is preferred. CALM systems are used for temporary mooring of crude shuttle tankers for the purpose of loading and unloading of crude products.

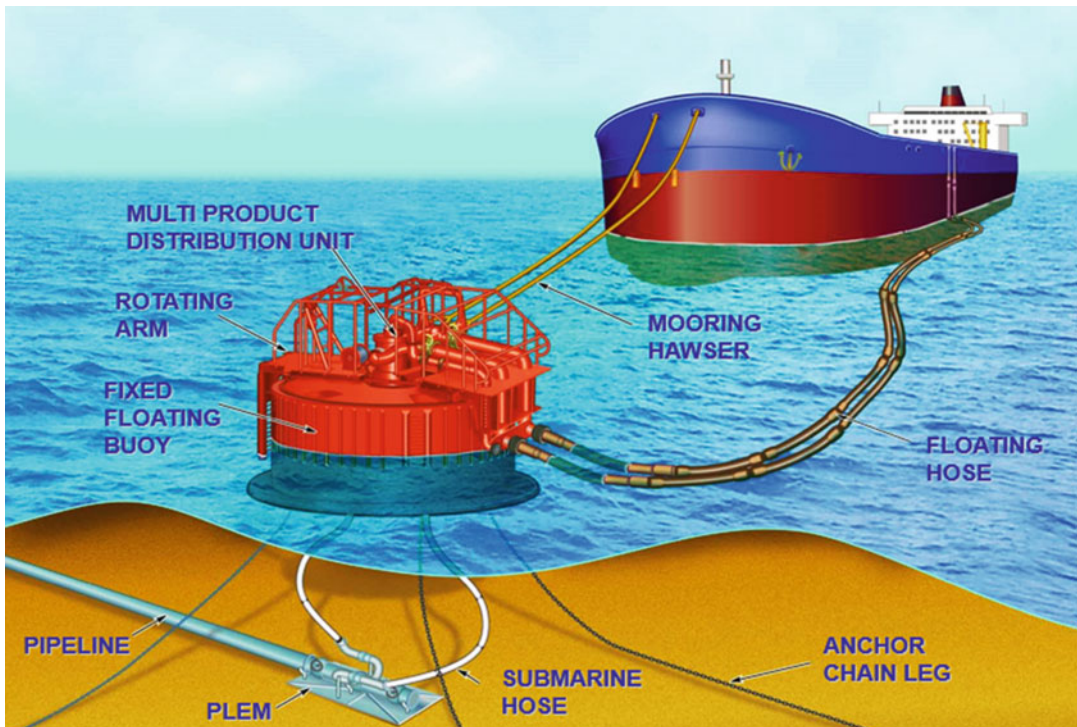
The CALM system consists of a floating buoy moored by typically six spread mooring lines, and the turntable which is mounted on top of the buoy



Single-Point Mooring, Fig. 13 External turret system



Single-Point Mooring, Fig. 14 Tower yoke mooring system (Courtesy of CNOOC)



Single-Point Mooring, Fig. 15 Catenary anchor leg mooring system (Courtesy of SBM)

for free rotation around the bearings. Like an FPSO, the shuttle tanker can weather vane to interact with the prevailing wind, waves, and current.

Cross-References

- [Catenary Mooring](#)
- [Single Point Mooring](#)
- [Taut Mooring](#)
- [Turret Mooring](#)

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Single-Point Mooring System

- [Catenary Mooring](#)
- [Internal Turret Single-Point Mooring \(SPM\) System](#)

Single-Point moorings

- [Spread Mooring System](#)

Size Effect and High-Pressure Zone of Sea Ice

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Introduction

In the physical mechanical test of sea ice, it is found that the strength of sea ice is not only related to the temperature, salinity, and other generating environment, but also closely related to the size of sea ice. By establishing the relationship between sea ice strength and size, the size effect of sea ice has been studied.

In the early research on ice load of offshore platform structure and ship structure in ice area, it is found that sea ice not only affects the overall stability of the structure, but also causes local damage to the structure. The problem of determining these “local pressures” is a long-standing one which has faced the designers of ships and offshore structures throughout the history of operations in ice covered waters. Local loads affect areas from 1 m² up to as much as 100 m² and determine the design of steel plate thicknesses plus spacing and size of bracing or concrete wall thicknesses and the amount of reinforcing (Masterson and Frederking 1993).

Definition

The *size effect* refers to the effect of material size on its mechanical properties. Size effect exists in many materials such as rock, concrete, ceramics, metal. Sea ice is a typical brittle material composed of crystal, which also has obvious size effect. The effect of sea ice size on its strength can be divided into two aspects, including the size

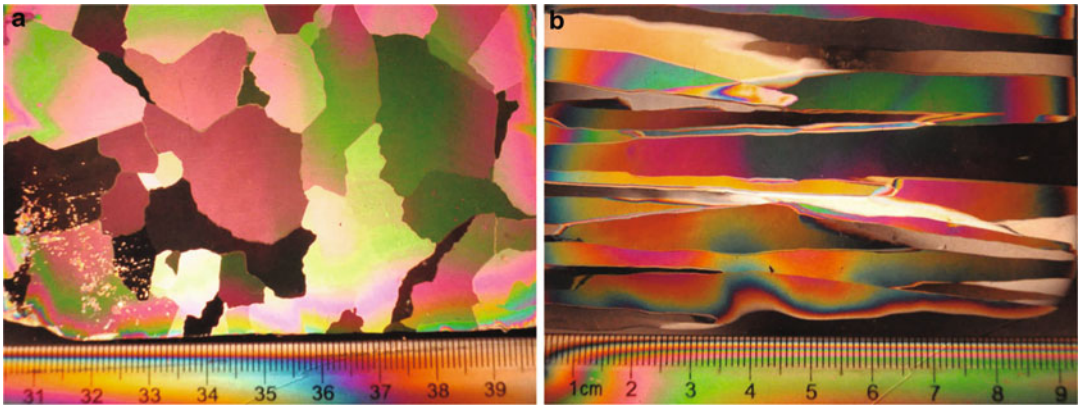
effect of ice specimens and the size effect of ice grain.

The *high-pressure zone* is the area of stress concentration caused by the nonsimultaneous destruction of the sea ice acting on the structure. A large number of measurement data and model tests show that the maximum local ice pressure of sea ice is much higher than the global ice pressure of the structure. Therefore, the high-pressure zones during the interaction of sea ice and structure may cause local deformation and damage of the structure, which will seriously threaten the stability of the whole structure.

Scientific Fundamentals

Ice Crystal Types

The crystal type and size of both sea ice and fresh water ice also have a great influence on the strength. Ice with different crystal types has different physical and mechanical properties. There are 13 crystal types of naturally formed ice. Previous studies have pointed out that ice is composed of hexagonal crystal, but due to the different growth environment and meteorological conditions, the texture characteristics of ice interior and the orientation of single grain in polycrystal are different. Therefore, ice with different crystal structure is formed. The crystal structure characteristics of sea ice mainly include grain shape, grain size, and its spatial distribution. The ice type can be defined according to the distribution characteristics of ice crystals, among which granular ice and columnar ice are most common shown in Fig. 1. The sea ice sheet has a two-layer structure: a granular ice layer on top and a columnar ice layer below. The granular ice is composed of particle grains with similar size, whose size is usually less than 10⁻² m. It is randomly arranged in space and has the characteristics of isotropic material. For columnar ice, the crystals are slender in the vertical direction (i.e., the growth direction of ice), while in the horizontal plane, the grains are closely arranged together. Its particle size ranges from 10⁻² m to 10⁻¹ m, and it has the characteristics of anisotropic materials. Hence, the columnar ice has significant anisotropy because of its



Size Effect and High-Pressure Zone of Sea Ice, Fig. 1 Different shape of ice crystal: (a) Granular ice and, (b) Columnar ice

columnar structure (i.e., the grain size in the vertical direction is larger than that in the horizontal direction). It mainly shows that the loading direction has an effect on its mechanical strength, while the grain size has no significant effect on its strength. Because of the diversity of crystal structure and grain size in ice, the influence of grain size should be considered according to different crystal types when determining the compressive strength of ice.

Global and Local Pressures

The ice pressure of structure can be divided into global ice pressure and local ice pressure. Figure 2 shows the definitions used for global and local pressures by the example of a cylindrical structure (Timco and Sudom 2013).

Global pressure is related to the global (or nominal) area, which is defined as the contact area that develops during the interaction. This is simply the area of the structure projected onto the dimensions of the ice feature at the appropriate amount of penetration. The size of this area could be small (e.g., laboratory tests), intermediate, or large (e.g., interaction of an ice sheet with an offshore structure). Global pressures are generally thought of with respect to overall structure design and stability. The global interaction area corresponds to the full area over which the ice exerts a force on a structure at a given moment in time during the process of a pressure-area interaction. The global ice pressure is the ratio of the ice load of

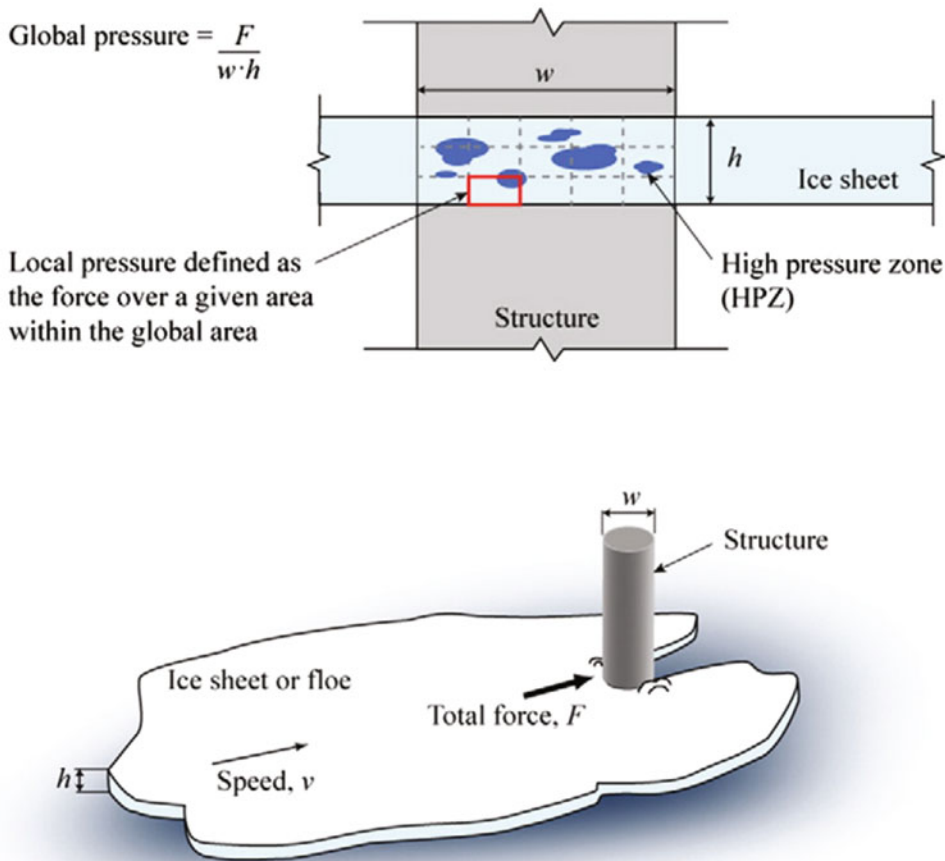
structure to the area of the whole contact surface, which represents the stress state of the whole structure. The area of contact surface can be equivalent to the product of structure width and ice thickness.

Local pressure occurs over a smaller, defined portion of a larger global area. For example, a local pressure could be captured on one instrumented pressure panel on the side of a large offshore structure. In this case, the pressure is generally calculated using the maximum ice load on the local measurement area during an event and the assumed corresponding local contact area. The local pressure is related to a part of the area of a structure to be designed; for example, a panel or the plating between frames. Because the area is inside a larger area of ice, its confinement conditions can be quite different than that for the global pressure. The high-pressure zones are the area of stress concentration caused by the non-simultaneous destruction of the sea ice acting on the structure.

Historical Development

Specimen Size Effect on the Strength

Generally, in the mechanical property test of materials, the size effect of material specimen can be divided into two kinds: the ratio effect between height and width and volume effect. Because of the transverse friction between the end face of the specimen and the indenter during the loading process, the ratio of the height and width of the



Size Effect and High-Pressure Zone of Sea Ice, Fig. 2 Grain size effect on the compressive strength of ice (Cole 1985)

specimen may affect the test strength. The other is the volume effect proposed for brittle failure. Under the condition of similar specimen composition, the larger its volume is, the more defects it contains and the greater probability of brittle failure is, so the test strength becomes lower.

In the compression test of sea ice, it is also found that the compression strength decreases with the increase of specimen volume. Sea ice is a compound material composed of pure ice crystal, brine, bubble, and other impurities. Hence, there are many defects in sea ice, which lead to that the composition of sea ice is more complex and the strength is lower than that of fresh water ice. When the internal defects of sea ice reach a certain degree, it is easy to form large salt discharge channels, and finally form cracks with different sizes in

different positions of the specimen. When the size of ice specimen is larger, there will be more defects inside the ice and the strength of ice specimen in the test will decrease. Kuehn et al. (1993) found that specimen size only has an influence on ice brittle failures and does not affect the ductile compressive peak strength according to the test results on the compressive strength of fresh water ice. In addition, the size effect of sea ice is also related to the material properties of compression test platens. When the strength is measured using superior platens, the size does not affect the brittle compressive strength. Specimen size does affect the brittle compressive failure strength when measured by using inferior platens, owing most probably to the effects of nonsimultaneity of failure in large specimens and of end-confinement in small specimens.

In this case, the strength decreases as the specimen size increases.

The tensile strength of ice decreases with increasing test specimen volume as shown in Fig. 3. Volume effects on the strength of brittle materials are usually described by a Weibull statistical distribution approach (Weibull 1951). The Weibull theory is often referred to as a “weakest-link” theory of fracture. In the Weibull theory, the probability of fracture is given by:

$$P = 1 - \exp(-v(\sigma/\sigma_0)^m) \quad (1)$$

where P is probability of fracture, σ is the applied tensile stress (which is assumed to be uniform over the stressed volume of the material), σ_0 is a constant, v is stressed volume, and m is the Weibull modulus. This expression leads to the following expression for the volume dependence of the strength of brittle materials:

$$\sigma_2/\sigma_1 = (v_1/v_2)^{1/m} \quad (2)$$

where σ_1 and σ_2 are the compressive failure strengths of specimens; v_1 and v_2 are the volume of specimens, respectively.

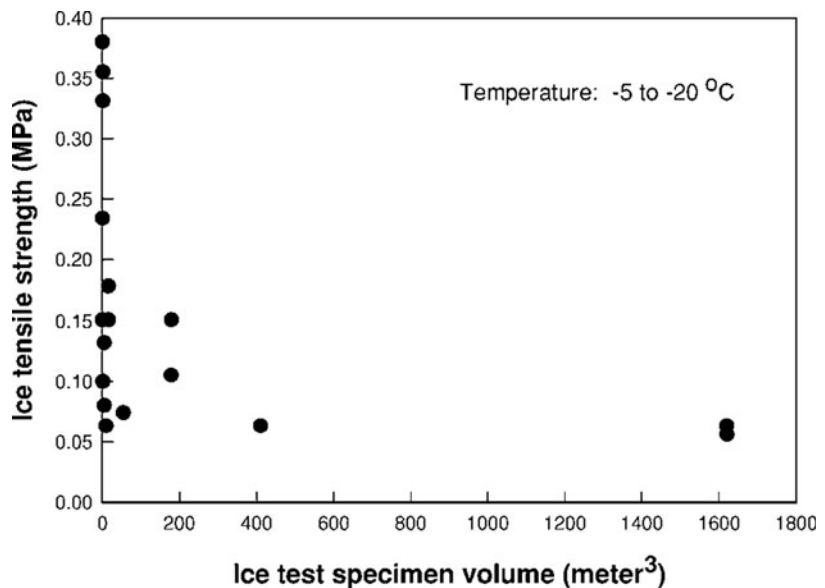
From the strength-volume data in Fig. 3, it was possible to obtain a Weibull modulus value for ice. The Weibull modulus of ice is estimated to

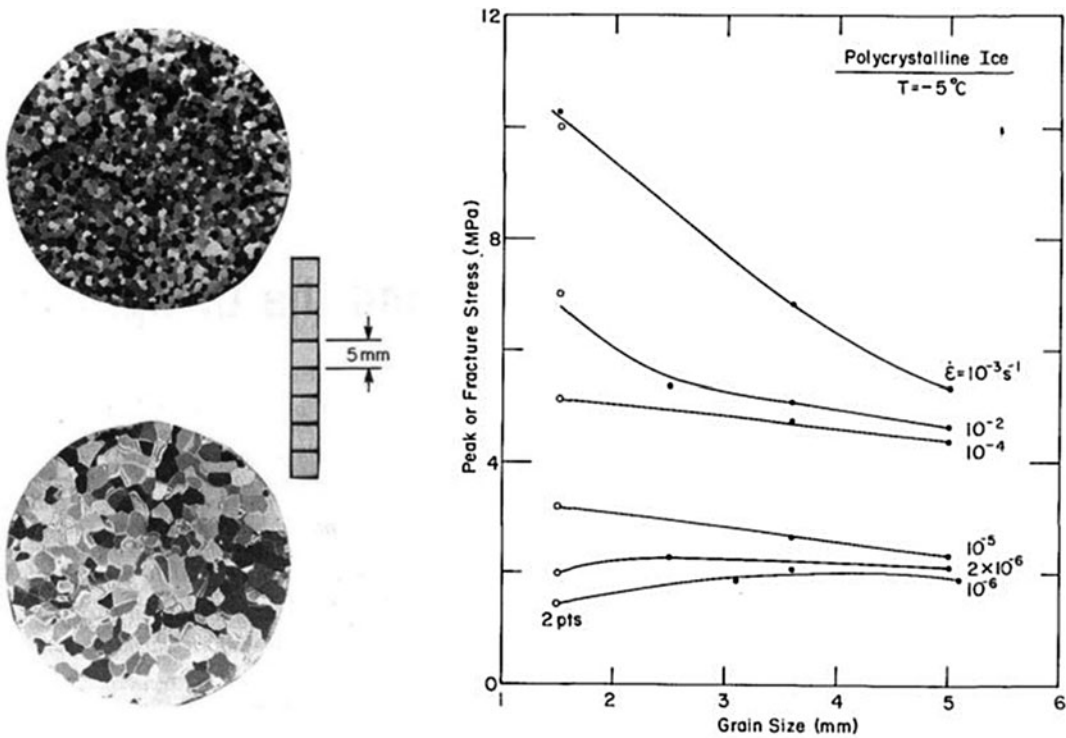
have a value of approximately 5. To our knowledge, this is the first time a Weibull modulus for ice has ever been put forward. By way of comparison, the Weibull modulus of brittle ceramic materials is typically in the range of 5 ~ 20. The higher the value of the Weibull modulus, the lower is the statistical scatter in the measured fracture stress.

Grain Size Effect on the Strength

Many scholars have studied the influence of grain size on ice strength through experiments, mainly for the granular ice. Cole (1985) conducted experiments on fresh water ice to analyze the effect of grain size on the internal crushing characteristics of polycrystalline ice. Meanwhile, the law of the compressive strength of ice changing with the grain size under different strain rates is summarized, as shown in Fig. 4. Schulson (1990) took the granular ice samples with different grain size as the research object and found that the brittle strength of ice is directly proportional to the square root of grain diameter, but it is only suitable for the brittle failure of ice. Found that the compressive strength of fresh water ice increased with the decrease of particle size. Jones and Chew (1983) found that when the size of the specimen must be more than 12 times of the grain size, the grain size has no effect on the strength; and when the grain size is between 0.6 and 2.0 mm, the

Size Effect and High-Pressure Zone of Sea Ice,
Fig. 3 Illustration of the global and local pressures (Timco and Sudom 2013)





Size Effect and High-Pressure Zone of Sea Ice, Fig. 4 Tensile strength of ice as a function of volume (Petrovic 2003)

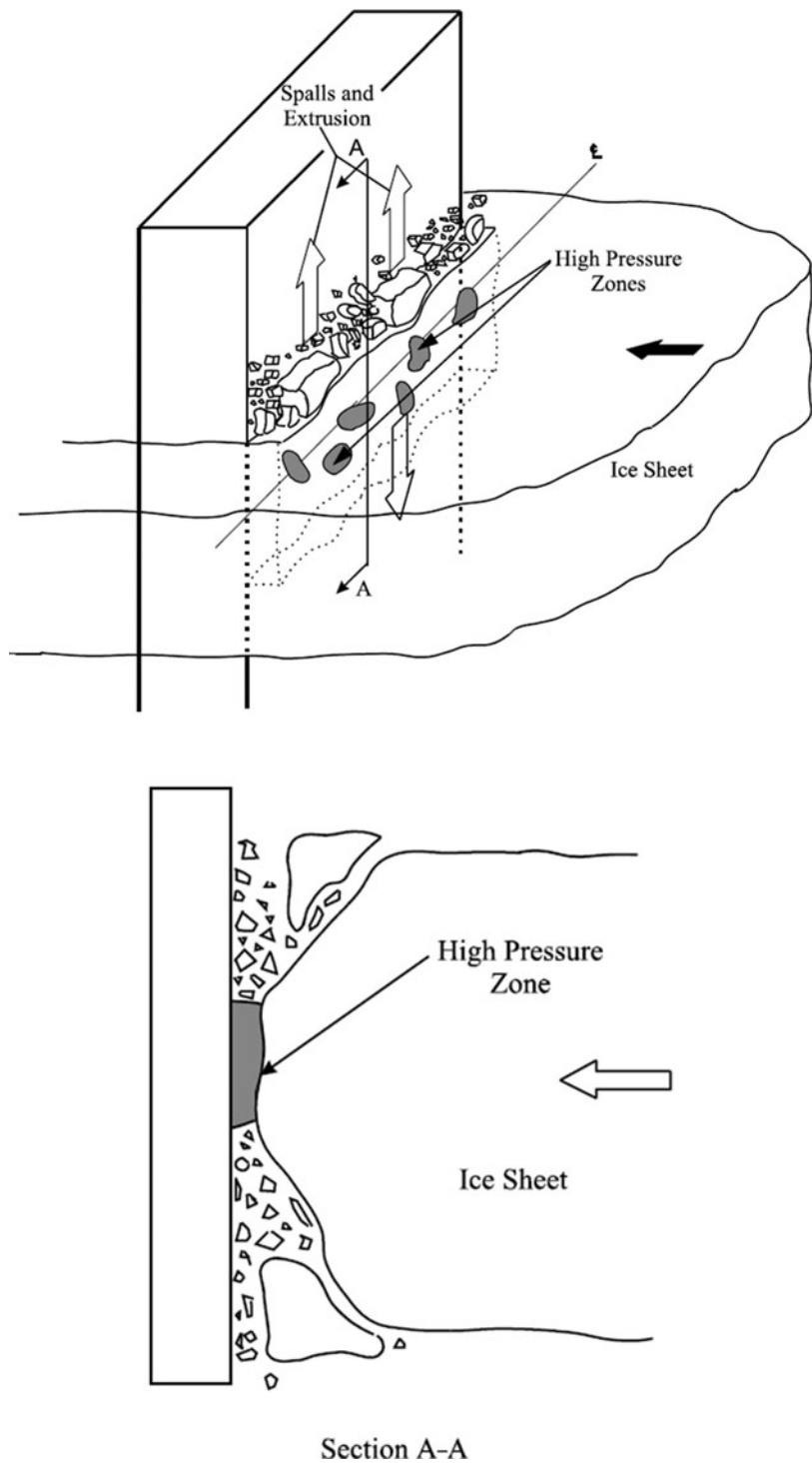
compressive strength is not significantly affected by the grain size. When the specimen size is the same, the ice specimen with small particle size has larger uniaxial compressive strength, which can be explained by dislocation theory (Schulson 1999). For the same material, the smaller the grain diameter is, the more the grain boundary will be, which will hinder the dislocation movement. Therefore, materials with smaller grains need higher deformation resistance, and macroscopically, the materials have higher strength. On the other hand, a possible process is the propagation of microcracks that are nucleated by dislocation pile-ups against grain boundaries.

Compared with granular ice, the columnar ice has significant anisotropy because of its columnar structure (i.e., the grain size in the vertical direction is larger than that in the horizontal direction). It mainly shows that the loading direction has an effect on its mechanical strength, while the grain size has no significant effect on its strength.

Generation and Distribution of High-Pressure Zones

As we all know, the ice load estimation is required in the design of ships and offshore structures for arctic and subarctic conditions. With the development of research on the structural strength of ice regions, the ice pressure is also important to improve the appropriate levels of structural strengthening. When the overall stiffness of the structure meets the design requirements of ice load, the local damage of the structure may still occur. For example, in the process of ship structure navigation, the collision of sea ice can cause local deformation or even damage to the hull. Therefore, people pay more attention to the generation, size, and distribution of ice pressure.

A lot of researches have been made on the physical process of sea ice crushing by means of field tests and observations. The causes and distribution of structural high-pressure zones are analyzed considering the influence of sea ice failure mode, as shown in Fig. 5 (Jordaan 2001). When the sea ice interacts with the vertical structure, the

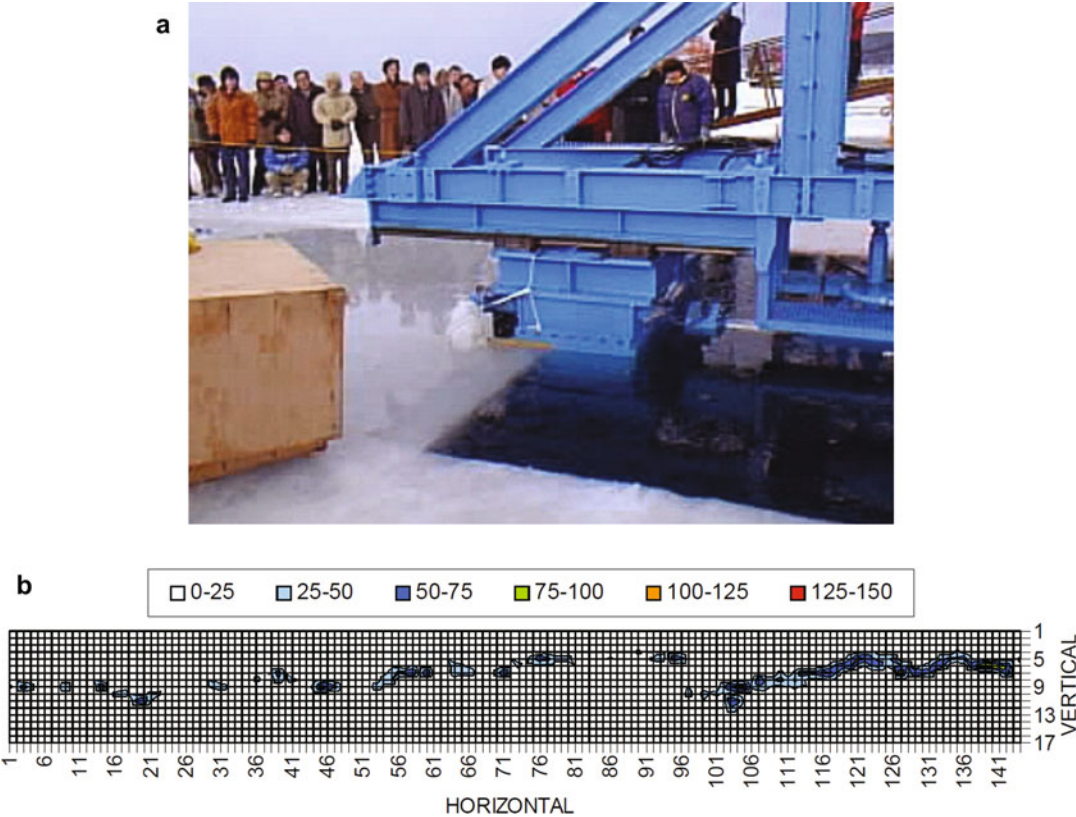


Size Effect and High-Pressure Zone of Sea Ice, Fig. 5 Schematic illustration of the main processes of spalling, extrusion, and high-pressure zone formation (Jordaan 2001)

upper and lower surface of the sea ice near the contact surface first crush. The broken sea ice is pushed out of the contact surface by the subsequent ice sheet. At the same time, the ice in the middle layer of the ice sheet has not yet broken, which forms a bulge and keeps interaction with the structure. Because of the continuous contact between the middle layer of ice sheet and the structure, the structure can form stable high-pressure zones. With the increase of ice load, the sea ice in the high-pressure zone will also be broken. The microcracks inside the sea ice gradually increase and continue to expand, and some larger ice blocks on the upper and lower surface will peel off again, and finally the sea ice will form a new bulge in the middle of the ice sheet. Therefore, the sea ice can cause continuous ice load on the structure, and the high-pressure zone of the structure also exists all the time.

According to the failure process of sea ice, the high-pressure zone mostly occurs near the center line of the contact surface and has a linear distribution, which is called “line-like load” (Jordaan et al. 2010). From the formation of high-pressure zones, the distribution of high-pressure zones is closely related to the failure mode of sea ice. Frederking (2004) analyzed an event for which an indenter of dimensions 1.5 m wide and 0.5 m high penetrated a sea ice sheet of 168 mm thickness at a rate of 3 mm/s, for a total penetration of 1000 mm. The entire indenter face was covered with tactile sensor elements which allowed direct observation of the pressure distribution during the indentation process. Figure 6 clearly shows that the pressure is certainly not uniform across the loaded indenter face. There is a “line-like” load along the center line of the indenter and two other regions of high-pressure zones along the top and bottom of the indenter

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Size Effect and High-Pressure Zone of Sea Ice, Fig. 6 Distribution of high-pressure zones with crushing failure of sea ice: (a) the JOIA project in Japan; (b) view of

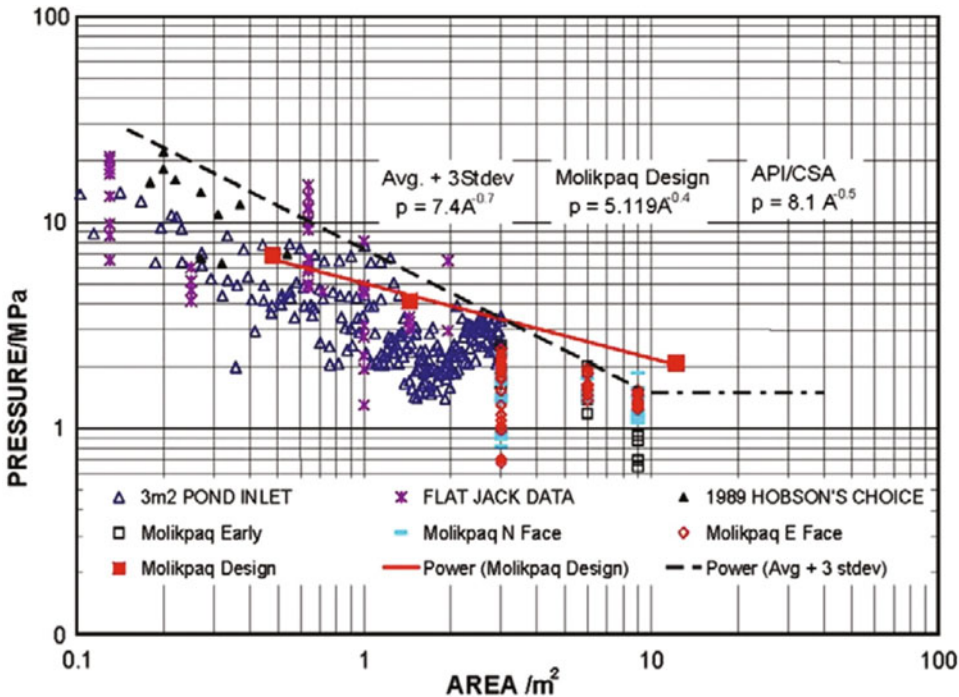
the pressure distribution on the face of the indenter in the JOIA tests analyzed by Frederking (2004)

face. This pressure distribution continually changed during the loading process but often resembled that shown in. Moreover, when the sea ice occurs the brittle crushing failure, the line-like distribution characteristics of the high-pressure zones on the vertical structure are hardly affected by the structural size and ice thickness, that is, it is independent of the size of the contact surface between the sea ice and the structure (Erceg et al. 2015).

Influence Factors of High-Pressure Zones

There is a strong perception in the ice mechanics community that during ice-structure interaction, the ice pressure always decreases as the area of contact increases. This understanding is often based on the pressure-area plot published by Sanderson (1988), which combines a large number of data sources and ice interaction situations on a single plot and shows a definite decrease in pressure with increasing area. The exponential fitting of the relationship between the local ice pressure and the contact area is widely recognized. Masterson and Frederking (1993) compiled a large

number of pressure-area points based on small- and medium-scale experiments, icebreaker impacts with ice, and larger-scale ice interactions with structures. In their analysis, local pressures were combined with global pressures measured over a smaller area. This combined data shows decreasing pressure with area with a functional form $p = 8.1A^{-0.572}$ for areas up to 19 m^2 , where p is the pressure (in MPa) and A is the area in m^2 . Above 19 m^2 , the pressure appears constant at $p = 1.5 \text{ MPa}$. Masterson et al. (Masterson et al. 2007) combined data from the MEDOF panels on the Molikpaq offshore platform, medium-scale field indenter tests and some flat jack tests to produce a combined pressure-area plot for local pressures. The data also showed a definite trend of decreasing pressure with increasing area, as shown in Fig. 7. Note that in contrast to the Masterson and Frederking (1993) plot, the Masterson et al. (Masterson et al. 2007) plot does not contain any data on ships in ice or global pressure-area data. The resulting upper bound pressure-area relationship for areas from 0.1 m^2 to 10 m^2 is given by $p = 7.4A^{-0.7}$. For areas greater



Size Effect and High-Pressure Zone of Sea Ice, Fig. 7 Local pressure-area relationship for fixed structures (Masterson et al. 2007)

than 10 m², the pressure is given as a constant 1.48 MPa. The ISO 19906 Arctic Structures Standard (ISO 2010) has adopted the Masterson et al. (Masterson et al. 2007) approach to provide guidance on measured pressures over specific local areas.

In addition, the law of ice pressure decreasing with the increase of contact area is not always unaffected. It will be changed by other loading environmental factors, such as loading rate, ratio of structural width to ice thickness, failure mode of sea ice and sea ice strength, etc. More importantly, the research shows that in some cases, these factors have more obvious influence on local ice pressure than the contact area (Timco and Sudom 2013). Sodhi (2001) pointed out the influence of loading rate on the failure mode and pressure distribution of sea ice. When the sea ice is loaded at low speed, it will have ductile deformation, and the ice pressure will increase with the increase of speed, and distribute on the whole contact surface; on the contrary, the sea will have continuous brittle failure under high-speed loading, and the ice pressure will decrease with the increase of ice speed, and concentrate on the center line of the contact surface. When the thickness of ice sheet increases, due to the change of sea ice failure mode, the distribution of local ice pressure is no longer concentrated in the middle line, and there may be multiple high-pressure zones (Jordaan 2001). Therefore, the value and distribution of structural ice pressure are closely related to the failure mode of sea ice. However, the research methods of high-pressure zone are mainly field or laboratory tests, so it is difficult to study the internal micro fracture characteristics of sea ice and the internal mechanism of high-pressure zones.

According to the research on the influencing factors of ice pressure, the Sanderson pressure-area plot is a mixture of vastly different data sources. As such, it is of interest in a general sense but is not meant to be applied for many specific situations. For specific applications such as structure or ship design, it is important to use a subset of only the appropriate data to define the most suitable pressure-area relationship. In doing so, other factors can be taken into account

including aspect ratio, loading rate, failure mode, and the physical and mechanical properties of the ice. In many cases, these factors are far more important than the area in defining the pressure (Timco and Sudom 2013).

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Sloping Structure

- Numerical Simulation Floe Ice–Sloping Structure Interactions
- Sloping Structure: Floe Ice Interactions
- Sloping Structure–Level Ice Interactions

Sloping Structure: Floe Ice Interactions

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Synonyms

Floe Ice; Sloping Structure

Definition

- Sloping Structure Offshore structure/ships with a sloping surface, which promotes the flexural failure of sea ice.
- Broken Ice Broken ice is loose ice, which consists of small floes that are broken up as a result of natural processes or active or passive intervention (ISO19906 2019).
- Floe Ice An ice field consists of ice floes of various sizes. The ice floes can be as large (in the order of kilometers) as being considered as level ice and as small (in the order of meters) as being considered as rubble ice (Lu 2014). This is a more general concept and includes

- Splitting Failure both the conventionally discussed “level ice” and “broken ice” conditions. It is a type of in-plane failure of an ice floe due the “wedge out” ice – structure contact force.
- Flexural Failure It is a type of out-of-plane failure of an ice floe featuring the radial cracking and circumferential cracking of the ice floe. The governing force is controlled by the initiation of circumferential cracks.
- Radial Cracking When an ice floe is under a vertical contact force, an out-of-plane deformation is taking place on the ice floe. Normally, radial cracks shall emanate from the loading area and propagate further away from the loading area. The radial cracks’ propagation distance is limited.
- Circumferential Crack Following the propagation of radial cracks, circumferential cracks shall develop. The initiation of circumferential crack determines the failure of an ice floe.

Scientific Fundamentals

Floe Ice

Ice floe’s size ranges from around 10 m to 10 km (Leppäranta 2011). Therefore, floe ice can be as large as “level ice” and can be as small as broken ice floes. Figure 1 shows a typical ice field in the Arctic and demonstrates the discrete features of the Arctic. The Arctic sea ice is by no means a continuous ice sheet as would often be assumed in the concept of “level ice.” Floe ice is a more general term that combines both the level ice and broken ice conditions (Lu 2014).
For sloping structure and level ice interactions, one can refer to the dedicated entry of this

encyclopedia. Here, we shall mainly present the broken ice features.

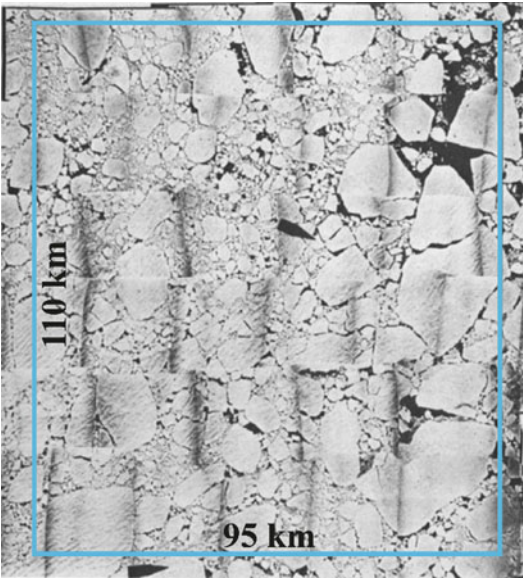
Broken Ice and Structure Interactions

Broken ice-structure interactions are not as well studied compared to those in the level ice condition. One reason is that apart from material properties, there are more parameters involved in describing a broken ice field, e.g., floe size,

geometry, ice concentration, ice thickness, and ice pressure. For level ice, apart from material properties, ice thickness is generally sufficient for ice loads calculations. Another reason is that a broken ice field typically leads to lower ice loads compared with those in a level ice condition. However, level ice is just a theoretical simplification; the actual Arctic generally consists of discontinuous features, such as ice ridges, leads, or ice floes of varying sizes, which form a broken ice field. A typical broken ice field in nature can be found in the Marginal Ice Zone (MIZ), which stretches approximately 100 km from the ice edge (Weeks 2010), where ice floes are primarily broken off by gravity waves (see Fig. 2).

Studying broken ice-structure interactions is of significant practical value. First, a great portion of a structure’s service time is within broken ice fields (e.g., see Fig. 3a). It is beneficial to gain an understanding about a structure’s most frequent loading condition.

Second, nowadays, the concept of floating Arctic offshore structures has become extremely attractive as opposed to traditional fixed Arctic offshore structures. This interest is driven by the desire to perform Arctic exploration and exploitation in deeper Arctic waters (i.e., deeper than 100 m (Hamilton 2011, Riska and Coche 2013)), and also the various flexibilities floating structures offer (e.g., disconnection in the face of extreme ice features). However, due to the limitation of

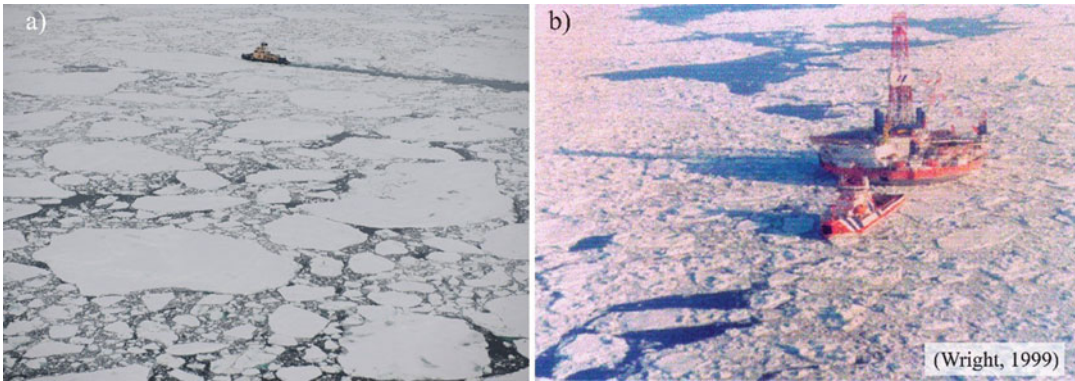


Sloping Structure: Floe Ice Interactions, Fig. 1 Mosaic of aerial photograph of summer pack by Hall in 1978 (Rothrock and Thorndike 1984)



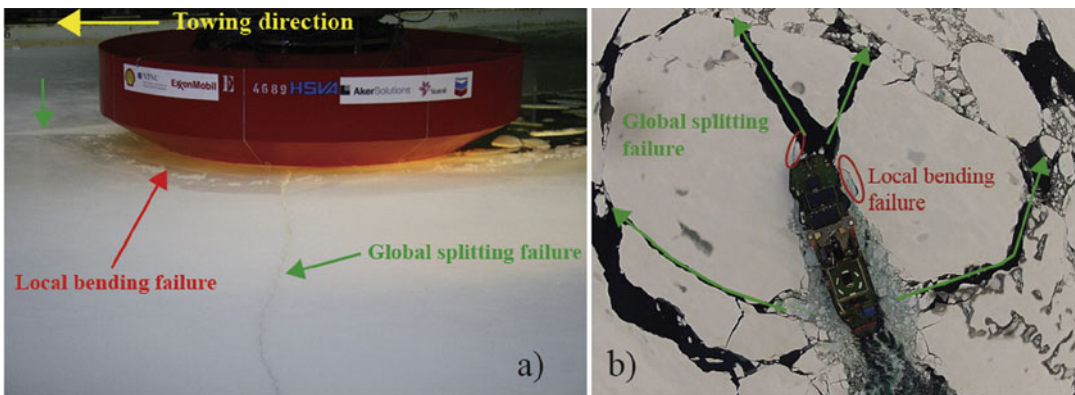
Sloping Structure: Floe Ice Interactions, Fig. 2 Marginal Ice Zone during the KV-Svalbard Research Cruise, March 2012 in the Northern Barents Sea: (a) a sharp transition between open water and MIZ

with short waves and small ice floes (less than 1 m); (b) MIZ with long waves and large ice floes (approximately 100 m across)



Sloping Structure: Floe Ice Interactions, Fig. 3 Examples of broken ice-structure interactions (a) Oden transiting in Fram Strait in August, 2012; (b) Ice

management operation generating a broken ice field for the protected vessel, i.e., Kulluk (after (Wright 1999))



Sloping Structure: Floe Ice Interactions, Fig. 4 Global splitting and local bending failure for (a) a conical structure in the model ice tank, HSVA, and (b) Oden in the Greenland Sea (September, 2012)

mooring lines and dynamic positioning (DP) systems, a floating system is typically supported with ice management operations (Palmer and Croasdale 2013), e.g., the floating structure, Kulluk, in the Canadian Beaufort Sea (see Fig. 3b). The goal of an ice management operation is to create a relatively “open” ice condition that consists of small ice floes with low concentration. Therefore, ice loads calculations within a broken ice field become increasingly important.

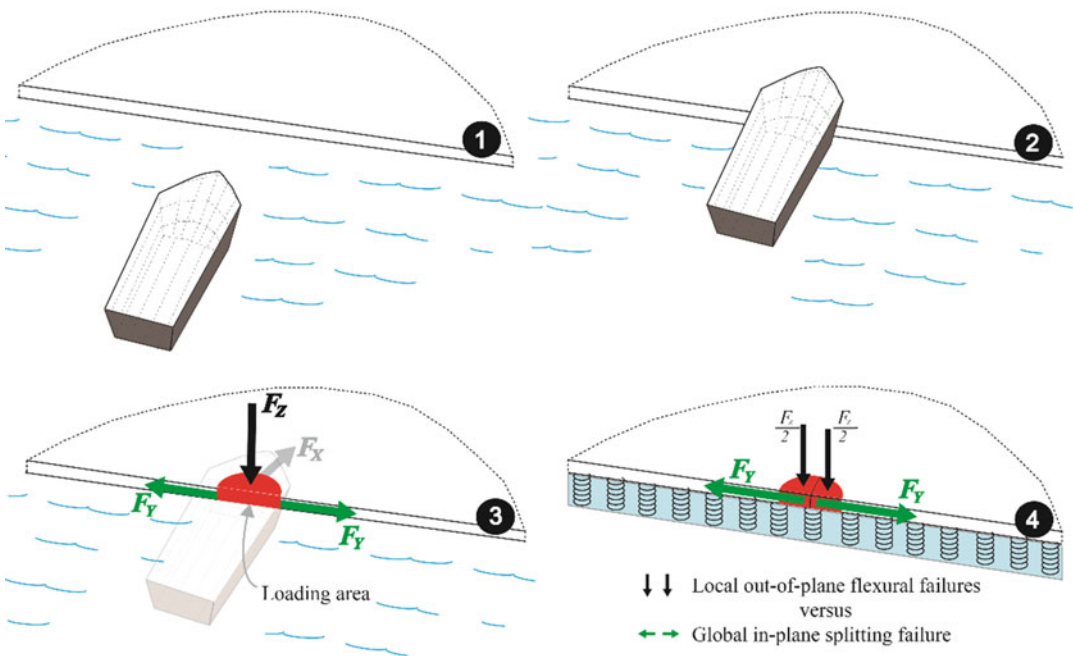
However, ice loads calculations for a sloping structure within a broken ice field are still in preliminary stage and are considered only for highly idealized scenarios. One of the most simple approaches is to idealize a broken ice field as level ice according to the *equivalent ice thickness*

concept (Keinonen 1996). Then, all the previously developed methods for level ice conditions can be applied by modifying the thickness information. However, as cited by Riska and Coche (2013), a separate study has shown that this concept only gives a better result for ship transit calculations (though a higher transit speed can be predicted) than for ice loads estimations.

Failure Modes of Floe Ice

During floe ice – sloping structure interactions – two major failure modes of the ice floes have been frequently observed (see Fig. 4). These are:

- Global in-plane splitting failure (Lu et al. 2015a)



Sloping Structure: Floe Ice Interactions, Fig. 5 Simplified global contact force and an ice floe's subsequent failures

- Local out-of-plane flexural failure (Lu et al. 2015b)

These two failure modes are competing with each other (Lu et al. 2016) and are frequently observed in both model ice tank tests (see Fig. 4a) and in field observations (see Fig. 4b).

Generally speaking, when the floe ice is rather huge or with significant confinement, only the local out-of-plane failure shall take place. However, when the floe ice is relatively small and with less confinement, the global splitting failure shall take place to fracture the entire ice floe and reduce the ice resistance. The global splitting failure of an ice floe can be considered as a load-releasing mechanism (Lu et al. 2015a).

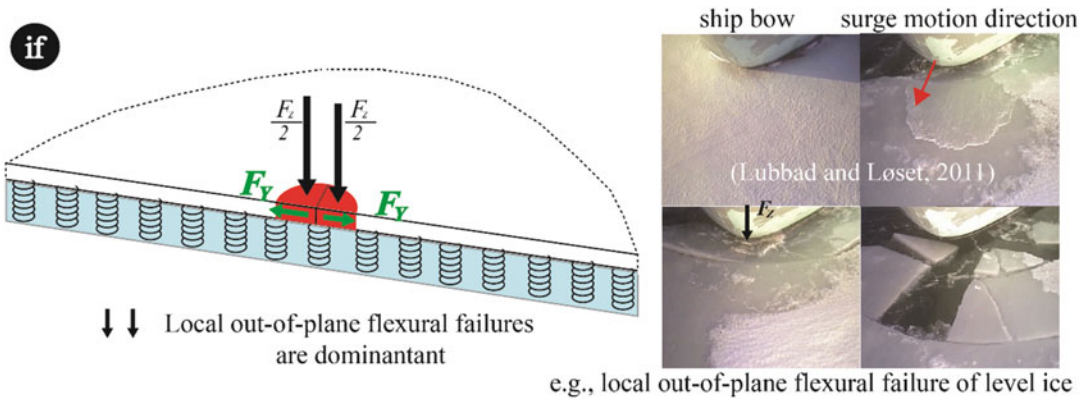
Interaction Process

As a sloping structure (e.g., an icebreaker) impacts with an ice floe, as shown in Fig. 5① to ②, complicated stress field forms within the contact area. However, from a global

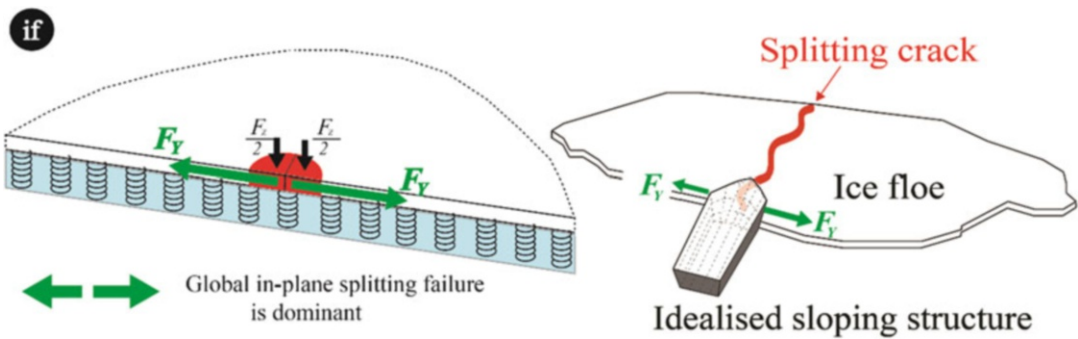
perspective, three different contact force components can be isolated along with the global coordinate system (see Fig. 5③). These are a vertical force component F_z , which leads to a potential out-of-plane flexural failure, a pair of horizontal loads F_y , which lead to a potential in-plane splitting failure, and another in-plane force component F_x , which increases the compression within the ice floe.

The force components F_z and F_y are considered as the direct source for the dominant out-of-plane failure and in-plane failure separately.

If a sloping structure is interacting with a relatively large ice floe (e.g., level ice), an ice floe's lateral boundary confinement is significant, or if the contact between the structure and the ice floe leads to a fairly large vertical force component, then the ice floe's dominant failure mode is expected to be a local out-of-plane flexural-type failure, as shown in Fig. 6. On the contrary, for a relatively small ice floe, who has limited/no lateral confinement, its dominant failure mode is expected to be a splitting failure mode, as shown in Fig. 7.



Sloping Structure: Floe Ice Interactions, Fig. 6 Dominant local out-of-plane flexural failure



Sloping Structure: Floe Ice Interactions, Fig. 7 Dominant global in-plane splitting failure

The competition between these two dominant failure modes at the initial floe ice – sloping structure contact – is illustrated in Fig. 5④. It is important to formulate the needed force to arouse these different failure modes and thereby determine which is the dominant failure mode.

Calculation Methods

Different methods are available to calculate the splitting and flexural type failure modes of an ice floe. These methods can largely be categorized into those based on numerical methods, analytical methods, and empirical methods.

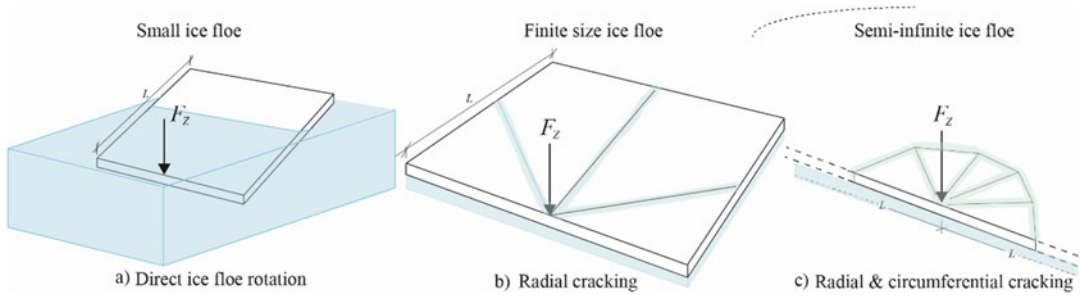
In this entry, we will mainly focus on the analytical methods and their derived conclusions. Related numerical methods are presented in a different entry.

Local Out-of-Plane Flexural Failure

The out-of-plane flexural type failure is mainly influenced by the size of the ice floe and ice thickness. Depending on the floe size, three different scenarios can take place (see Fig. 8):

- Direct rotation of a small ice floe
- Radial crack of an ice floe with size less than 2ℓ (the formulation of ℓ is given in Eq. (2))
- Circumferential crack formation

The direct rotation of a small ice floe can simply be a floating rigid body problem, and its formulation has been given by Lu et al. (2016). The radial crack of a small ice floe is a “plate on elastic foundation” problem. Analytical solution of this type of problem can be solved by the symplectic method (Li et al. 2013). These “failures” in Fig. 8a and b occur usually for small ice floes and



Sloping Structure: Floe Ice Interactions, Fig. 8 Different scenarios of an ice floe under an out-of-plane vertical force

contribute to relatively small ice load (i.e., F_Z). Usually it is the out-of-plane bending failure of a semi-infinite ice floe that leads to the largest vertical contact force. Such failure process involves the development of both radial cracks and circumferential cracks (see Fig. 8c) (Kerr 1976; Kerr and Kwak 1993). It is normally the initiation of circumferential crack that leads to the final failure of the ice floe. Extensive research has been conducted in

this regard (see e.g., (Bažant 2008, Kerr 1976, Kerr and Kwak 1993, Dempsey et al. 2000, Sodhi 1995, 1996, 1997; Dempsey et al. 1995, 1998; Nevel 1958, 1961, 1965)).

As a static approximation, for the vertical direction contact force F_Z , which leads to the out-of-plane bending failure, the simplified formula by Nevel (1972) can be adapted to in Eq. (1):

$$F_Z = \begin{cases} \frac{\sigma_f h^2}{3} \tan\left(\frac{\alpha}{2}\right) \left[1.05 + 2\frac{\delta}{\ell} + 0.5\left(\frac{\delta}{\ell}\right)^2 \right] & (\alpha \leq 90) \\ \frac{m\sigma_f h^2}{3} \tan\left(\frac{\alpha}{2m}\right) \left[1.05 + 2\frac{\delta}{\ell} + 0.5\left(\frac{\delta}{\ell}\right)^2 \right] & (90 < \alpha \leq 180) \end{cases} \quad (1)$$

in which,

σ_f is the flexural strength of sea ice, [kPa] α is the angle of the wedge where the local bending failure occurs, in [deg]

δ is the maximum penetration depth and is considered equivalent to the loading area size at the wedge tip, [m]

m is the number of wedges that are bent off from the original ice sheet and is set as a random number between 2 and 3 (based on lab and field observations)

ℓ is the characteristic length [m] of sea ice and is calculated by Eq. (2):

$$\ell = \left(\frac{Eh^3}{12\rho_w g} \right)^{1/4}, \quad (2)$$

in which, E , ρ_w , and g are the Young's modulus [kPa], water density [kg/m^3], and gravitational acceleration [m/s^2], respectively.

Global In-Plane Splitting Failure of an Ice Floe

The in-plane splitting type failure, however, is not as thoroughly studied as the out-of-plane type flexural failure. Among many obstacles to achieve a general analytical solution is the arbitrary geometries of an encountered ice floe. Bhat (1988) and Bhat et al. (1991) offered finite element method (FEM) results for the idealized square and disk-shaped ice floe's splitting. Later, Dempsey et al. (1993) offered the weight function-based analytical solution to characterize the splitting of a square ice floe. The analytical solution was

further extended to rectangular ice floes of arbitrary width to length ratio (Dempsey and Mu 2014). After summarizing previous research effort (Lu et al. 2015a) and developing analytical formulas for crack kinking in off-center splitting scenario (Lu et al. 2018b, c), the analytical framework to characterize the splitting failure of an ice floe can be constructed. However, the utilized analytical solutions to describe the splitting of an ice floe of arbitrary geometry are not an exact solution; it is more an approximation. To achieve more accurate results on the splitting failure of an arbitrary ice floe, a computational approach can be adopted, e.g., by the eXtended Finite Element Method (XFEM) (Lu et al. 2018a). In this entry, we shall only focus on the analytical fracture approach.

In summary, for the splitting failure modes, two types of splitting scenarios can be considered: the analytical formulas to deal with the central

splitting and off-center splitting scenarios are presented in Eqs. (3) and (4) to calculate the “splitting force term” F_Y (see Fig. 7).

$$F_Y(a) = \frac{hK_{IC}\sqrt{L}}{H(a, 0)} \quad (3)$$

in which,

K_{IC} is the fracture toughness of sea ice, [$\text{kPa}\sqrt{\text{m}}$]
 $H(a, 0)$ is the weight function of a rectangular ice floe with a centered edge crack. Its formulation can be found in Dempsey and Mu (2014)
 a is the nondimensional crack length given by $a = A/L$, whereas A is the crack length and L is the length of the an idealized rectangular ice floe

$$\begin{aligned} W_1 &= \left(\frac{F_X}{hK_{IC}} \right)^2 \left[f_1^2 \left(\frac{A_0}{W_1}, \beta_{YX} \right) + f_2^2 \left(\frac{A_0}{W_1}, \beta_{YX} \right) \right] \\ \beta_{YX} &= \frac{F_Y}{F_X} \\ f_1 \left(\frac{A_0}{W_1}, \beta_{YX} \right) &= \sqrt{\frac{4\pi}{\pi^2 - 4}} \left(\frac{A_0}{W_1} \right)^{-1/2} \left(\beta_{YX} - \frac{1}{\pi} \right) + \frac{2.0284(A_0/W_1) + 2.9890}{(A_0/W_1)^{-1.3569} + 1.0499} \beta_{YX} + \frac{1}{2} \frac{0.1856 - 0.2174(A_0/W_1)^{0.1635}}{(A_0/W_1)^{-6.2861} + 0.4092(A_0/W_1)^{0.1635}} \\ f_2 \left(\frac{A_0}{W_1}, \beta_{YX} \right) &= \frac{1.8134(A_0/W_1) + 0.5498}{(A_0/W_1)^{-1.4702} + 1.2041} \beta_{YX} + \frac{1}{2} \frac{0.4051 - 1.3238(A_0/W_1)^{0.4339}}{(A_0/W_1)^{-0.9953} + 1.0099(A_0/W_1)^{0.4339}} \end{aligned} \quad (4)$$

In Eq. (4),

A_0 is the length of the initial radial crack due to local bending failure, in [m]; it can be calculated as $A_0 = 2\ell$, with the characteristic length ℓ formulated in Eq. (2).

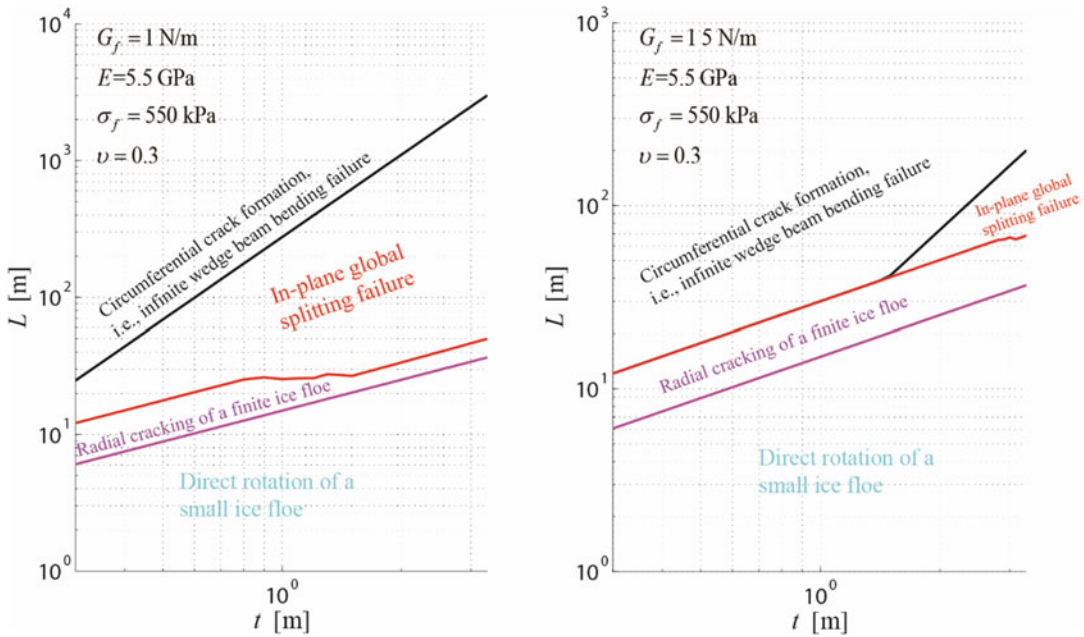
W_1 is the distance from the contact point to the closest free edge. The width of the bounding box of an arbitrary floe can be written as: $W = W_1 + W_2$ and $W_1 < W_2$. In general, $W_2 \geq 1.5 \cdot W_1$ for Eq. (4) is required to be valid. Otherwise, Eq. (3) is a better approximation.

$\beta_{YX} = F_Y/F_X$ represents the contact force ratio (see Fig. 5). The contact force F_X opposite to the

ship's transit direct has a tendency to close the splitting crack.

Failure Map

With the previous introduced analytical formulae, we can investigate each parameter's influence on the failure modes during floe ice – sloping structure interactions. It is the failure mode contributing to the least ice load that shall take place. Given this assumption and using typical ice properties and thickness information, two examples of failure map are shown in Fig. 9. In these two failure



Sloping Structure: Floe Ice Interactions, Fig. 9 Failure map of an ice floe while interacting with a sloping structure with $\beta_{YX} = 0.5$

maps, for illustration purpose, the contact property has been fixed as $\beta_{YX} = F_Y/F_X = 0.5$.

The general observation from the failure maps is:

- The direct rotation of an ice floe is anticipated if floe size $\leq \ell$.
- Radial or circumferential cracking of a finite size rectangular ice floe is expected if the floe length or width falls in the range of $(\ell \sim 2\ell)$.
- Ice floes with both length and width larger than 2ℓ can be considered as a semi-infinite ice floe whose failure is featured by sequentially forming radial and circumferential cracks. Nevel's (1972) approximate solutions (Eq. (1)) can be utilized to calculate the vertical force that leads to this failure type.
- The in-plane splitting failure only takes place for intermediate ice floes, and thicker ice has a high change to be split.

Applications

As floe ice is a more general ice condition in the Arctic, the theories and method presented in this

entry have wide applications regarding Arctic structural design and Arctic marine operations.

The analytical solutions introduced can be implemented in a numerical simulator to model the fracture of ice floes. This has been partly or fully achieved in simulators, e.g., (Lubbad and Løset 2011; Septseault et al. 2014; Dudal et al. 2015). These simulators have been widely used in Arctic offshore structural designs, e.g., calculation of the global ice load on the Swedish ice-breaker Oden (Raza et al. 2019; Lubbad et al. 2018), the salvage operation of the stranded fishing boat, Northguider (Berg et al. 2020; Lubbad et al. 2020), etc.

Moreover, the theoretical results have great implications to Arctic marine operations. For example, an often-raised question on the target floe size for an ice management operation can be found in the failure map. As ice failure is less likely when the floe size is less than 2ℓ , this floe size can be set as the target. Additionally, during ice management operations, the parallel channel's spacing can be optimized by using Eq. (4) presented above. The aim is to fracture more ice with minimal operation effort (i.e., more spacing distance between parallel channels).

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Sloping Structure–Level Ice Interactions

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Synonyms

Level ice; Sloping structure

Definition

Sloping structure: Offshore structure/ships with a sloping surface which promotes the flexural failure of sea ice.

Level ice: Level ice is a region of ice with relatively uniform thickness (ISO19906 2019).

Ice breaking load: The force required to break the incoming level ice. It takes place in the initial ice breaking phase. Normally the beam on Winkler-

type foundation formulation can be utilized to calculate this force term. However, for wide sloping structures, the influence from rubble accumulation should be accounted for (Lu et al. 2014a).

Ice breaking length: During level ice and sloping structure interactions, ice fails mainly in flexural failure mode and presents certain length characteristics. The general trend is that higher interaction velocity leads to shorter ice breaking length. More secondary ice breakings (e.g., aroused by rubble accumulation) also leads to shorter ice breaking length.

Ice rotating load: The force required to rotate the already broken ice. This happens during the ice rotating phase. Depending on the rubble accumulation amount, ice block length, and ventilation effect, secondary ice breaking process can also take place.

Ventilation effect: During ice rotating phase, depending on the ice rotating speed, it possible that the water does not have fill the top of the rotating ice block, leaving a ventilated area above. This introduces a significant amount of extra force during the ice rotating phase (Valanto 2001; Lu et al. 2012a).

Backfill effect: During ice–sloping structure interactions, due to the inherent slow interaction speed, the ventilated area can be partially flooded by the backfill water. This can alleviate certain amount of ice rotating load. This load term is highly dependent on the rotating speed. Usually a fully ventilated scenario is assumed for level ice–ship interactions (Valanto 2001) and a partially ventilated process can be assumed for level ice–wide sloping structure interactions (Lu et al. 2013, 2014a).

Scientific Fundamentals

Sloping Structure Types

The sloping surface of an sloping structure can be conical, multifaceted, or flat (ISO19906 2019); and the structure can have either downward or upward sloping surfaces.

The conical sloping structure has been widely utilized in the confederation bridge piers (Mayne 2007), jacket offshore structures (Yue et al. 2007),

wind turbine foundations (Barker et al. 2005), light houses (Määttä et al. 1996; Brown and Määttä 2009), etc.

In addition, it is possible to separate sloping structures into those that are wide and narrow sloping structures (Lu et al. 2014a).

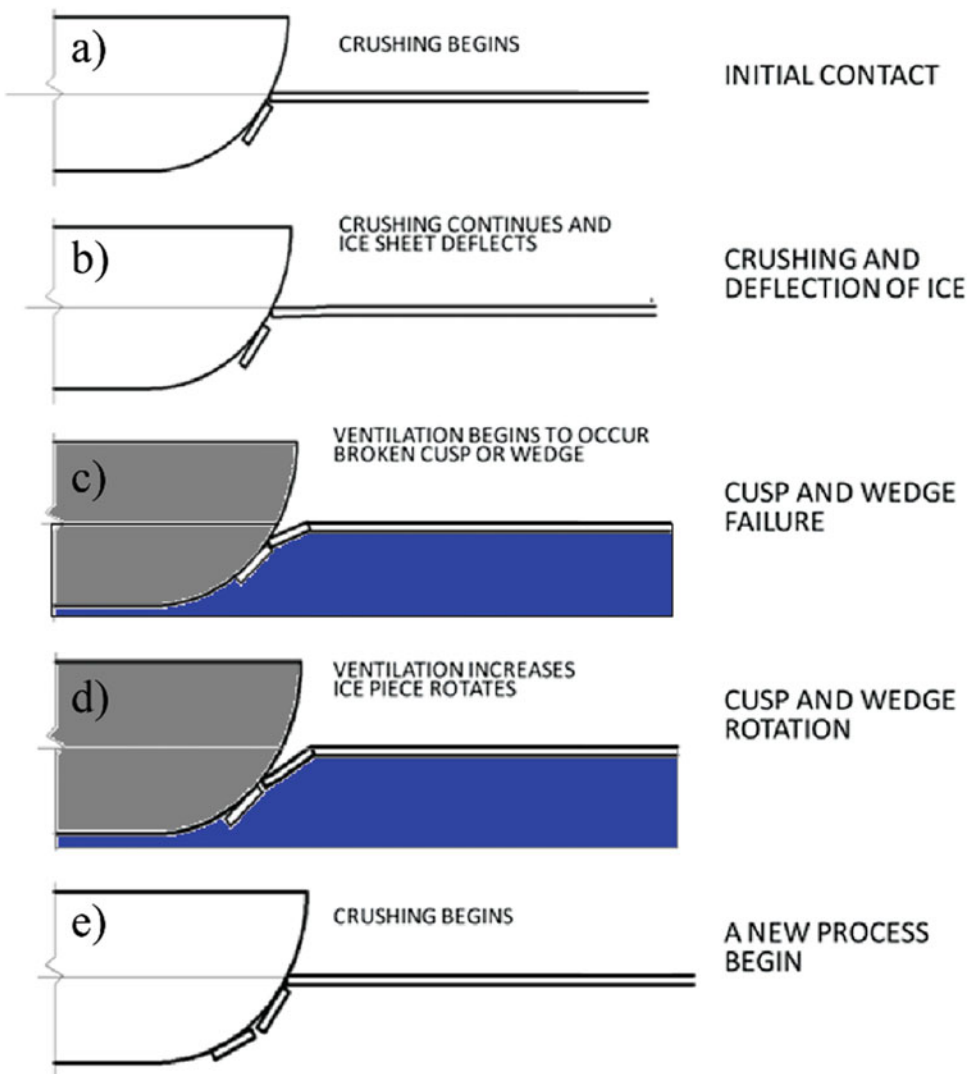
Interaction Process

Level Ice–Ship Interactions

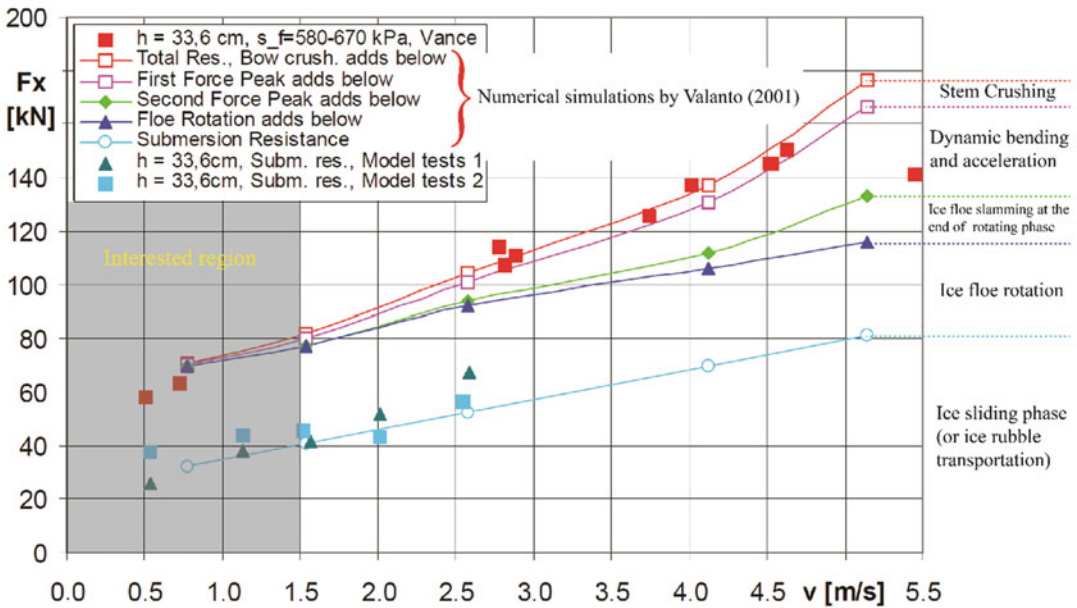
Regarding level ice–ship interactions, previous researchers, e.g., (Kotras et al. 1983; Naegle

1980; Lindqvist 1989; Kämäräinen 2007), have identified at least three major interaction phases and several ice load components that can be studied separately. For example, the level ice–ship interaction processes were depicted by Kotras et al. (1983) as:

- Icebreaking phase (shown in Fig. 1a, b, and c), in which crushing failure of ice occurs at the initial contact followed by bending (or flexural) failure of the intact ice sheet into wedge-shaped broken ice blocks.



Sloping Structure–Level Ice Interactions, Fig. 1 Level ice and ship interaction processes. (After Kotras et al. 1983)



Sloping Structure–Level Ice Interactions, Fig. 2 Comparative contributions of different load components versus interaction speed. (After Valanto 2001)

- Ice rotating phase, in Fig. 1c and d denote the ice rotating process involving the “ventilation” effect (i.e., no water flooding on top of the rotating ice piece, which creates huge pressure differences above and below the rotating ice blocks).
- Ice rubble sliding phase: as a new interaction circle begins, the last plot in Fig. 1e implicitly illustrates that ice rubbles sliding along the ship hull signifies the rubble transportation process.

Along with this depiction, Valanto (2001) quantified the comparative contribution of different force components, as shown in Fig. 2.

Considering the ice drifting velocity, the shaded area in Fig. 2 is approximately the interested region for level ice and fixed or floating offshore structure interactions. Within the shaded area, several of the force components identified by Valanto (2001) are expected to be absent or insignificant (e.g., ice floe slamming at the end of an ice rotating phase). In general, we highlight the potential largest three force components, i.e., the ice loads due to the ice-breaking phase (i.e.,

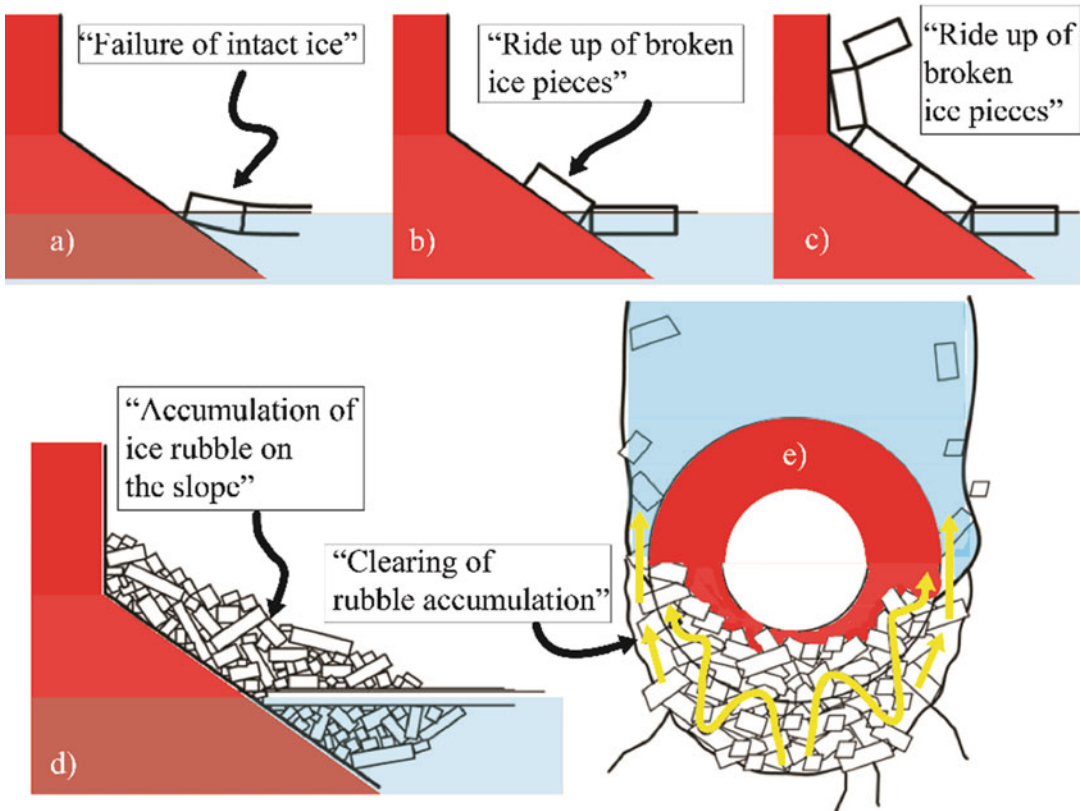
labeled as “dynamic bending and acceleration” in Fig. 2), the ice loads due to the ice rotating phase (i.e., labeled as “ice floe rotation” in Fig. 2), and the rubble transportation phase (i.e., labeled as “ice sliding phase” in Fig. 2). Figure 2 shows that the ice sliding phase contributes almost half the total ice resistance an ice-going ship encountered.

The Lindqvist formula (1989) can be utilized to calculate these different force terms.

Level Ice–Narrow Sloping Offshore Structure Interactions

In terms of level ice interacting with a sloping offshore structure, ISO19906 (2019) illustrates the process in Fig. 3 with an example of an upward sloping conical structure. A vast literature source exists that proposes various calculation methods for all of these force components (detailed literature review can be further explored in literature, e.g., (Lu et al. 2014a)).

As an analogy to the level ice–ship interaction research introduced in the previous section, the plots can also be categorized into:



Sloping Structure–Level Ice Interactions, Fig. 3 The interaction process between level ice and an upward sloping structure

- Ice breaking phase shown in Fig. 3a
- Ice rotating phase shown in Fig. 3b
- Ice rubble transportation phase shown in Fig. 3c, d, and e

Unique physical process occurs within these three different phases. From a mechanical point of view, for the ice breaking phase, the major physical process is fracturing of the incoming level ice. Ice material can be treated as purely elastic; for the ice rotating phase, depending on the interaction speed, rather fierce fluid–structure interaction scenarios occur (an ice piece can either be treated as an elastic body or a rigid body depending its actual size (Lu et al. 2013); for the rubble transportation process, it is primarily a multibody interaction system, and each single ice rubble can be treated as a rigid body; or the entire rubble volume could even be idealized as a

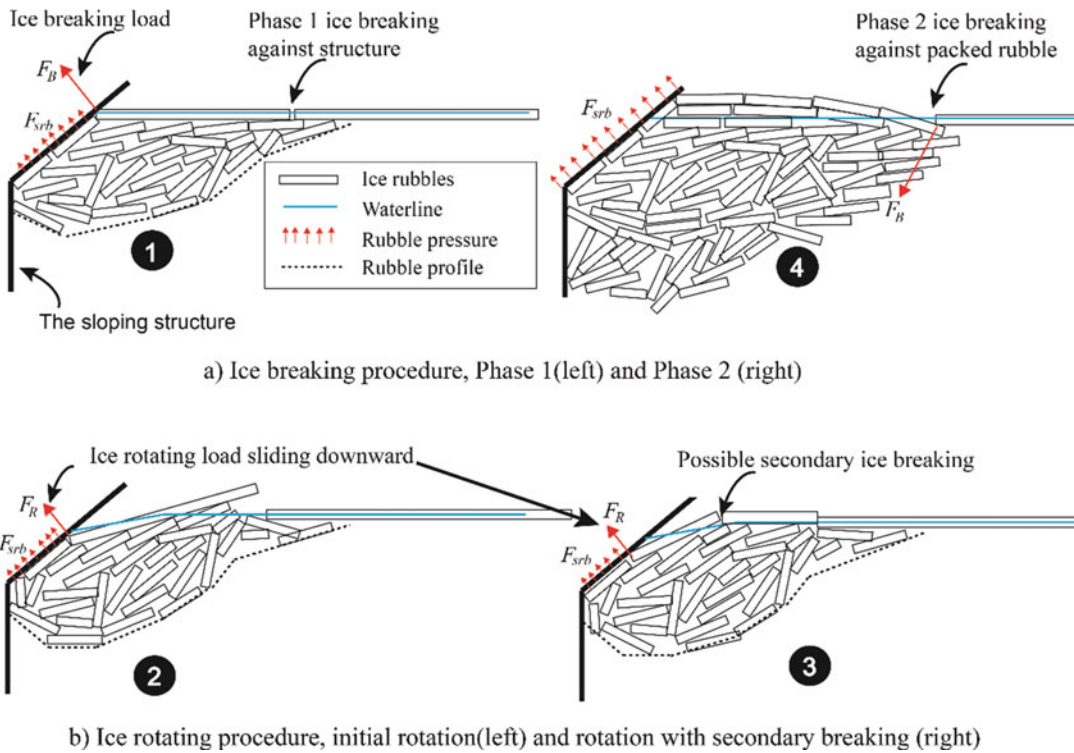
continuum (e.g., Serré and Liferoev 2010; Serré 2011).

The formulas proposed by Croasdale and Cammaert (1994) and adopted in the standard (ISO19906 2019) can be utilized to calculate these different force components.

Level Ice–Wide Sloping Offshore Structure Interactions

The same three interaction phases as presented before they can be identified for the interaction between level ice and a wide sloping structure. The only difference with regard to wide sloping structure interaction is the influence of rubble accumulation. This is illustrated in Fig. 4.

The presence of ice rubbles influences both the icebreaking and ice rotating phases. Relatively new formulations, comparing to those utilized by Croasdale and Cammaert (1994), can be found in



Sloping Structure–Level Ice Interactions, Fig. 4 The proposed interaction mechanism between level ice and a wide sloping structure under the influence of rubble accumulation

the literature (Lu et al. 2014a). Alternatively, one can also resort to numerical simulation approach, e.g., Discrete Element Method (DEM) (Paavilainen and Tuhkuri 2012, 2013; Paavilainen et al. 2006, 2009, 2010, 2011), to simulate the rubbing process and calculate the ice load on a sloping structure.

The design ice load in this case is found to be the buckling of force chains (Paavilainen and Tuhkuri 2013).

Calculation Methods

As the previous section illustrates, the interaction between level ice and a sloping structure involves distinct physical processes such as ice fracture, fluid structure interaction, and ice rubble accumulations. Different calculation methods are available for different interaction processes, and it is also possible to calculate all three processes in one

set of formulations/numerical simulations. Different categories can be identified as follows:

- A purely analytical approach: e.g., analytical solutions for level ice and sloping structure interactions (Croasdale and Cammaert 1994; Ralston 1977, 1980; Lindqvist 1989; Lu et al. 2013, 2014a). These analytical solutions typically involve a large number of assumptions, though are effectively suitable for generating a sizeable result pool for probabilistic analysis for the design stage.
- A purely numerical approach: earnest ice material modeling based on the finite element method (FEM) that accounts for the fracturing of ice floes and the discrete element method (DEM), which accounts for the rubble transportation process. With regard to the ice fracture simulation, the cohesive zone model utilized in FEM (e.g., Lu et al. 2012b, 2014b; Konuk et al. 2009a, b; Konuk and Yu 2010,

Kuutti et al. 2013, Wang et al. 2018, 2019) and DEM (Tuhkuri and Polojärvi 2018) are of great potential. More methods of this category shall be elaborated in the entry ► “Numerical Simulation Floe Ice–Sloping Structure Interactions”.

- A hybrid “empirical + numerical” approach with both empirical and numerical implementations, e.g., the numerical simulators developed by Liu et al. (2006) and Su et al. (2010) used empirical formulae to calculate the ice breaking and rubble transportation phases, where a numerical approach was adopted to detect contact between the ship hull with an intact ice sheet.
- A hybrid “analytical + numerical” approach to simulate general ice–structure interactions (including ice–sloping structure interactions). Analytical formulas to simulate the typical ice failure modes (Lu et al. 2015a, b, 2016, 2018a, b) are available, and can be coupled with existing multibody dynamic simulators to perform ice–sloping structure interaction simulations, for example (Lubbad and Løset 2015; Lubbad et al. 2015, 2018a, b; Tsarau et al. 2018).

Applications

The theories and methods presented here have wide applications in most man-made structures’ design in icy waters. For example, ice-going vessel’s global ice load calculations with the purely analytical methods (Suyuthi et al. 2011; Kjerstad et al. 2018; Kim et al. 2020); designs of ice-breakers with the empirical + numerical method (see the reference above); and calculations of ice load on a drilling island in the Caspian Sea (Croasdale 2012), on light houses (Määttänen and Hoikkanen 1990; Määttänen et al. 1996), on bridge piers (Brown and Määttänen 2009; Mayne 2007), and on various offshore platforms in the high North.

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SLS – Serviceability Limit States

► Design of Renewable Energy Devices

SMB: Surface Marker Buoy

► Surface Buoy

Soft YOKE Single Point Mooring System

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Synonyms

Mooring system; Positioning system; Single point mooring system

Definition

The soft YOKE single point mooring system is a kind of marine single point mooring device (Deng 2012). The ship such as the oil tanker is positioned in the predetermined sea area through a freely

rotatable fixed point, and meanwhile, has the functions of transporting well flow, electricity, and communication. The soft YOKE single point mooring system can make the hull such as Floating Production Storage and Offloading System (FPSO) have the wind direction effect, allowing the moored hull to rotate 360° around the mooring point, so that the hull always along the resultant force direction of wind, waves, and flow (Xiao 2016). It is in the state of minimum stress, and it also reduces the motion response of the hull and the mooring load on the hull. The soft YOKE single point mooring system is one of the important devices in the offshore oil development. It has two main functions: one is to locate the hull such as FPSO; the other is used as the terminal for delivering crude oil. The soft YOKE single point mooring system can moor the super large FPSO or oil tanker, which is strong against the marine environment. However, due to the rigid connection to the seabed, the system is mainly suitable for shallow water operations (Wei 2012).

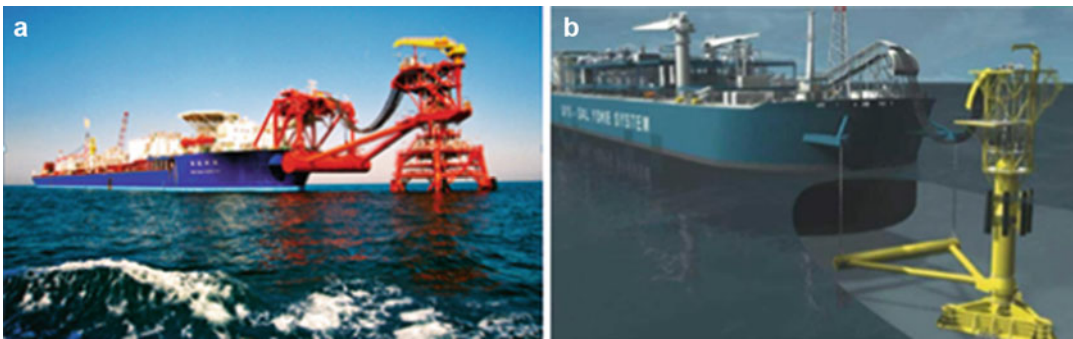
Development History

In 1974, a single point mooring device for steel structure connecting oil tankers was developed in the world (Shouwei et al. 2006). The A-shaped steel frame was used instead of the steel wire rope to connect the oil tanker, which avoided the collision between the oil storage tank and the floating tank, and reduced a lot of maintenance work. It is called rigid arm single point mooring system. In 1980, the first floating production, storage, and loading and unloading system of rigid arm single point mooring was installed in the Philippine waters. The device can be used for oil and gas processing, storage, and external transportation. The rigid arm single point mooring device is shown in Fig. 1.

The rigid arm mooring method is mainly composed of a triangular steel structure and a tower (Geng-xian 2009). The triangular steel structure is connected to the tanker at one end and to the tower at the other end. This mooring method allows the tanker to rotate to the direction that is least affected by environmental forces, but the surge



Soft YOKE Single Point Mooring System, Fig. 1 Rigid arm single point mooring system



Soft YOKE Single Point Mooring System, Fig. 2 Soft YOKE single mooring system. (a) Above water soft YOKE single mooring (b) Under water soft YOKE single mooring

and pitch motion of the tanker is constrained. The load of the tanker's surging process is transmitted to the rigid arm and the tower, which is easy to cause damage to the triangular steel frame.

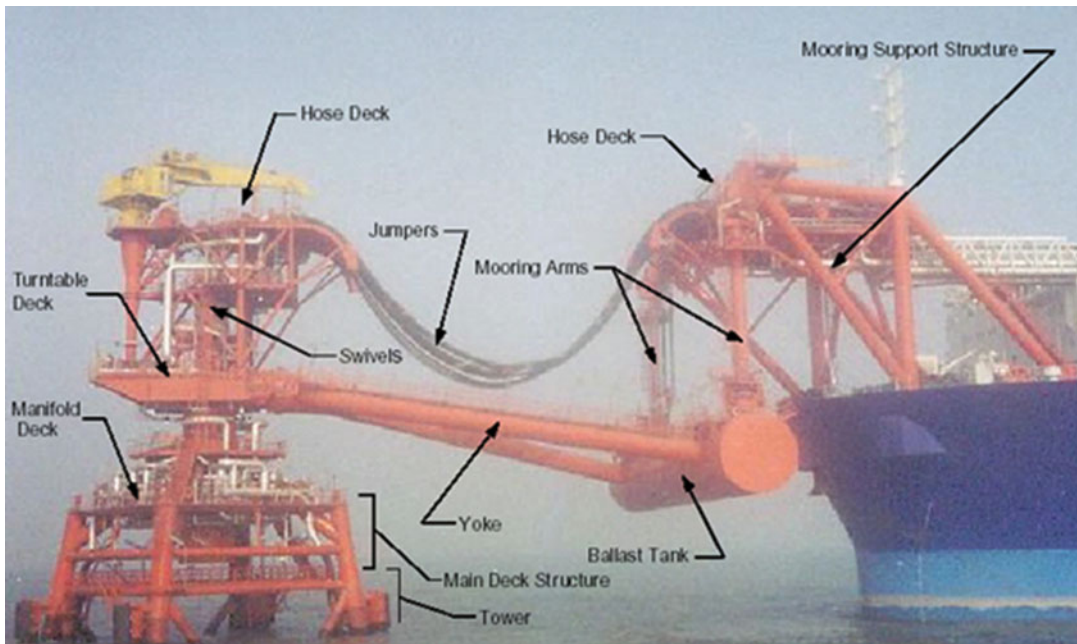
In November 1981, soft YOKE single mooring system was developed and it was designed and installed at offshore oil field in Philippines. According to the position of YOKE, above or below the water surface, soft YOKE single mooring system can be divided into above water soft YOKE single mooring system and underwater soft YOKE single mooring system. At soft YOKE single mooring system was well developed, as shown in Fig. 2.

Operational Principle and Component

The tower soft YOKE single mooring system mainly includes mooring arms, ballast tank, YOKE, jumpers, hose deck, turntable deck,

swivels, manifold deck, main deck structure, and tower. A mooring support structure at the bow of the vessel is installed for connecting the mooring device as shown in Fig. 2. The upper end of the mooring arm is hinged to the mooring support structure on the tanker, the lower end of the mooring arm is hinged to the ballast tank, and the YOKE is hinged to the turntable deck. The jumpers deliver crude oil from the pipeline to the tanker. This form of single point mooring facilities allows six degrees of freedom motion of tanker. The quality of ballast tanks is hundreds of tons, and it can provide the recover stiffness during the surge and pitch motion of the tanker, as shown in Fig. 3 (Shouwei et al. 2006; Geng-xian 2009; Yao 2013).

The upper and the lower part of the mooring arm are connected with the FPSO bow mooring bracket and the soft YOKE, respectively, through three axis hinges. When the mooring system is in



Soft YOKE Single Point Mooring System, Fig. 3 The composition of single point mooring facilities

equilibrium, the mooring leg of the suspended YOKE is vertical. YOKE is driven by mooring legs when FPSO surges, and YOKE generates restoring force under the action of ballast gravity, which makes FPSO return to the equilibrium position of surge motion. In addition to the weather vane effect, such single point mooring device does not restrict the roll and yaw motion of the tank. Surge and pitch motion can also occur. When the wind vane effect moves the tanker to the minimum force position, the surge motion is decayed because of the inertia of the hundreds of tons ballast tank. It reduces the force acting on YOKE and horizontal force on tower, thus the tower can be prevented from being damaged by a large horizontal load. It is the advantage of soft YOKE single mooring system.

The mooring structure of the on-water soft YOKE mooring system is above the water surface. All structures, especially the mooring arms and rotating parts, are above the water surface for monitoring and maintenance. It has a defect that horizontal thrust acts on the top of the tower structure, causing a large horizontal overturning moment. It is easy to cause the tower structure to

fall; the underwater soft YOKE mooring system is below the water surface, and the force of YOKE acts on the bottom of the tower, and the overturning moment acting on the tower is small, which is beneficial to the tower to maintain stability. The shortcoming is that the mooring arm and the rotating component are below the water surface, especially the mooring arm is subjected to a large environmental force, which easily causes damage, and the underwater structure and components are not convenient for monitoring and maintenance. At present, the offshore oil and gas development field tends to use the on-water soft YOKE mooring system.

The functions of the various parts of the single-point soft YOKE mooring system are as follows:

1. Tower-Jacket Structure

The single-point mooring system has a slip ring channel on which the liquid medium is continuously transported, so that corresponding oil risers, water pipes, and cable guards are also disposed under the tower. To prevent damage to these components, these pipes are placed inside the tower.

In addition, since the single-point tower is the center of rotation of the FPSO weathervane movement, it is subject to greater mooring forces.

2. Manifold Deck

The lowermost deck above the tower-jacket is a manifold deck, which is mainly equipped with oil and water pipelines, receiving ball pigs, cable junction boxes, and other facilities. Its center is penetrated by the general column, and the oil and water pipes are merged and then enter the upper slip ring system (fluid rotating passage) from the general column.

3. Turntable Deck

Above the manifold deck is a turntable deck, which is located in the middle of the tower-jacket and is the most special point of a single point. As a boundary between the single point mooring fixed part and the rotating part, the lower manifold deck and the tower are fixed structures, and the oil water passage, the electric instrument slip ring and the hose deck above it are all rotating parts. At the center of the turntable deck is the main bearing that rotates in a single point system, which supports the slip ring system, leak recovery system, and automatic lubrication system.

4. Hose Deck

Hose deck is in the top of the mooring tower, on one side of which near the FPSO the arc plate is set to make the arrangement of the jumper hose and cable convenient, on the other side the lifting crane is set to daily operation. In addition, hose deck is also equipped with hoses deck winch is used to drag the hose and cable.

5. Mooring Connected Components

FPSO is connected to the single point mooring tower by ship stem mooring support, mooring legs, soft YOKE, and turntable, which is shown in Figs. 4 and 5.

6. Slip Ring System

Slip ring system is a key component of the single point mooring tower, which is composed of communication slip ring, electro-slip ring, liquid slip ring (also called fluid rotary joint), and so on from top to bottom. The liquid slip ring is the core of oil-water transportation,

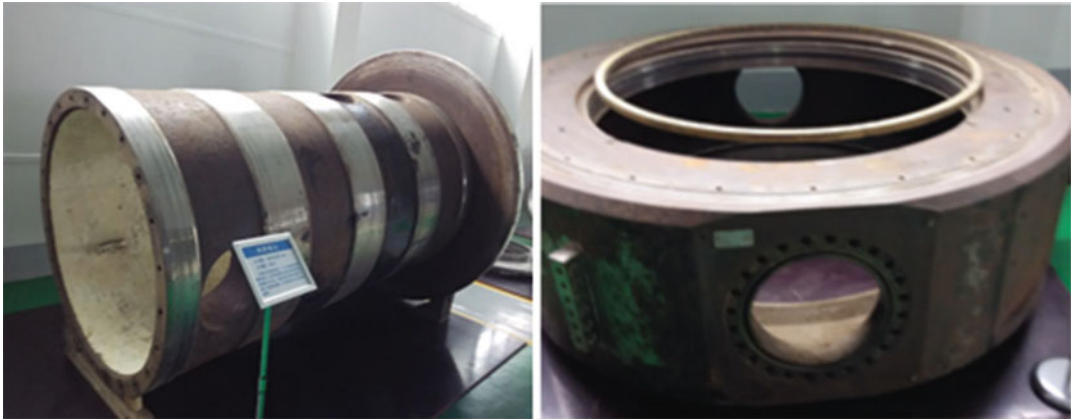


Soft YOKE Single Point Mooring System, Fig. 4 Ship stem mooring support



Soft YOKE Single Point Mooring System, Fig. 5 Mooring legs and YOKE

connecting the fluid pipeline equipment between the rotating part and the stationary part, as shown in Fig. 6. A plurality of fluid rotating joints is usually arranged to be stacked together. Each rotating joint consists of an inner ring (stator) and an outer ring (rotor), of which the inner ring is fixed on a single point of the general column and connected with the riser, the outer ring is connected to the mooring device and rotated with the FPSO.



Soft YOKE Single Point Mooring System, Fig. 6 The inner ring and outer ring of liquid slip ring



Soft YOKE Single Point Mooring System, Fig. 7 HYSY117. (Source CNOOC)

Application

With the development of the Soft-YOKE Single-point Mooring System, the YOKE-type SPM is less used for the offshore engineering structures. Due to the mooring arms and YOKE structure, a large horizontal thrust occurs. When the water depth is increasing, the height of the tower, as well as the horizontal thrust of the YOKE, is also enlarged. In case of the system collapsing, the main dimensions of the tower structure should be amplified, which leads to the explosive cost incensement. Therefore, the Soft-YOKE SPM is suitable for the shallow water-depth area.

At present, the worldwide largest FPSO moored by Soft-YOKE SPM is HYSY117 (Geng-xian 2009), which is working on the Penglai 19-3 offshore oilfield in Bohai Sea, China. Its overall length is 323 m, and width is 63 m. The oil storage capacity is two million BBLs, and the daily processing capacity of the crude could reach 190,000 BBLs, which means the HYSY117 could handle ten million cubic meters crude annually. According to the Soft-YOKE SPM, the FPSO could be long-term moored in its working area without extrication for its designed lifetime, say 25 years for HYSY117 (Yao 2013). Moreover, the whole system could bear 100-year sea state (Fig. 7).

Cross-References

- [External Turret Single Point Mooring System](#)
- [Internal Turret Single-Point Mooring \(SPM\) System](#)
- [Mooring System](#)
- [Positioning System](#)
- [Single Point Mooring System](#)

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Solid Core PCFs (SC-PCFs)

- [Fiber Optic Hydrophone](#)

Sonar

- [Sonar Technology](#)

Sonar System

- [Sonar Technology](#)

Sonar Technology

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Synonyms

[Active sonar](#); [Doppler current profiler](#); [Doppler velocity log for navigation system in underwater vehicle](#); [Passive sonar](#); [Side-scan sonar](#); [Sonar](#); [Sonar system](#)

Definition of Sonar

The word “SONAR” is the English abbreviation of Sound Navigation And Ranging. It is an electronic equipment that uses the propagation characteristics of sound waves underwater to complete tasks such as underwater detection, positioning, and communication through electro-acoustic conversion and information processing. Since the propagation of electromagnetic waves in water is much larger than that of sound waves, and no more efficient information transmission channels than sound waves have been found, sound waves are still the most ideal carrier for information acquisition in the ocean. As a result, the detection technology of underwater targets, sonar technology, was born (Elfes 1987).

Historical Moment

In 1906, Lewis Nixon, a British naval soldier, invented the first sonar, a passive listening device that was applied in the battlefield to detect submarines hidden in the water during World War I. These sonars can only be passively listened, so they belong to passive sonar. In 1915, French physicist Paul Langevin in cooperation with Russian electrical engineer Constantin Chilowski invented the first active sonar and detected the submarine by listening to the echoes reflected by the submarine. Their groundbreaking

work has had a profound impact on the subsequent sonar design (Naik et al. 2017).

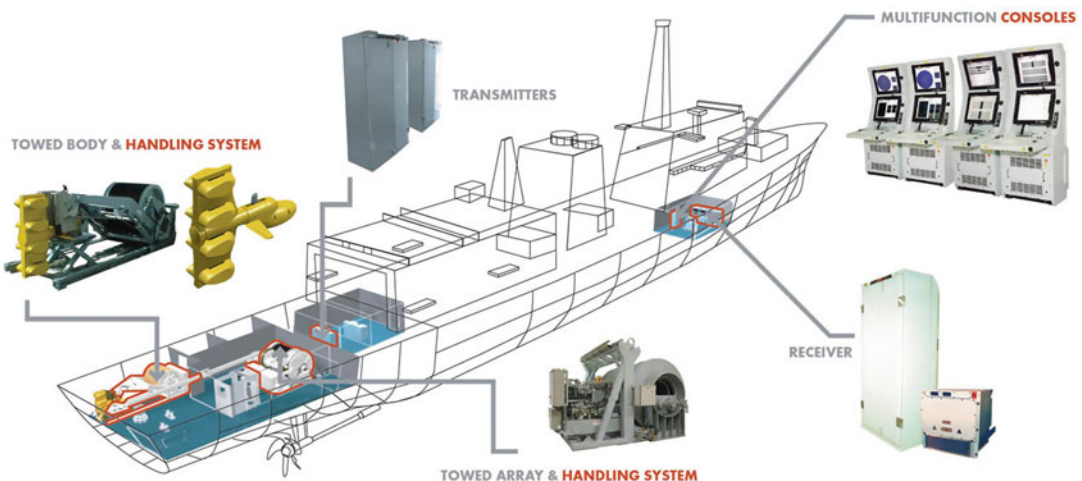
Composition of Sonar

The sonar system mainly consists of two components: the dry end and the wet end. The wet end is mainly composed of hydroacoustic transducers or a transducer array, and the dry end is mainly composed of a signal source, a transmitting device, a signal processing platform, a power supply device, and a display control unit. The array is composed of a number of hydroacoustic transducers arranged in a certain geometric pattern, and its shape is usually a spherical shape, a cylindrical shape, a flat plate shape, or a line array. It consists of the receiving array, the transmitting array, or the array that can both receive and transmit. The wet end system also includes connecting cables, underwater junction boxes, preamplifiers, lifting, swinging, pitching, retracting, towing, hoisting, placing device matched with the transmission control of the sonar array, and other auxiliary equipment such as sonar shroud. The dry end system generally has a subsystem for transmitting, receiving, displaying, and controlling, wherein the signal processing platform is an information data processing operation center. The composition of the UK 2087 sonar system is shown in Fig. 1.

The transducer is the basis of the sonar equipment. It is a device that converts sound energy and other forms of energy such as mechanical energy, electrical energy, and magnetic energy. Its working principle is to use some special materials that can expand and contract under the action of electric or magnetic fields. This effect is usually called piezoelectric effect or magnetostrictive effect. This effect causes vibrations on the surface in contact with water to radiate sound waves into the water and vice versa. The transducer has two main purposes: one is to emit sound waves underwater, called a “transmitting transducer,” which is equivalent to a speaker in the air; the other is to receive sound waves underwater, called a “receiving transducer.” It is equivalent to a microphone in the air. Transducers can often be used to transmit and receive sound waves at the same time in actual use; and transducers specifically used for receiving are also called “hydrophones” (Liu and Gong 2008). Figure 2 shows the commonly used underwater acoustic transducer for sonar.

Classification of Sonar

There are many kinds of sonar which can be classified according to the working mode, equipment object, installation platform, practical use, and technical characteristics. For example, it can



Sonar Technology, Fig. 1 The UK 2087 sonar system (<https://www.savetheroyalnavy.org/listening-to-the-ocean-the-secretive-enablers-in-the-underwater-battle/>)

be divided into active sonar and passive sonar according to the working mode; it can be divided into shipboard sonar, submarine sonar, airborne sonar, and shore-based sonar according to the equipment object; according to the installation platform, it can be divided into bow sonar, flank sonar, towed sonar, buoy sonar, etc.; according to the actual use, it can be divided into detection sonar, imaging sonar, positioning sonar, communication sonar, etc.; and according to technical characteristics, it can be divided into parametric array sonar, synthetic aperture sonar, multi-beam sonar, continuous wave sonar (Wang 2009). Figures 3, 4, and 5 show several typical sonars.

Active Sonar

Active sonar emits some form of sound wave signal into the water. When a sound wave

encounters a target such as a submarine or a fish group in the water and because of the acoustic characteristics of the target are greatly different from the seawater, a large reflection is generated. The reflected echo contains many information of the target, so the presence of the target can be judged according to the received echo signal, and the distance, azimuth, speed, and other parameters of the target are measured or estimated, as shown in Fig. 6. Specifically, the distance between two targets can be inferred by the time interval of echo signal and the transmitted signal, and the direction of the target can be inferred from the normal direction of the echo wave front, and radial velocity of the target can be inferred from the frequency shift between the echo signal and the transmitted signal. In addition, the shape, size, nature, and motion state of the target can be identified by the amplitude, phase, and variation of the echo. Most of the active sonars use a pulse system. In recent years, continuous wave systems have also been adopted, which are suitable for detecting icebergs, reefs, shipwrecks, seabed, fish schools, frogmen, mines, and submarines.

Passive Sonar

The sonar that uses the receiver transducer array to receive the noise or signal emitted by the target itself to detect the target is called passive sonar. Since the passive sonar itself does not emit a signal, the target will not be aware of the presence of sonar. The biggest difference between



Sonar Technology, Fig. 2 Underwater acoustic transducer

Sonar Technology, Fig. 3 Bow sonar (<https://www.navsea.navy.mil/Home/Warfare-Centers/NUWC-Newport/What-We-Do/Detachments/Seneca-Lake/Facility-Power/>)

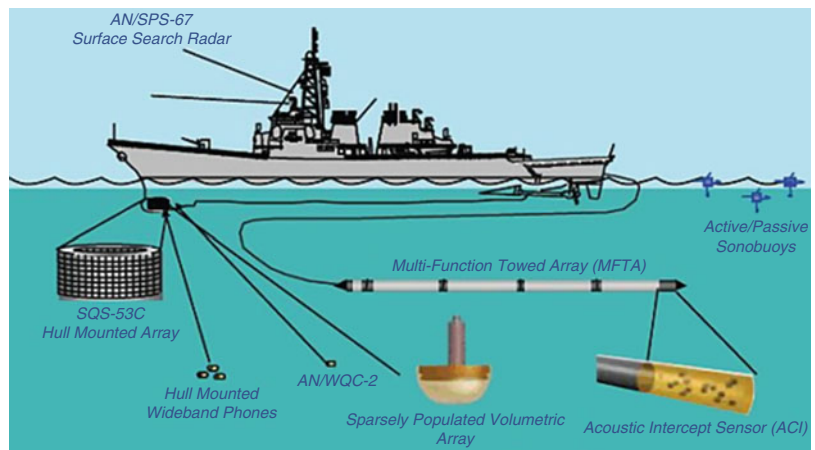


Sonar Technology,

Fig. 4 Dipping sonar
(<https://www2.l3t.com/oceansystems/contact.htm>)

**Sonar Technology,**

Fig. 5 Towed AN/SQQ-89A(V)15 sensor suite
(<https://www.militaryaerospace.com/computers/article/16715038/navy-asks-lockheed-martin-to-build-towedarray-sonar-systems-for-navy-surface-warships>)

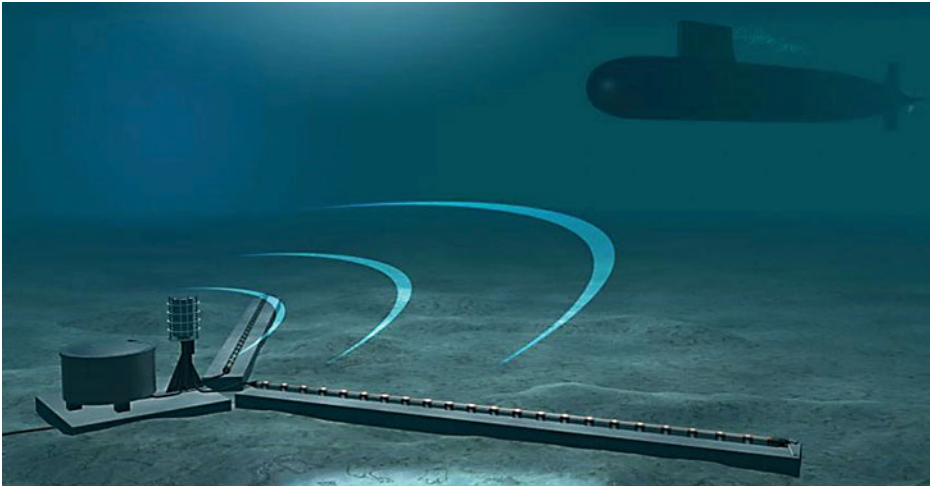


passive sonar and active sonar is that the former cannot obtain the waveform of the noise or signal emitted by the target in advance and usually the target noise is used as a signal, which becomes very weak after long-distance propagation. As a result, passive sonar often works in low signal-to-noise ratio conditions and thus requires more signal processing measures than active sonar. Passive sonar has no launching parts and generally works continuously, suitable

for detecting surface ships, submarines, torpedoes, etc.

Application Field of Sonar

The application of sonar technology is generally divided into military and civilian use, and the military dominates. It is the main technical means used by underwater navies for underwater



Sonar Technology, Fig. 6 Active sonars (<https://www.g.com/mil/20715297/>)

surveillance. It is used to detect, locate, classify, and track underwater targets for water. It is also used for the navigation and communication to ensure the navigation safety and information interaction between ships and submarines or all kinds of underwater unmanned platforms. Sonar technology has also been widely used in civilian use, and sonar equipment for various purposes has been born, mainly used for marine environmental parameter acquisition, marine resource exploration, underwater navigation and positioning, underwater information transmission, etc. With the development of sonar technology, civilian sonar equipment will play an increasingly important and irreplaceable role in marine development.

Military

Anti-submarine Detection

Submarines rely on their excellent concealment and powerful weapon delivery capabilities, and the threat is increasing day by day. Therefore, anti-submarine warfare is the top priority of underwater warfare. For this reason, people have developed various anti-submarine sonar equipment based on various platforms and even anti-submarine internet. At present, the anti-submarine detection sonar mainly includes the stern sonar, the side sonar, the tow sonar, on

the ship and the submarine, the hoisting sonar, the throwing buoy sonar on the plane, the shore-based sonar on the shore, the sonar submarine standard on the specific sea area, etc. Although people laid a large net for the submarine, the actual situation is that anti-submarine is still very difficult.

Small Target Detection

There are many kinds of small underwater targets, including carrying platforms, such as various underwater unmanned vehicles, and weapons platforms, such as torpedoes, mines, and frogmen. The detection of small targets usually requires a higher operating frequency to improve the spatial and temporal resolution of the sonar, to achieve the effect of the target imaging, thereby identifying the target and assessing the threat level of the target. Small target detection sonars are usually classified according to the object of detection, such as torpedo alarm sonar, hunting thunder sonar, and anti-frogman sonar. In recent years, the development of surface underwater unmanned platforms has brought new opportunities and challenges to small target detection.

Underwater Sound Guidance

Underwater weapons are typically torpedoes and mines. In order to improve the effectiveness of these weapons, various types of sensors are generally required to give them intelligent sensing

capabilities. For example, torpedoes have evolved from early direct torpedoes to today's acoustic self-guided torpedoes. The self-guided sonar including the torpedo head and the wake self-guided sonar on the back of the torpedo, such as the sound fuze of the mine, are guided by acoustic means to improve the hit rate.

Underwater Acoustic Communication

Replacing electromagnetic waves with sounds wireless communication underwater is currently the most effective technical approach. Submarine sailing underwater requires information exchange through underwater acoustic communication; cluster control of underwater unmanned aerial vehicles also requires underwater acoustic communication support; underwater anti-submarine network construction is even more inseparable from underwater acoustic communication. In recent years, underwater preset weapons have also appeared, and wake-up and control are also required through underwater acoustic communication. At present, underwater acoustic communication has become a bottleneck restricting many underwater military applications.

Civilian Use

Ocean Mapping

Marine resource exploration, coastal engineering construction, and submarine engineering construction are all inseparable from the support of seabed topography and geology. The acoustic detection of seabed topography and geology is the fastest and most convenient technical means, which has been rapidly developed one of the important research areas of marine surveying and mapping. Equipment for marine mapping using sonar technology includes single-beam echo sounder, side-scan sonar, multi-beam sonar, synthetic aperture sonar, shallow profiler, etc., which can measure the seabed in different accuracy through various platforms.

Speed Measurement

Using the relationship of velocity to Doppler, the velocity can be estimated by measuring the

Doppler shift of the echo. The Doppler current profiler uses two sound beams at a certain angle to illuminate the water body. The suspended particles moving in the water body cause the Doppler effect of the water echo, which can be measured by measuring the Doppler shift between the two beams. It is estimated that the suspended particles are also the velocity of the water flow. When the sound beam is irradiated to the sea floor, the speed of the carrier relative to the seabed can be obtained by measuring the Doppler shift in the seafloor echo.

Underwater Positioning

When underwater submersibles work underwater, they generally need to know their relative positions in real time to achieve precise control. Acoustic methods are often used for underwater positioning, and the basic principle is triangular positioning. According to the needs of measuring distance and positioning accuracy, it is usually divided into long baseline positioning, short baseline positioning, and ultrashort baseline positioning, and the positioning accuracy can reach up to two thousandths. Ultrashort baseline positioning systems are typically installed on underwater unmanned platforms.

Marine Fisheries

Ocean fishing is a large mechanized fishing activity, so it is usually necessary to detect the fish population first and perform reasonable network operations based on the detection results. The fish finder is a sonar system that can be used to discover the movement, the location, and the range of the fish, which can greatly improve the production and efficiency of fishing. In addition, acoustic barriers can be used in marine farms to prevent shark intrusion and to prevent the escape of lobster fish.

Underwater Networking

With the increasing research activities of underwater production and the wide application of underwater unmanned platforms, the formation of underwater wireless networks in local areas will help to improve work efficiency and further realize intelligent management. For example,

a water-based router and a mobile node or a fixed node with a hydroacoustic communication module underwater can form an underwater Wi-Fi network, which can be applied to seabed observation network, offshore oil and gas platform data collection, underwater submersible cooperative networking operation, etc.

Influencing Factors

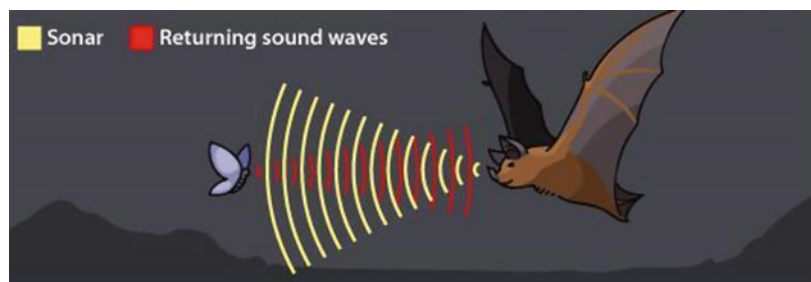
There are many factors affecting the performance of sonar work. In addition to the technical conditions of the sonar itself, the influence of external conditions is very serious. Some factors are more direct, such as propagation attenuation, multipath effect, reverberation interference, ocean noise, self-noise, target reflection characteristics, or radiated noise intensity, which are mostly related to marine environmental factors. For example, sound waves are affected by the uneven distribution of seawater medium and the influence of sea surface and sea floor during the propagation, which will cause refraction, scattering, reflection, and interference, which will cause sound line bending, signal fluctuations, and distortion, resulting in changes in the propagation path and appearance of sound shadow area, which seriously affects the range and measurement accuracy of the sonar. Factors affecting the performance of sonar work in addition to the technical conditions of the sonar itself, the influence of external conditions, are very serious. The more direct factors are propagation attenuation, multipath effect, reverberation interference, ocean noise, self-noise, target reflection characteristics, or radiated noise intensity, which are mostly related to marine environmental factors.

For example, sound waves are affected by the uneven distribution of seawater medium and the influence of sea surface and sea floor during the propagation, which will cause refraction, scattering, reflection, and interference, which will cause sound line bending, signal fluctuations, and distortion, resulting in changes in the propagation path and appearance. The sound shadow area seriously affects the range and measurement accuracy of the sonar. Modern sonar can choose the working depth and pitch angle of the array according to the sound velocity profile of the sea area and use the different propagation paths of sound waves (direct sound, sea bottom reflection, convergence area, deep sea channel) to overcome the adverse effects of underwater sound propagation conditions to improve the sonar detection distance.

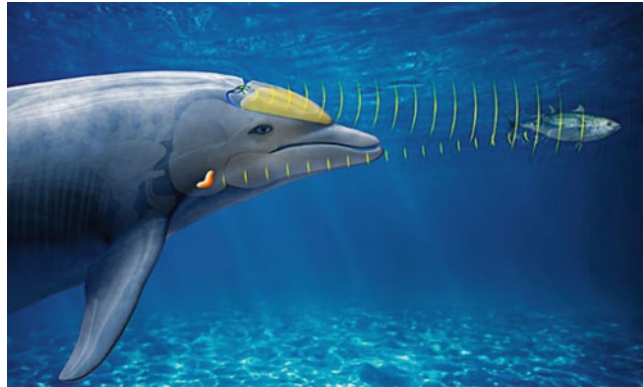
Biological Sonar

Interestingly, sonar is not a patent of human beings, and many animals have their own “sonar.” The bat uses the throat to emit 10–20 ultrasonic pulses per second and receives its echoes with the ear, as shown in Fig. 7. With this “active sonar,” it can detect very small insects and 0.1mm thick metallic line. Insects such as moths also have “passive sonar,” which can clearly hear bat ultrasound outside the 40m, and thus often escape the attack. However, some bats can use high-frequency ultrasound or low-frequency ultrasound beyond the insect detection range, so that the catch rate of catching insects is still high. It seems that animals also carry out “sonar warfare” like humans.

Sonar Technology,
Fig. 7 Bat sonar (<https://askabiologist.asu.edu/echolocation>)



Sonar Technology,
Fig. 8 Dolphin sonar
 (Sarfati 2019)



Marine mammals such as dolphins and whales have “underwater sonar,” as shown in Fig. 8, which use different sound signals for predation, play, social, and alarm, and other marine mammals such as seals and sea lions can also emit the sonar signal. Numerous studies have shown that the sensitivity of dolphin’s sonar is very high. They can identify a wire with a diameter of 0.2 mm and a nylon rope with a diameter of 1 mm, can find fish school in a few hundred meters away, and can move flexibly and quickly without touching the bamboo poles in the case of blindfolded eyes through the pool filled with bamboo rafts. The dolphin sonar’s “target recognition” ability is very strong; they not only can identify different fish, can distinguish different materials such as brass, aluminum, bakelite, plastic, etc. but also can distinguish the echoes of their own voices and the sound waves replayed by people recording their voices; moreover, dolphin sonar also has emotional expression ability, and it has been confirmed that dolphins are animals with “language” and their “conversation” is through its sonar system.

In addition, animals that have lived in the depths of the ocean for many years have severely degraded their vision and have to develop super-sensitive auditory systems for hunting prey and anti-avoidance attacks. Their sonar performance is beyond the reach of modern human technology. Solving the mystery of this animal sonar has always been an important research topic in modern sonar technology.

Terrible Problems Sonar Brings

A report by the National Natural Resources Defense Council (NRDC) shows that increasing ocean noise, such as military sonar, is affecting the lives of dolphins and whales because they must rely on sound for mating, foraging, and escaping natural enemies. According to the report, the low level of ocean noise affects the long-term behavior of marine organisms, which in turn leads to their hearing loss and even death. RDC’s research suggests that the current scientific community has no controversy about military sonar that can harm, kill, and extensively destroy marine mammals. The US Environment and the Cetacean Protection Organization have also worked for many years to protect marine mammals from the effects of US sonar. The results show that sonar is closely related to the mortality of cetaceans. In addition, sonar also reduced the success rate of halibut and other fish predation but also affected the fish reproduction rate and the behavior of giant turtles. The inner ear of some fish has also been seriously injured, which directly threatens their survival (Erbe et al. 2018).

Development Trend

Due to different uses, the development trend of military sonar and sonar for civil use are not the same. Military sonar pursues a longer range of action, so it presents a development trend of low frequency, high power, and high sensitivity, while

sonar for civil use pursues more precise measurement results. Therefore, the development trend will be high frequency and broadband. With the development of materials, electronics, information, and other technologies, the advancement of sonar technology will be further promoted in underwater array, signal processing, hardware implementation, and overall design method. The new generation of sonar will have higher sensitivity and farther detection distance and higher resolution.

Cross-References

- [Active Sonar](#)
- [Doppler Current Profiler](#)
- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)
- [Passive Sonar](#)

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Sound, Velocity, Temperature, Pressure (SVTP)

- [Underwater Information Sensing Technology](#)

Space-Time Trellis Codes (STTC)

- [Underwater Acoustic Communication](#)

SPAR Platform

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Synonyms

[MinDoc \(Minimum Deepwater Operating Concept\)](#); [SCF \(Single Column Floater\)](#); [SCR \(Steel Catenary Riser\)](#); [TCell \(Truss Cell Spar\)](#); [TTR \(Top Tensioned Riser\)](#)

Definition

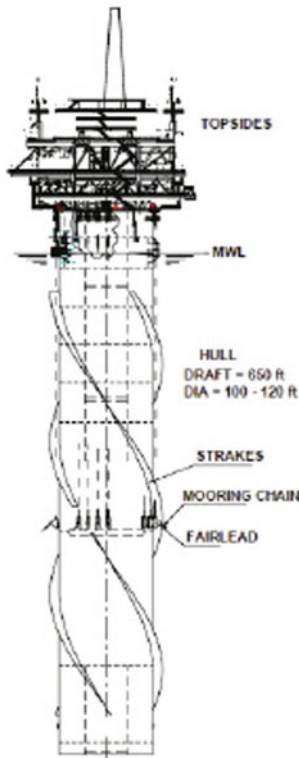
A spar is a type of floating oil platform typically used in very deep waters and is named for logs used as buoys in shipping that are moored in place vertically. Spar production platforms have been developed as an alternative to conventional platforms. The deep draft design of spars makes them less affected by wind, wave, and currents and allows for both dry tree and subsea production. Spars are most prevalent in the US Gulf of Mexico; however, there are also spars located offshore Malaysia and Norway. ([https://en.wikipedia.org/wiki/Spar_\(platform\)](https://en.wikipedia.org/wiki/Spar_(platform))).

Spar Platform Types

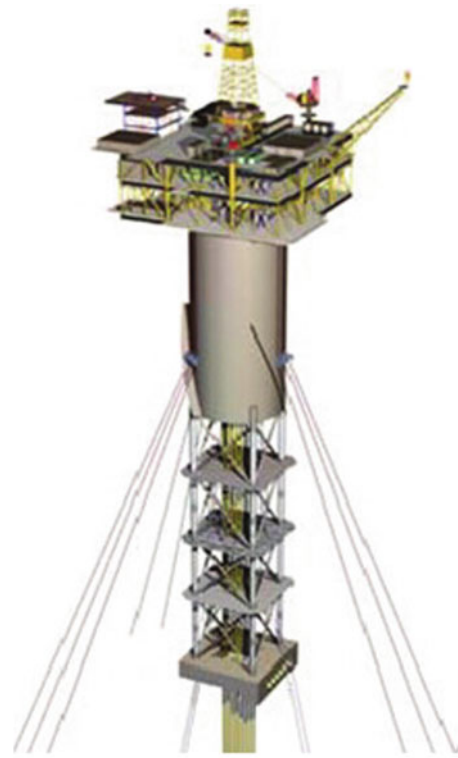
Spar platforms currently in service and under construction in the world can be divided into three types, which are Classic Spar, Truss Spar, and Cell Spar.

Classic Spar – First Generation

The main body of the Classic Spar is a cylinder suspended vertically in the water. The overall diameter is large. The length is generally between 20 m and 40 m. The main draught is above 100 m, and the center of gravity is located deep below the waterline. Its bulky interior is divided into a multi-story and multicabin structure using vertical water-blocking bulkheads and horizontal decks. It consists mainly of three parts: hard tank,



SPAR Platform, Fig. 1 Classic Spar



SPAR Platform, Fig. 2 Truss Spar

midsection, and soft tank. The hard tank is located at the top of the main body and is the main source of buoyancy for the entire Spar platform. The middle section is the part from the bottom of the variable ballast tank to the top deck of the temporary float. Its main function is to rigidly connect the hard and soft cabins of the main body of the Spar platform. The soft cabin is located below the middle section and mainly provides ballast (Fig. 1).

Truss Spar – Second Generation

The design concept is to replace the middle structure of the Classic Spar cylinder with a truss structure. It uses an open space frame structure to connect the top hard and bottom soft compartments, which can save about 50% of the steel; and the central open truss structure makes the area of the Truss Spar main body greatly reduced, which lowers the horizontal external load of the platform, thereby reduces the horizontal response of the platform and the drag load caused by the

circulation. The truss structure also provides lateral support for TTR and SCR. There are normally three heave plates installed evenly in the truss, and the application of the heave plate can significantly reduce the vertical motion amplitude of the platform. Compared with Classic Spar, Truss Spar has better response performance and greatly reduces the amount of steel used. Therefore, it has been widely used and becomes the most popular type of Spar platform (Fig. 2).

Because Truss Spar is the most popularly used type, its performance characteristics are introduced. Compared to other types of Spar platforms, the Truss Spar platform offers superior response performance and lower cost. This is mainly determined by its structural characteristics and reflects its superiority.

1. Due to the adoption of the open frame structure, it leads to the weight reduction. Meanwhile, the main surface area of the platform is greatly reduced, and the horizontal external

force load of the platform is reduced, thereby reducing the motion response of the platform in the horizontal plane.

2. The heave plate structure on the open body of Truss Spar platform greatly improves the stability of the platform. It not only can provide a certain ballast weight but also greatly lowers the heave motion amplitude, because the heave plates interact with the upper and lower seawater and produce a large viscous damping and added mass of heave motion. The natural frequency of the heave motion is further reduced, evading the likelihood of resonance in the wave frequency.
3. Due to the truss structure of the platform, the weight and external load are reduced, and a smaller-dimensional mooring system can be used.
4. Flexible subsea pipelines and steel catenary risers can be attached to the hard and soft tanks of Spar platform.

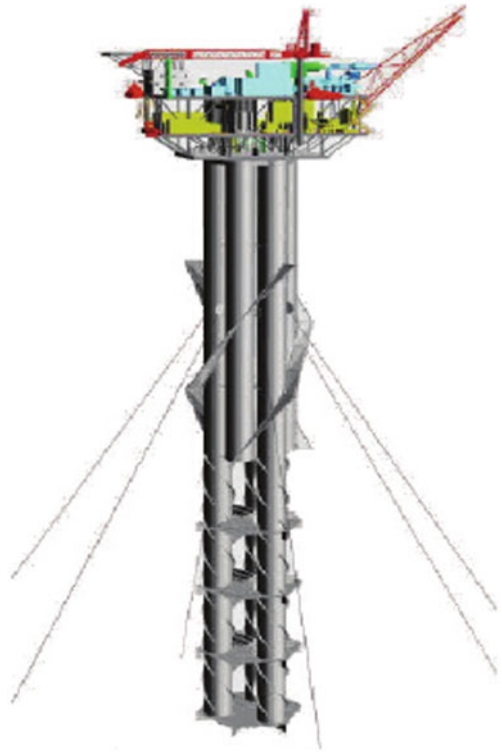
Cell Spar – Third Generation

There is only one Cell Spar in the world. Its hull is composed of several cylinders. The upper part of cylinders of the Cell Spar provides the overall required buoyancy. The lower part of the cylinders is formed by extending three of the outer cylinders to the bottom, and the cylinders are connected by several layers of heave plate structures, which meet the strength requirements of the main body and increase the added mass and damping. Ballast tank is also placed at the bottom of these cylinders to ensure adequate stability of the platform. The application of cylinders with smaller diameter makes it easy for construction, but also brings the challenge of welding heave plates in a uniform way (Fig. 3).

Other Concept of Spar Platforms

There are some other types of Spar platforms that have been proposed and constructed, and they are listed as below.

1. The Min Doc (Minimum Deepwater Operating Concept) platform was first used by the ATP Oil and Gas Development Company in the



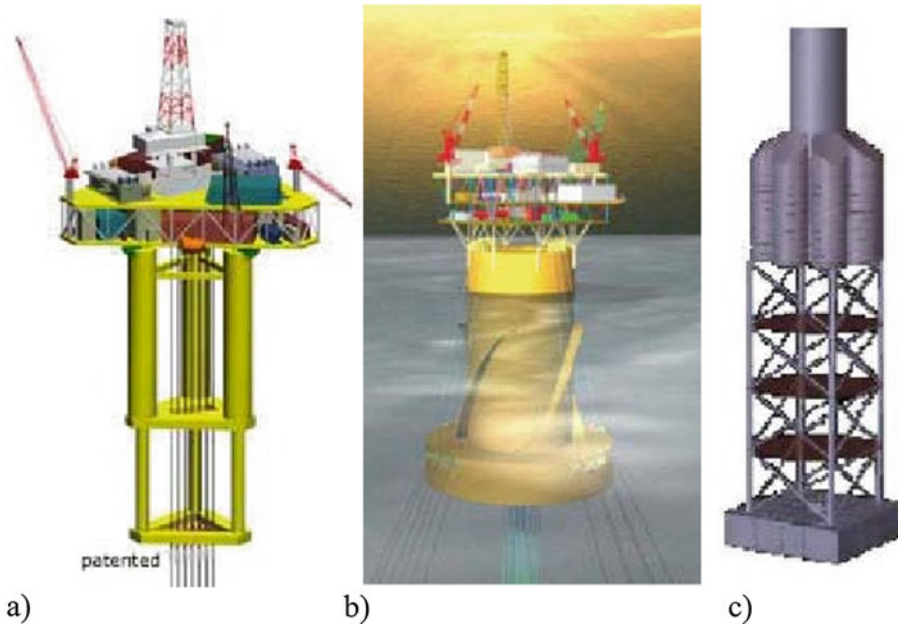
SPAR Platform, Fig. 3 Cell Spar

Gulf of Mexico in 2009 and installed in the depth of 1219 m in the Telemark Hub block.

2. The Single Column Floater (SCF) concept is similar to the Spar platform. Its structure can be divided into the upper module, the platform body, and the riser and mooring system, as shown in Fig. 4.
3. Combining the characteristics of the Truss Spar and multicolumn Spar platforms, the concept of the TCell Spar platform is also proposed.

Dry Tow Operation

Dry tow is an important offshore operation, especially for Spar and TLP platforms. Non-self-propelled offshore equipment such as drilling platforms, pipe-laying vessels, and living platforms are costing hundreds of millions of dollars, weighing 10,000 tons and extending as long as several hundred meters. These extremely large



SPAR Platform, Fig. 4 Other concepts of Spar platforms: (a) The MinDoc platform, (b) SCF, (c) TCell platform

pieces have to be towed from the land manufacturing base to the ocean or from one operation area to the other area for ocean transportation. Dry tow is a large-scale structure transportation method commonly used in offshore engineering field. However, due to the long towing time on the sea, the meteorological conditions, possible severe sea conditions, and technical conditions of the navigations, dry towing transportation is still relatively difficult and risky. Safe and reliable dry tow transportation requires not only reliable hard power support but also strong, mature support of soft power.

Generally speaking, long-distance transportation for large-scale offshore platforms such as Spar and TLP are adopting dry tow and wet tow both. Dry tow is to load the floating offshore platform on a semi-submersible ship and transport it. Figure 5 shows the dry tow process of a Spar platform on a semi-submersible ship.

During the dry tow operation process, engineers need to design two processes. One process is to tow the Spar or TLP platform onto the semi-submersible ship at the dock area, and the other process is separate TLP or Spar platform away

from the semi-submersible ship after they arrive at the operation sea site.

The process of towing Spar or TLP platform onto semi-submersible ship deck is a critical process. In the process of transferring Spar or TLP platform onto the semi-submersible ships, a good weather window is more conducive to execute operations, and it generally needs to consider various environmental conditions such as tidal variation, wind speed, and surge flow rate. Among these environmental factors, tidal variation is of the most importance, and it must be included in the design.

Upending Operation

As the spar platform becomes more widely used, the performance of the spar platform during transportation, construction, and installation needs to be noticed. When the SPAR platform is sent to the installation location, it needs to be righted, that is, the process of changing it from a horizontal status to a vertical status. The load on the motion and structure of the spar platform during the upending process is completely different from that of the

SPAR Platform,

Fig. 5 Dry tow of a spar.
(Picture from www.offshore-technology.com)



SPAR Platform, Fig. 6 The whole process of spar platform upending: (a) The starting moment (b) The middle moment (Picture from www.equinor.com). (c) The final moment. (Picture from www.statoil.com)

spar platform during its normal operation. The process of upending is achieved by injecting water into the ballast tank, which starts from the tank near the bottom of the spar platform.

The induced motions and loads during the upending process are of great importance to be checked. At the beginning of the upending, the slowly injected water broke the hydrostatic

moment balance of the horizontally floating spar platform on the water surface, and the bottom of the spar platform began to sink, forcing the spar platform to begin to slowly right up. The joint of the truss will bear a large total longitudinal bending moment, which is a severe test for the structural strength of the platform. Therefore, the main longitudinal bending moment of the joint between the hard chamber and the truss during the upending process needs to be focused on. In addition, the movement of the platform during the upending process also needs attention, such as the platform's six-DOF motions. The surge and sway motions have an important impact on the construction safety of the upending process. Meanwhile, the pitching motion of the platform is related to the success of the upending process. What's more, the platform's heave motion has a great influence on the stability of the platform during the upending process (Fig. 6).

Cross-References

- [Catenary Mooring](#)
- [Concrete Platform](#)
- [Mooring System](#)

Sparse Long Baseline (Sparse LBL)

- [Long Baseline Underwater Acoustic Location Technology](#)

Spatial Modulation (SM)

- [Underwater Acoustic Communication](#)

Spatial Variability

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Synonyms

[Inherent variability](#)

Definition

Spatial variability refers to the phenomenon that a quantity that is measured at different spatial locations exhibits values that differ across the locations.

Scientific Fundamentals

Sources and Engineering Influences of Spatial Variability

Natural materials (such as soil, rock, water, and air) often possess physical properties (such as density, strength, conductivity) that are associated with spatial variability. This is because the natural materials have been formed by a combination of various geologic, environmental, and physical-chemical processes that are themselves random processes in time and space. Many of these processes are continuing and can be modifying the natural materials in situ. As a result, the physical properties of natural materials display complex patterns of variation in space – in two or three dimensions – as well as variation with time. The variability may be small or large. There is a greater tendency for the properties to be similar in value at closely neighboring points than at widely spaced points because neighboring materials are more likely to have gone through similar natural processes during their formations. Spatial correlation length (or scale of fluctuation), θ , is

often used to quantify the correlation structure of spatially varying property. It provides an indication of the distance within which the property values show relatively strong correlation. A large θ implies that the properties fluctuate slowly with distance, while a small θ implies that the fluctuation is very rapid. The correlation length is usually direction-dependent. For example, soil properties usually have a smaller correlation length in vertical direction than in horizontal direction due to the deposition process which causes the soil properties to vary more significantly in vertical direction (Phoon and Kulhawy 1999).

Spatial variability of the properties of natural materials can pose significant influences on the evaluation of the performance and reliability of any engineering system built on, in, or with natural materials. Neglecting the spatial variability sometimes can lead to an unconservative estimation of the safety status of an engineering system. For example, Griffiths and Fenton (2004) found that the failure probability of a slope can be underestimated if the spatial variability of soil strength parameters is not considered. Therefore, the spatial variability needs to be rationally characterized and considered in the analysis of engineering systems for at least three purposes:

1. Characterizing the engineering system with spatially varying parameters
2. Predicting the response (behavior, performance, sustainability) of the engineering system
3. Providing basis for engineering decisions (in design, maintenance, or dynamic control)

Spatial variation of the natural material properties is essentially of a passive type. The material property at a given location is deterministic, but its value is unknown until accurately measured. Sampling at all locations is usually impractical and uneconomical, and measurement and testing errors tend to dilute the value of the information (Vanmarcke 1984). Therefore, the tasks of prediction, analysis, and decision-making must usually proceed on the basis of incomplete information about the spatially varying material properties,

and this fact requires a robust and effective technique to model the spatial variability.

Random Field Modeling of Spatial Variability

Spatial variability is usually decomposed into a smoothly varying trend $T(l)$ and a residual variation about the trend $R(l)$ as follows:

$$P(l) = T(l) + R(l) \quad (1)$$

where $P(l)$ is a spatially varying property and l is the location. The trend is modeled by a deterministic function obtained by regression analysis. The residuals are correlated and represent the inherent spatial variability. The most commonly used method to quantify the inherent variability is to model $R(l)$ by a homogeneous random field (Vanmarcke 1984). A random field is called homogeneous if (1) $R(l)$ shares the same mean and standard deviation across all locations and (2) the correlation among $R(l)$ at two different locations depends only on their separation distance rather than their absolute locations. These conditions are usually satisfied if the data are detrended and extracted from a homogeneous material. The correlation structure of a random field is usually described by a correlation matrix C as:

$$C_{ij} = \rho(\tau_{ij}) \quad (2)$$

where τ_{ij} is the separation distance between locations l_i and l_j and $\rho(\cdot)$ is the autocorrelation function denoting the correlation coefficient among $R(l)$ at two different locations separated by τ_{ij} . If there is enough field data, $\rho(\tau_{ij})$ can be estimated by a statistical analysis. However, in most cases, field data is limited. Therefore, it is more common to use the limited field data to fit a theoretical autocorrelation function. Table 1 summarizes the frequently used autocorrelation functions. The correlation structure is important for improving estimates at unsampled locations. Aside from the correlation structure, the mean (μ) and standard deviation (σ) of the inherent variability are also basic parameters of a random field model. If the

Spatial Variability, Table 1 Commonly used autocorrelation functions

Type	1-D autocorrelation function	Scale of fluctuation	2-D autocorrelation function
Triangular	$\rho(\tau) = \begin{cases} 1 - \tau/e & \text{for } \tau \leq e \\ 0 & \text{for } \tau > e \end{cases}$	$\theta = e$	$\rho(\tau_x, \tau_y) = \begin{cases} \left(1 - \frac{\tau_x}{\theta_h}\right)\left(1 - \frac{\tau_y}{\theta_v}\right) & \text{for } \tau_x \leq \theta_h \text{ and } \tau_y \leq \theta_v \\ 0 & \text{for others} \end{cases}$
Exponential	$\rho(\tau) = \exp(-\tau/a)$	$\theta = 2a$	$\rho(\tau_x, \tau_y) = \exp\left[-2\left(\frac{\tau_x}{\theta_h} + \frac{\tau_y}{\theta_v}\right)\right]$
Second-order autoregressive	$\rho(\tau) = \exp(-\tau/c)(1 + \tau/c)$	$\theta = 4c$	$\rho(\tau_x, \tau_y) = \exp\left[-4\left(\frac{\tau_x}{\theta_h} + \frac{\tau_y}{\theta_v}\right)\right]\left(1 + \frac{4\tau_x}{\theta_h}\right)\left(1 + \frac{4\tau_y}{\theta_v}\right)$
Gaussian	$\rho(\tau) = \exp[-(\tau/b)^2]$	$\theta = \sqrt{\pi}b$	$\rho(\tau_x, \tau_y) = \exp\left[-\pi\left(\frac{\tau_x^2}{\theta_h^2} + \frac{\tau_y^2}{\theta_v^2}\right)\right]$
Exponential-cosinoidal	$\rho(\tau) = \exp(-\tau/d) \cos(\tau/d)$	$\theta = d$	$\rho(\tau_x, \tau_y) = \exp\left[-\left(\frac{\tau_x}{\theta_h} + \frac{\tau_y}{\theta_v}\right)\right] \cos\left(\frac{\tau_x}{\theta_h}\right) \cos\left(\frac{\tau_y}{\theta_v}\right)$

Note: θ_h and θ_v are the scales of fluctuation in the horizontal and vertical directions, respectively; τ_x and τ_y are the horizontal and vertical distances between two locations, respectively

homogeneous conditions are satisfied, the mean of $R(I)$ is a constant value of zero because it is fluctuating equally about the trend. The standard deviation can be evaluated by the field data as:

$$\sigma = \sqrt{\frac{1}{n-1} \sum_{i=1}^n [R(I_i)]^2} \quad (3)$$

where n is the number of data points and $R(I_i)$ is the residual at location I_i .

With the mean, standard deviation, and the correlation structure, a random field can be generated to model the spatial variability. A variety of methods of generating random fields are available, including the local average subdivision method, covariance matrix decomposition, fast Fourier transformation, and the turning-band method. In the following, the covariance matrix decomposition method is introduced. Readers who are interested in other random field generation methods can refer to Fenton and Griffiths (2008). The covariance matrix decomposition method involves three steps:

1. Decompose correlation matrix C into the product of a lower triangular matrix and its transpose by Cholesky decomposition:

$$C = LL^T \quad (4)$$

2. Generate a standard Gaussian random field G :

$$G = LU \quad (5)$$

where U is a vector of independent standard normal random variables.

3. Transform the standard Gaussian random field into the target random field based on the mean, standard deviation, and the probability distribution of the target field. For instance, if the target random field $R(I)$ follows a lognormal distribution, it can be obtained by:

$$R(I) = \exp(\lambda + \xi G) \quad (6)$$

$$\xi = \sqrt{\ln[1 + (\sigma/\mu)^2]} \quad (7)$$

$$\lambda = \ln(\mu) - 0.5\xi^2 \quad (8)$$

Figure 1 shows a random field realization of the spatially varying strength parameter (q_c) of a two-layer soil (Li et al. 2018). The spatial variability is well characterized by the random field.

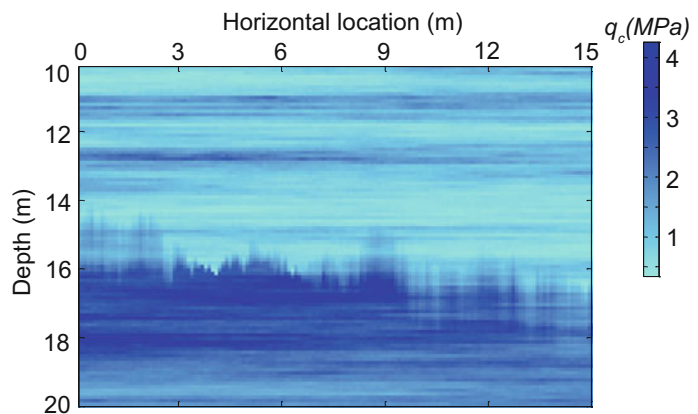
To evaluate the impact of spatial variability on the responses of an engineering system, random field simulations are usually integrated within a Monte Carlo simulation framework in a probabilistic manner. Let F denote the response of a system, e.g., factor of safety, deformation, etc. F is a function of $R(I)$:

$$F_i = f(R_i(I)) \quad (9)$$

where $R_i(I)$ is the i th realization of the random field and F_i is the value of F in the i th simulation.

Spatial Variability,

Fig. 1 A random field realization of the spatially varying strength parameter (q_c) of a two-layer soil (Li et al. 2018)



By generating a large number of random field realizations and repeatedly calling Eq. (9), one can obtain sufficient data of the response. Afterward, statistical analysis can be conducted to evaluate the quantities of the response such as the mean, standard deviation, and the probability distribution type, which can be used to assess the response of the system considering the spatial variability of the parameters.

Key Applications

Characterization of Spatial Variability

To characterize the spatial variability of a seabed soil, the trend, autocorrelation function, and scale of fluctuation should be investigated. Baecher and Christian (2003) presented the methods to characterize spatial variability. Generally the characterization consists of four steps (Lacasse et al. 2014): (a) visual inspection of data by soil unit and preliminary second-moment data analysis, (b) identification of a deterministic trend function and decomposition, (c) identification of a suitable uncertainty model, and (d) quantification of the uncertainty (mean, variance, standard deviation, scale of fluctuation).

The spatial variability of seabed soils has been reported for the North Sea, the Gulf of Mexico, offshore Australia, etc. It is found that the undrained shear strength (s_u) of clay followed a normal or lognormal distribution with its

coefficient of variation (COV) ranging between 5% and 35%. Uzielli et al. (2006) analyzed the cone resistance of Troll clay off the shore of Norway and reported that the undrained shear strength increased with depth. The mean value of the Troll clay increased from 12.7 kPa to 36.3 kPa with its standard deviation increasing from 2.0 kPa to 5.3 kPa. Bienen et al. (2011) found that the COV of offshore soils varied between 2% and 41% with a mean value of 23% based on data from 14 offshore clay sites. The spatial variation of the seabed soils was investigated using random field theory. The scale of fluctuation of the undrained shear strength and cone penetration resistance are reviewed and summarized in Table 2 (Li et al. 2016b). The vertical scale of fluctuation ranges from 0.05 m to 14 m. The scale of fluctuation in the horizontal direction varies over a large range from 7 m to 9000 m, which is much larger than that in the vertical direction.

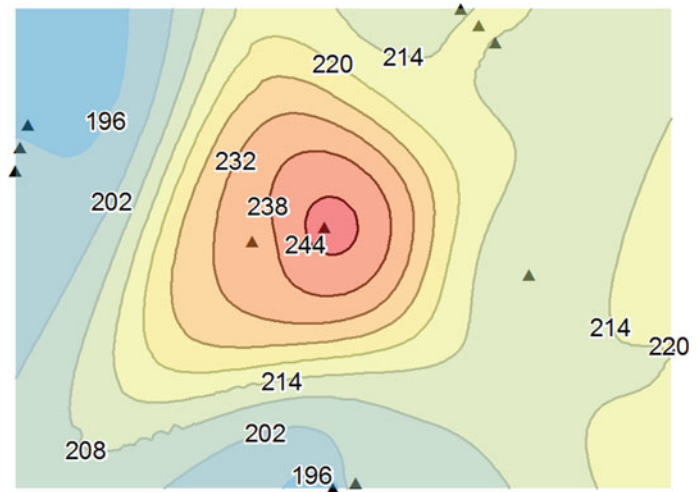
Properly accounting for the spatial variability in seabed soils may reduce substantially the uncertainties in soil parameters and the uncertainties associated with a design. Unfortunately, one is never able to gather enough subsurface data to get an exact picture of the variation of a soil property. One must therefore interpolate the soil properties within a large volume. A stochastic interpolation technique that can account for the spatial variability and minimize the variance is the kriging approach (Matheron 1963; Nadim 1988).

Spatial Variability, Table 2 Scales of fluctuation for offshore soils (Li et al. 2016b)

Soil	Location	Property	Scale of fluctuation (m)		Reference
			Horizontal	Vertical	
Offshore soils	North Sea	Cone penetration resistance	35.4–62	–	Tang (1979)
Sand	North Sea	Cone penetration resistance	26	0.4	Wu et al. (1987)
Offshore soils	–	Cone penetration resistance	24.6–66.5	–	Keaveny et al. (1989)
Offshore soils	–	Undrained shear strength	–	0.48–7.14	Keaveny et al. (1989)
Silty clay	North Sea	Cone penetration resistance	7–24	1.4–2.0	Lacasse and Lamballerie (1995)
Clay	Norwegian trench	Undrained shear strength	–	0.05–0.08	Uzielli et al. (2006)
Offshore soils	Gulf of Mexico	Undrained shear strength	9000	14	Cheon and Gilbert (2014)
Clay	Timor Sea	Cone penetration resistance	317	–	Li et al. (2015a)

Spatial Variability,

Fig. 2 Predicted net penetration resistance using kriging interpolation technique (units of kPa; Li et al. 2015a)



Li et al. (2015a) presented the spatial variability of seabed soils for the Laminaria site in the Timor Sea between Australia and Indonesia. There are only seven piezocones and six T-bar penetrometer tests in this site. A kriging method was presented to estimate the net penetration resistance at locations of interest, which further facilitated the evaluation of the spatial variability. Figure 2 shows the estimated values as a contour map of the net penetration resistance along with the measured data of the site. The predictions indicate that the net penetration resistance is larger in the central part of the site. This prediction provides enough data to evaluate the spatial variability of the seabed soils. A single exponential function was observed to best fit the autocorrelation data, and, subsequently, the horizontal scale of fluctuation for these data is estimated to be 317 m. The kriging interpolation technique was used to interpolate rock depth (Dasaka and Zhang 2012) and soil stratification (Li et al. 2016a).

Spudcan Response in Spatially Variable Seabed Soil

Mobile jack-up platforms are key contributors to the development of offshore oil and gas reservoirs in shallow to moderate water depths. A jack-up is self-installed by lifting the hull from the water and pushing the large spudcans (approximately 20 m in diameter) into the seabed. The influence of the spatially varied properties of seabed soils on the bearing capacity of the embedded spudcan

foundation is studied by Cassidy et al. (2013) and Li et al. (2015b). To simulate the spatial variability of the seabed soils, the undrained shear strength of clay was modeled as a random field. Random fields are mapped into a nonlinear finite-element analysis to reveal the failure mechanisms of the spudcan in spatially varied soils.

Random fields of undrained shear strength were generated using the local average subdivision algorithm. Finite-element analyses were performed using the nonlinear finite-element software ABAQUS 6.11. The soil domain was discretized into many zones onto which the random field of undrained shear strength was mapped. The soil properties vary from zone to zone to reflect the spatial variability of the soil. Monte Carlo simulations were performed for considerable realizations of the soil random field and the subsequent finite-element simulations. For each analysis, the bearing capacity of a spudcan foundation increases with increasing applied displacement until plateauing at the failure value, which is termed the ultimate bearing capacity. The results are presented in terms of the bearing capacity factor N_c for each realization:

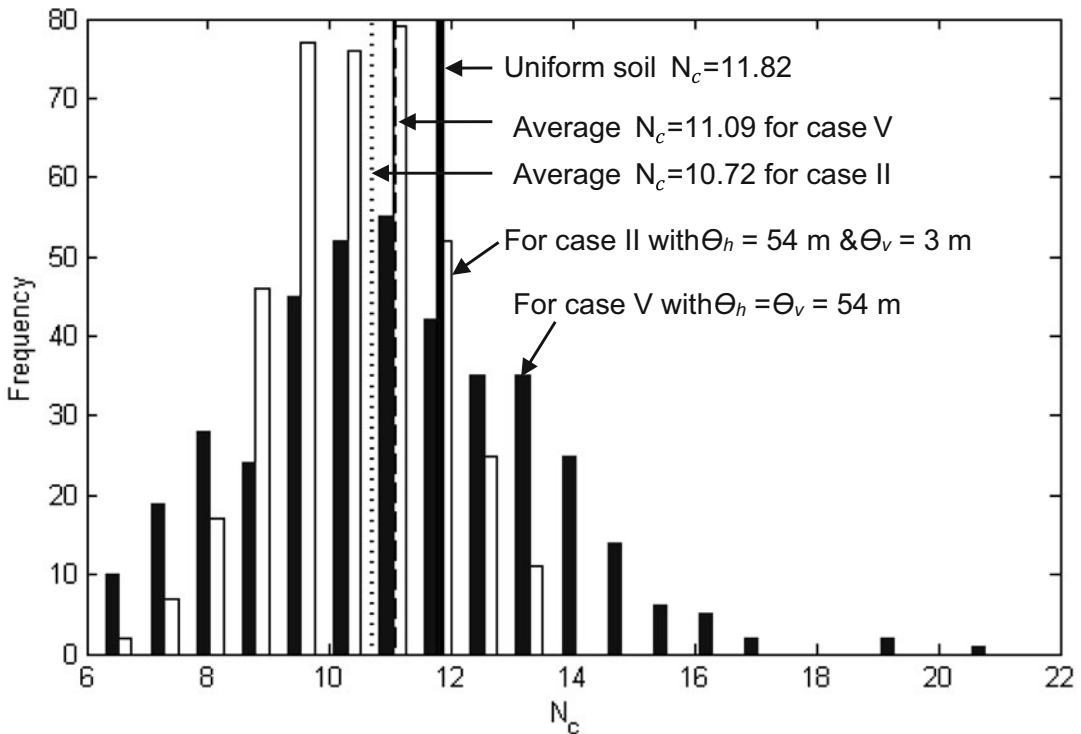
$$N_c \frac{q_i}{\mu_s} \quad (10)$$

where q_i is the bearing pressure computed for the i th realization in each case and μ_s is the mean value of the undrained shear strength.

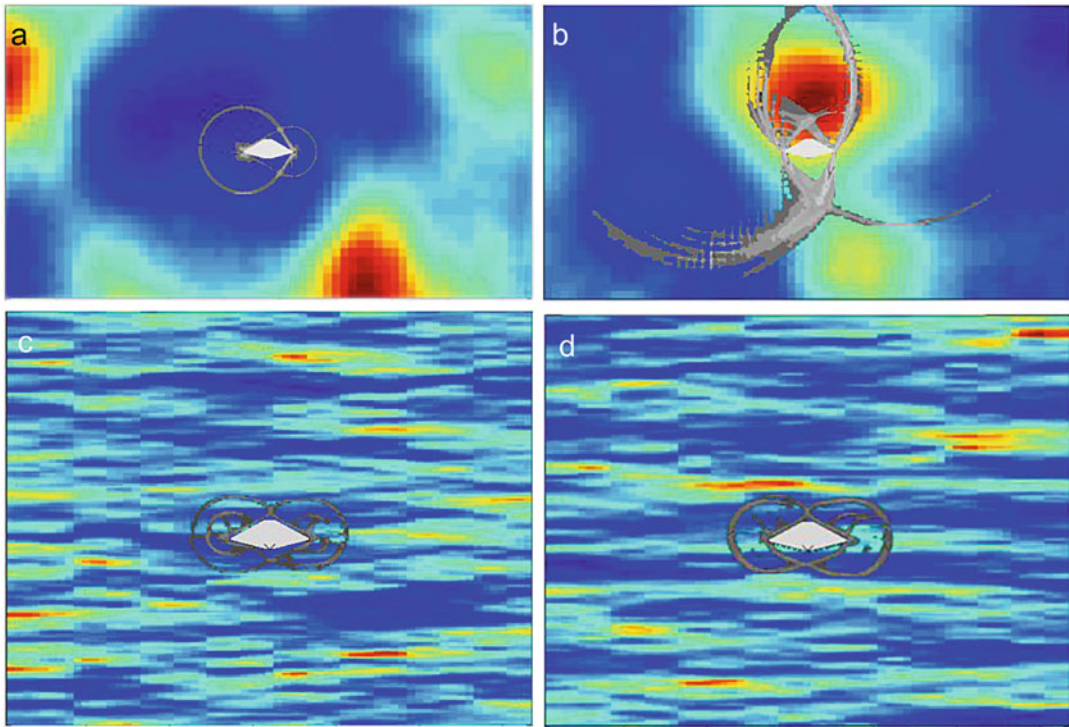
Results show that the ultimate bearing capacity of the spudcan varies significantly with the spatial variation of the seabed soils. Figure 3 shows the change in the bearing capacity factor for random fields with a horizontal scale of fluctuation of 54 m and a vertical scale of fluctuation of 3 m. The mean bearing capacity factor is 10.72, which is smaller than that of the uniform soil (e.g., 11.82). The bearing capacity factors in 93% of the random cases are smaller than that in the uniform soil. The worst cases occur when a localized failure path easily forms in the weak soils. When layers of strong soils around the foundation hinder the development of a continuous shear path, a relatively large bearing capacity can occur.

When the scale of fluctuation in the vertical direction is large, the correlation is strong for a relatively large volume of soils, as shown in Fig. 4. If a foundation is encompassed by a large volume of weak soils, a shear plane is developed locally in the weak soils (Fig. 3a).

Hence, the resistance is limited by the relatively small influential soil volume and the small undrained shear strength. In contrast, if a foundation is surrounded by a large volume of strong soils, the shear plane tends to develop in far-away weak soils, and a large volume of soils is thus mobilized, as shown in Fig. 3b. This long shear plane, combined with the large shear strength along the path, leads to a large bearing capacity in this realization. Therefore, the discrepancy in different realizations can be significant for cases with large scales of fluctuation. As the vertical scale of fluctuation decreases from 54 m to 3 m, the variation in the shear strength in the vertical direction becomes intensive, as shown in Fig. 3c, d. The shear planes pass through both the weak soils and strong soils in each realization, which result in similar bearing capacities. Hence the spatial variability of seabed soils significantly affects the bearing capacity of foundations.



Spatial Variability, Fig. 3 Distribution of the ultimate bearing capacity factor for the spudcan foundation



Spatial Variability, Fig. 4 Shear planes for selected realizations of Case II and Case V: (a) Realization No. 48 of Case V ($N_{cu} = 6.15$); (b) Realization

No. 400 of Case V ($N_{cu} = 21.15$); (c) Realization No. 355 of Case II ($N_{cu} = 9.79$); (d) Realization No. 201 of Case II ($N_{cu} = 10.74$) (Li et al. 2016b)

Cross-References

- [Bearing Capacity of Spudcans](#)
- [Pile Capacity](#)
- [Reliability Based Design \(RBD\)](#)

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Spatiotemporal Model

► Ship Operational Environment

SPDM – Self-Propelled Type Driving Mechanism

► Underwater Driving Subsystem

Special Marine Vehicle

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Definition

The term “special marine vehicle” here is intended to encompass all the classifications that are not

clearly included in other categories of this book. In other words, it is more like the “miscellaneous” of all the surface and underwater marine vehicles.

Scientific Fundamentals

Basic Type

Unmanned Surface Vehicle (USV)

Unmanned surface vessels (USVs) have also been called autonomous surface craft (ASC). As the name implies, they remove the operators from the platform and allow new modes of operations (Manley 2008) based on innovations in high-payload, high-speed small craft design and unmanned systems technology. USVs extend the success of autonomous underwater vehicle to surface vehicle, which could provide safer and cheaper choice to meet the various civil and military demands.

USVs are used by academic labs, corporations, and government users (Manley 2008). Typical applications include environment monitoring and survey, bathymetric mapping, object searching, general robotics research, etc.

Leisure and Sports Ships

The most common type of leisure ships worldwide is called “yacht.” There are two different classes of yachts: sailing and power boats. Sailing boat is the boat for which the primary means of propulsion is wind power. With the rise of the steamboat and other types of powerboat, sailing vessels in general came to be perceived as luxury or recreational vessels. Later the term came to encompass large motor boats for primarily private pleasure purposes as well.

Rowing boat, which is propelled by using the motion of oars in the water, displacing water, and moving the boat forward, is another common type of ships for recreational or sports purpose. There are two ways of connecting oars with ships, i.e., paddling and rowing. Rowing requires oars to have a mechanical connection with the boat, while paddles are handheld and have no mechanical connection.

Engineering Vessels/Ships

The concept of the engineering ship is usually defined as the ship that are equipped with various corresponding special equipment or engaged in various engineering and technical services for the requirements of different engineering operations. It has numerous types and complex equipments. And it has strong professionalism, which applies more new technologies and new equipments with different characteristics.

Icebreaker

An icebreaker is a special-purpose ship or boat designed to move and navigate through ice-covered waters and provide safe waterways for other boats and ships. For a ship to be considered an icebreaker, it requires three traits that most normal ships lack: a strengthened hull, an ice-clearing shape, and the power to push through sea ice. In the ice-covered region (e.g., the Baltic Sea, etc.), the main functions of icebreakers include escorting ships safely through ice-filled waters or freeing immobilized ship surrounded by ice. In difficult ice conditions, the icebreaker can also tow the weakest ships. As offshore drilling moves to the Arctic seas, icebreaking vessels are needed to supply cargo and equipment to the drilling sites and protect the drillships and oil platforms from ice by performing ice management. In the past, such operations were carried out primarily in North America, but today Arctic offshore drilling and oil production are also going on in various parts of the Russian Arctic (Riska 2011; Segercrantz 1989).

Rescue and Repair Ships

Rescue ships are a type of salvage vehicle. They are tasked with searching and rescuing lives and salvage property in wartime or in peacetime (Małyszko and Wielgosz 2016). Rescue ships are required to be robust enough to provide assistance in maritime accidents, such as combat salvage, rescue towing, firefighting, emergent repair, ocean towing, and heavy lift (Southworth 2008). The Navy designed the first seagoing “tug” the NAVAJO (ATF-64) class in 1939, to support logistic requirements for World War II.

A repair ship is a naval auxiliary ship designed to provide maintenance support to warships. Repair ships provide similar services to destroyer, submarine and seaplane tenders, or depot ships but may offer a broader range of repair capability including equipment and personnel for repair of more significant machinery failures or battle damage.

Key Applications

Basic Type

Unmanned Surface Vehicle (USV)

Military Uses USVs have the potential to reduce risk to manned forces; provide the necessary force to accomplish reconnaissance and patrolling missions (Bertram 2008; Caccia et al. 2007), search and rescue, and anti-terrorism/force protection (Pastore and Djapic 2010); perform tasks which manned vehicles cannot; and do so in a way that is affordable for the navy. Figure 1 shows an example of the weapon-loaded USV patrolling in Persian Gulf.

Environmental Sciences For the civil purposes, USVs are designed primarily for oceanographic surveys (Campbell et al. 2012), seafloor mapping, environmental monitoring, platform to predict natural disasters, and undertaking pollutant



Special Marine Vehicle, Fig. 1 An operational weapon-loaded USV deployed for maritime security and defense in the Persian Gulf (Bertram 2008)



Special Marine Vehicle, Fig. 2 Springer USV developed by Marine & Industrial Dynamic Analysis Research Group (MIDAS) at the University of Plymouth (Naem et al. 2006)

tracking and cleanup in rivers, reservoirs, inland waterways, and coastal waters, particularly where shallow waters prevail. Figure 2 shows a USV for environmental monitoring.

Commercial Exploitation USVs are developed to play a key role in future hydrocarbon exploration and exploitation, such as oil, gas, and mine explorations (Pastore and Djapic 2010), offshore platform/pipeline construction, and maintenance (Bertram 2008). The use of USVs can contribute to reduced personnel costs, improved personnel safety, widened weather window of operations, increased operational precision, and more environmentally friendly activities (Breivik et al. 2008). Figure 3 shows an industrial-level USV which is designed for target tracking.

Leisure and Sports Ships

Yacht The length of a yacht normally ranges from 10 m up to dozens of meters. A luxury craft smaller than 12 m is more commonly called a cabin cruiser or simply a cruiser. A superyacht generally refers to any yacht (sail or power) above 24 m, and a megayacht generally refers to any yacht over 50 m. The size of a yacht is small in relation to typical cruise liners and oil tankers (www.merriam-webster.com). Figures 4 and 5 show typical cruiser and megayachts, respectively.



Special Marine Vehicle, Fig. 3 An industrial-level USV developed by Rafael of Israel (<https://en.wikipedia.org>)



Special Marine Vehicle, Fig. 4 The “Lazzara” 80’ “Alchemist” on the California Coast (<https://commons.wikimedia.org>)

Windsurfing Windsurfing (as shown in Fig. 6) refers to the use of sail power to ride a rudderless, no-cabin boat to make a water sport. It consists of a board usually 2.5–3 m long, with displacements typically between 60 and 250 l. As a water sport project between sailing and surfing, the windsurfing consists of a plate with a steady plate, a gimballed mast, a sail, and a sail rod. Athletes use the natural wind blowing on the sails, stand on the board, operate the sails through the sail levers, and make the speed of the valve plate run on the water surface.

Dragon Boat As shown in Fig. 7, it is a long and slender rowing boat with a dragon’s head at its bows, used for racing with a mixed crew (Ayto and Crofton 2006). It has its roots in an ancient

Special Marine Vehicle,**Fig. 5** Motor yacht Samar

(<https://commons.wikimedia.org>)



Special Marine Vehicle, Fig. 6 Windsurfing (<http://www.toopen.com>)

folk ritual of contending villagers, which has been held for over 2000 years throughout southern China (https://en.wikipedia.org/wiki/Dragon_boat). It is also one of a family of traditional paddled long boats found throughout Asia, Africa, the Pacific Islands, and also Puerto Rico.

Engineering Vessels/Ships

Dredger Dredgers are responsible for digging waterways and river silt to facilitate the passage of other ships. And they can also be used for sand reclamation. At present, dredgers mainly include cutter suction dredgers (fighting bucket dredgers), trailing dredgers, chain bucket dredgers,

grab dredgers, and shovel dredgers (Qian 2008). Figure 8 shows the cutter suction dredger “Tian Jing Hao.”

Piling Vessel The piling vessel (the middle ship as shown in Fig. 9) is used for piling operations on the water. There is a piling frame at the end of the deck, which can bend forward and backward to meet the needs of the battered pile. The piling boat is a non-self-propelled ship and is towed in place by pushing (towing) wheels (near bow and stern of the dredger in Fig. 9). With the continuous enlargement of the scale of the port construction and the deepwater development of the offshore engineering, the performance requirements of the engineering operations on the piling vessel have also continuously increased.

Crane Vessel A crane vessel, crane ship, or floating crane is a ship with a crane specialized in lifting heavy loads. The crane vessels are generally used for offshore construction. Conventional monohulls are used, but the catamaran or semisubmersible-type crane vessels are often larger as they have increased stability. On a sheer leg crane, the crane is fixed and cannot rotate, and the vessel therefore is maneuvered to place loads (https://en.wikipedia.org/wiki/Crane_vessel). Figure 10 shows a typical crane vessel for offshore construction.

Special Marine Vehicle,
Fig. 7 Dragon boat racing
 between two teams (<http://ormskirkscouts.org.uk>)



Special Marine Vehicle,
Fig. 8 Tian Jing Hao
 dredger (<http://img1.voc.com.cn>)



Pipe-Laying Vessel Pipe-laying vessel is the main equipment for the development and construction of deepwater oil and gas fields and is also responsible for the installation of floating production platforms laying of submarine pipelines and installation of riser system. The main methods of pipe laying in deep water are S-lay, J-lay, reel lay and carousel lay, and vertical lay (Huang et al. 2011; Leffler et al. 2003; Gerwick 2007). Figure 11 shows a vertical lay-type vessel.

Icebreaker

Steam-Powered Icebreakers The first true modern seagoing icebreaker (Record 1951) was built at the turn of the twentieth century. Icebreaker Yermak (as shown in Fig. 12) was built in 1897 at the Armstrong Whitworth naval yards in England under contract from the Russian Navy. The ship borrowed the main principles from Pilot and applied them to the creation of the first polar icebreaker, which was able to run

Special Marine Vehicle,**Fig. 9** Piling vessel (<http://www.kinca.cn>)**Special Marine Vehicle,****Fig. 10** Crane vessel (<http://i.ce.cn>)

over and crush pack ice. The ship weighed 5000 tons, and its steam-reciprocating engines delivered 10,000 horsepower. The ship was so well built that it was only finally decommissioned and scrapped in 1963, making it one of the longest serving icebreakers in the world.

At the beginning of the twentieth century, several other countries began to operate purpose-built icebreakers. Most were coastal icebreakers, but

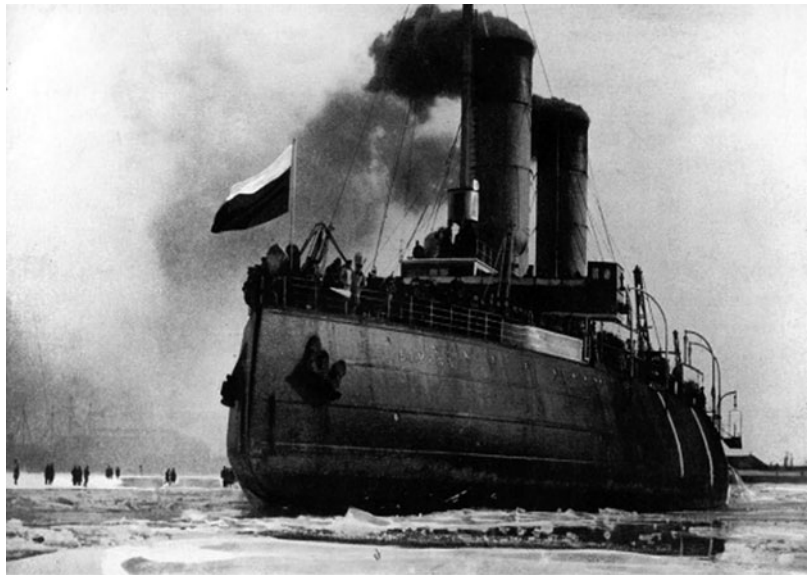
Canada and Russia also built several oceangoing icebreakers of around 10,000 t displacement.

Diesel-Powered Icebreakers Diesel-powered icebreakers are the mainstream form. The world's first diesel-electric icebreaker was the Swedish icebreaker Ymer in 1933, which was followed by the Finnish Sisu, the first diesel-electric icebreaker in Finland, in 1939 (Koehler and Oehlers

Special Marine Vehicle,
Fig. 11 Pipe-laying vessel
<http://www.cnss.com.cn>



Special Marine Vehicle,
Fig. 12 Yermak, the first
 modern polar icebreaker
<https://en.wikipedia.org>



1998). Both vessels were decommissioned in the 1970s and replaced by much larger icebreakers in both countries, the 1976-built Sisu in Finland and the 1977-built Ymer in Sweden.

In the United States, the need of research in Scandinavia and the Soviet Union led to the design of Wind-class icebreaker in 1941, which had a very strongly built short and wide hull, with a cut away forefoot and a rounded bottom. Powerful diesel-electric machinery drove two stern and one auxiliary bow propeller (Preston

1998; Silverstone 1970). These features would become the standard for postwar icebreakers until the 1980s. Figure 13 shows the USCGC Healy, which became the first unaccompanied US surface vessel to reach the North Pole in 2015.

In Canada, the first diesel-electric icebreaker HMCS Labrador started to be built in 1952. Then in 1969, Canada's largest and most powerful icebreaker, the 120-m CCGS Louis S. St-Laurent, was delivered. A multi-year mid-life refit project (1987–1993) saw the ship get a new bow and a

Special Marine Vehicle,
Fig. 13 USCGC Healy in
north of Alaska ([https://en.
wikipedia.org](https://en.wikipedia.org))



Special Marine Vehicle,
Fig. 14 Russian nuclear
icebreaker Arktika, the first
surface ship to reach the
North Pole ([https://en.
wikipedia.org](https://en.wikipedia.org))



new propulsion system. On August 22, 1994, Louis S. St-Laurent and USCGC Polar Sea became the first North American surface vessels to reach the North Pole.

Nuclear Icebreakers Russia currently operates all existing and functioning nuclear-powered icebreakers. The first one, NS Lenin, was launched in 1957 and entered operation in 1959, before being officially decommissioned

in 1989. It was both the world's first nuclear-powered surface ship and the first nuclear-powered civilian vessel.

The second Soviet nuclear icebreaker was NS Arktika (Fig. 14), the lead ship of the Arktika class. In service since 1975, she was the first surface ship to reach the North Pole, on August 17, 1977.

In May 2007, sea trials were completed for the nuclear-powered Russian icebreaker NS

50 Let Pobedy. The vessel was put into service by Murmansk Shipping Company, which manages all eight Russian state-owned nuclear icebreakers. The keel was originally laid in 1989 by Baltic Works of Leningrad (now St Petersburg), and the ship was launched in 1993 as the NS Ural. This icebreaker was intended to be the sixth and last of the Arktika class and currently is the world's largest icebreaker.

Rescue and Repair Ships

Deep-Submergence Rescue Vehicle A deep-submergence rescue vehicle (DSRV) is designed for immediate rescue operation in the incident for submarine mishap (Auxillia 2017). While DSRV is the term most often used by the US Navy, other nations have different designations for their vehicles. In a rescue mission, they are capable of searching for and rescuing life by themselves after transported to the submarine area by lifeboat or submarine ship. Figure 15 shows the submarine rescue diving recompression system.

Convoy Rescue Ship Convoy rescue ship (as shown in Fig. 16) is designated convoy rescue ships which accompanied some Atlantic convoys

to rescue survivors from ships that had been attacked. Rescue ships were typically small. They were usually stationed at the rear of the center column of the convoy.

Self-Propelled Semisubmersible Repair Ship Referring to the technology of floating ship, maintenance ship, and related engineering ship, a self-propelled semisubmersible repair ship (as shown in Fig. 17) is built on the basis of semisubmersible ship (Gou 2015). They are regarded as ship maintenance platform floating on the sea. Their task is transport of large goods, salvage, and reducing maintenance time.

Offshore Subsea Pipeline Maintenance Vehicle (SPMV) A SPMV is a type of special underwater vehicle which can navigate under the water, automatically detect and track the submarine pipeline and accurately locate the seabed, and carry out the inspection and maintenance of the pipeline (Li 2008). SPMV includes complete power machinery, propulsion, and manipulating equipments and is also equipped with full monitoring and maintenance equipments. They are tasked with survey of submarine pipeline, inspection of laid cable, and maintenance of submarine pipeline and laid cable.

Special Marine Vehicle,
Fig. 15 SRDRS, the submarine rescue diving recompression system (<https://www.sohu.com>)



Special Marine Vehicle,

Fig. 16 Rathlin 1599 t, built 1936, in rescue service from October 2, 1941, sailed with 47 convoys, including Convoy PQ 17, rescued 634 survivors (<https://en.wikipedia.org>)

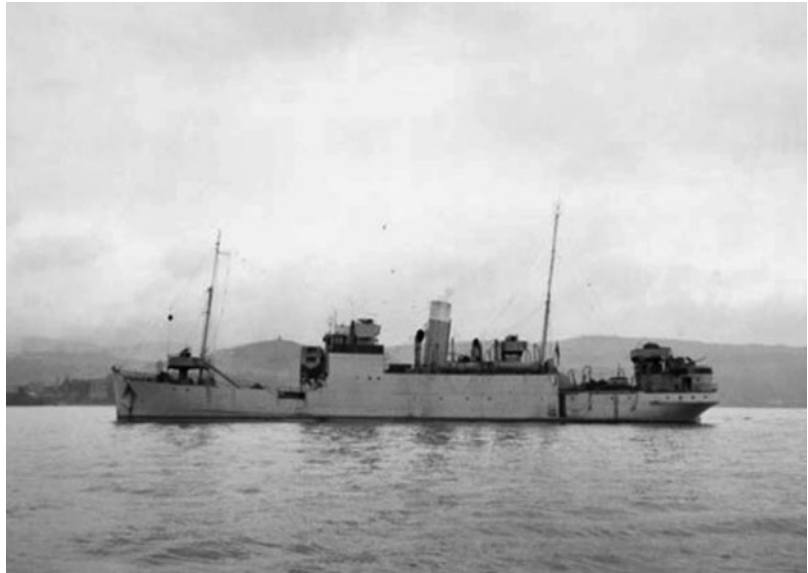
**Special Marine Vehicle,**

Fig. 17 Dockwise Vanguard – the largest semisubmersible vessel in the world (<https://www.fleetmon.com>)

**Cross-References**

- Fishing Ships
- Luxury Cruises
- Offshore Vessel
- Research Ship
- Service Ships
- Submarine
- Surface Warships
- Transport Ship

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SPI (The Soil Plugging Index)

► Offshore Pile Driving

SPM (Single-Point Mooring)

► FPSO

Spread Mooring

► Mooring System

Spread Mooring System

Yong Luo

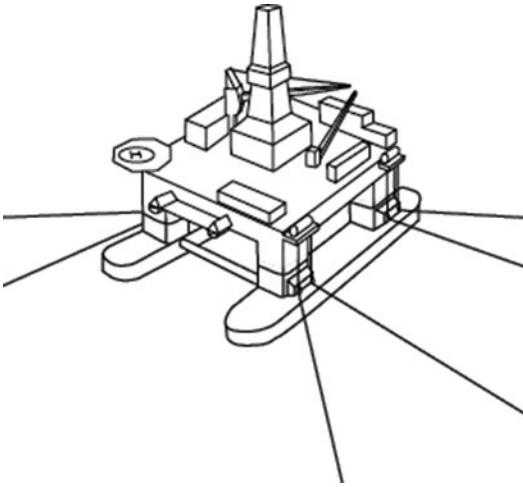
Shanghai Jiao Tong University, Shanghai, China

Synonyms

Multiple line moorings; Offshore mooring system; Single-point moorings; Station-keeping system

Definition

The passive mooring is the most commonly adapted station-keeping method for offshore floating facilities. An offshore mooring system usually consists of one or multiple mooring lines that connect the floater to the seabed to retain floater's position and limit offset to enable designated operations in various environments and water depths. There are two main categories of offshore mooring systems. The single point mooring system with mooring lines connected to the point of rotation of the floater such that it only restrain the floater position while allowing weather vanning. The spread mooring system utilizes multiple spread mooring lines connected to the multiple



Spread Mooring System, Fig. 1 Typical spread mooring system. (Courtesy of Vryhof)

locations of the floater to restrain its position as well as heading. For both types of mooring systems, the mooring line is usually made up of chain, wire rope, polyester, anchor, and connectors. In the spread mooring system, all mooring lines are pretensioned to form catenary or taut-leg profile and spread out in all directions (Fig. 1).

Historical Development

The mooring system can trace its origin to navigation. As ships sail around the world, the anchor mooring holds vessel's position relative to a point on the seabed of a waterway without connecting the vessel to shore. As such, mooring lines have been used for station-keeping of floating vessels in various situations. For example, they are used to berth boats or ships at quaysides, and often referred to as quayside moorings. In association with the development of offshore oil and gas exploration, the spread mooring system was further developed for station-keeping of various offshore platforms for operations of drilling, production, pipe laying, workover, etc.

Oil and gas have been produced offshore since the late 1940s. Bottom-founded structures such as fixed platforms were initially used for shallow-water operation. As exploration and production

moved to deeper waters and more distant offshore locations, the number of floating production systems increased, which utilized the spread mooring systems for station-keeping.

The first mobile offshore drilling unit (MODU) was the Mr. Charlie. It was a submersible barge built specifically to float on its lower hull for transportation to the location. It had to run a sequence of ballasting in order to rest on the bottom to begin drilling operations in 1954 (Veldman and Lagers 1997). It was rated for 40-ft water depth, even though strictly speaking, it was not floating during the drilling operation.

The first semisubmersible floating production platform was the Argyll FPF converted from the Transworld 58 drilling semisubmersible in 1975 for the Hamilton Brothers Argyll oil field in the North Sea. The SPAR platform was developed in later years as offshore production moves further into deeper waters. The first spar was the Neptune spar installed in 1997 in GOM. The first FPSO was the Shell Castellon, built in Spain in 1977 (Jay Schempf 2007).

Principles of Spread Mooring System

Mooring Functions

The functions of the mooring system can be summarized as:

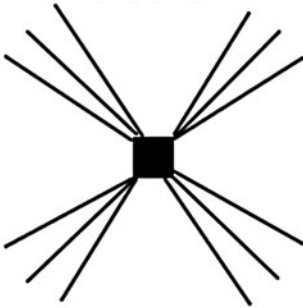
1. To maintain the floating structure on station
2. To have sufficient strength and fatigue life reserves to guarantee the safety and operability
3. To limit the excursion of the floater for the sake of risers and umbilicals, and maintaining adequate clearance with adjacent facilities

Mooring Spread

In a typical spread mooring system, the floating platform maintains the station-keeping by groups of mooring lines that connect the hull with anchors on the seabed. The mooring lines are spread out so that platform's lateral movement as well as the heading is restrained. The number of mooring lines and their lengths and dimensions depends on the environment and water depth at the location.

A typical spread mooring system can have a minimum of three mooring lines to as many as say 36 or even more lines. As a rule of thumb, the number of lines should be kept to a minimum as fewer lines; fairleads/winches mean less material costs and less installation time. However, as the number of lines reduces, the required line size for a given mooring system increases accordingly that would cause difficulty in line fabrication and installation.

The selection of mooring spread pattern should consider a number of factors such as system engineering, installation, directionality of environment, and mooring material cost. The equal spread design in which all mooring lines are spread symmetrical with equal spread angles has been used for ease of design and installation. Often the mooring lines are symmetrically grouped into three or four groups to improve the mooring system performance and create space for risers and other subsea facilities (Fig. 2).



Spread Mooring System, Fig. 2 Mooring line spread

Spread Mooring System, Fig. 3 Catenary mooring system. (Courtesy of Vryhof)

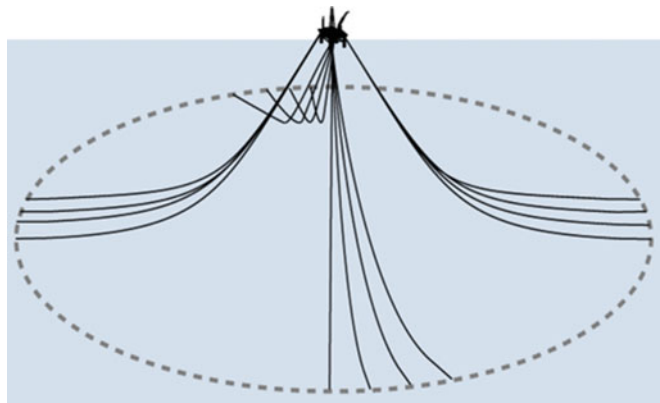
The mooring design typically has symmetrical design, i.e., all mooring lines are of the same composition and spread symmetrically with regards to direction. In some cases, the designer would take advantage of the directionality of the environment or the riser bundle to layout the mooring pattern in a nonsymmetric way. The use of a noneven spread or a nonuniform anchor radius can reduce the mooring system cost but may increase the engineering and installation efforts.

Mooring Line Configuration

According to the mooring line configuration, the mooring system can be further divided into **catenary mooring system** (Fig. 3) and **taut-leg mooring system** (Fig. 4).

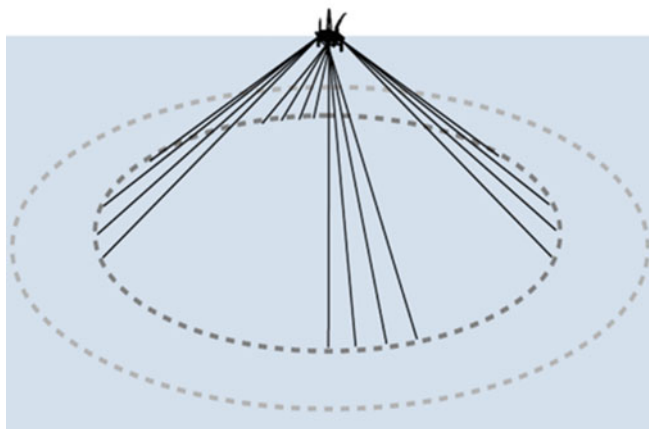
The catenary mooring system is the most commonly used system in shallow water. It gets its name from the shape of the free-hanging catenary line as its configuration changes due to vessel offset and motions. As shown in the figures below, the weight of the mooring material forms a catenary line shape of mooring legs, and during dynamic responses the catenary mooring system has a line profile that is part of the mooring line lying on the seabed in the static equilibrium position. The catenary mooring system uses more line material for the sake of compliance to floater offset and motion.

The taut-leg mooring system has no line lying on seabed in the static equilibrium position, and the mooring lines are naturally taut from anchor to seabed to the fairlead on the floater. The taut leg



Spread Mooring System,

Fig. 4 Taut-leg mooring system. (Courtesy of Vryhof)



system may use wire rope or polyester rope that is pretensioned until taut. The rope comes in at a 30–45° angle on the seabed where it meets the anchor (suction piles or vertically loaded anchors), which is loaded vertically. When the platform offsets horizontally with wind, waves and current, the lines stretch, and this generates the so-called restoring force.

The anchor footprint of such a mooring system is smaller and also the system uses less line material compared to the catenary mooring system. However, as the lines are taut, the compliance to floater offset and dynamic response is mostly from the line tensile stretch which is only possible in deep or ultra-deep waters where the mooring lines are sufficiently long.

The semitaut system combines taut lines and catenary lines in one system. The line configuration is usually catenary in the static equilibrium position and becomes taut when vessel offsets and the windward lines are stretched.

Working Principle

The magnitude and directionality of the prevailing environment of the wind, waves, and current are the driving parameters of mooring system configuration. They not only dictate the selection of mooring system type, but also the overall system configuration and size.

Under the environmental loading of wind, waves, and current, the floater offset from its mean equilibrium position where mooring forces of all spread mooring lines are balanced in the

horizontal plane. As the floater offset, larger tensions are being generated among the windward lines and tensions on leeward lines reduce (see Fig. 5 for illustration). The imbalance of mooring forces generates a so-called mooring restoring force that counterbalances the environmental forces. The larger the offset, the bigger the mooring restoring force until the acting environment force is balanced by the mooring restoring force. In this way, the offset of the floater is also kept within the design limit (Ma et al. 2019).

In addition to the balance of mean loads, there are also dynamic responses that are generated by the wave and wind actions. The mooring system is designed to accommodate these dynamic responses.

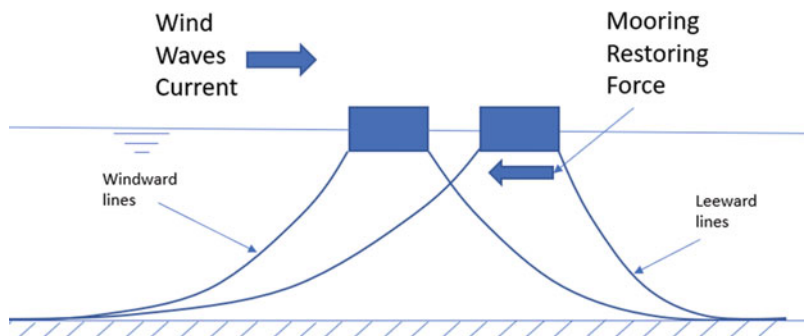
Mooring Components

As the mooring line is typically made up of synthetic fiber rope, wire, and chain or a combination of the three. In some cases, buoyancy modules as well as clump weights can be deployed along the mooring line to achieve specific objectives such as seabed clearance, more desired mooring performance, etc. Environmental factors – wind, waves, and currents – often drive the selection of mooring line materials.

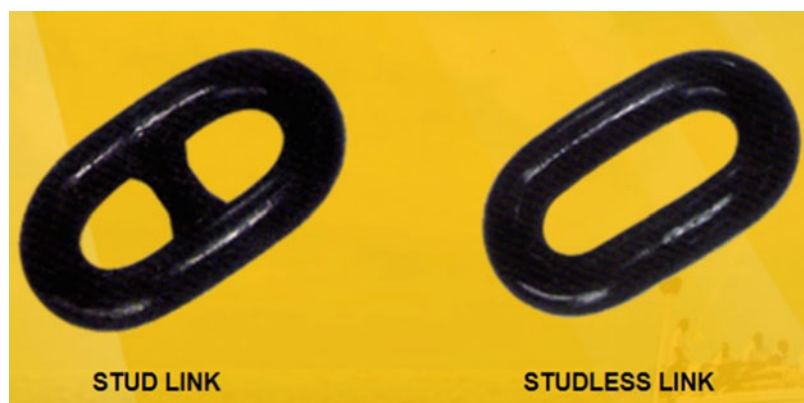
Chain

Chain is extensively used in all mooring systems; especially in shallow waters, the so-called

Spread Mooring System,
Fig. 5 Working principle
of spread mooring system



Spread Mooring System,
Fig. 6 Studlink and
studless chain. (Courtesy of
AsAc)



all-chain design is very popular. It is a simple and effective design taking advantage of the fact that chain is sturdy, has good resistance to seabed abrasion, is heavy to generate the desired catenary shape for compliance to vessel offset and motion, and provides added holding capacity to the anchor. To enhance the station-keeping performance, some of them fit clump weights on ground chain near the touch-down point. The additional weight can increase the restoring force of the mooring system, as the vessel would have to lift those weights before it can offset further.

In terms of chain construction, there are predominantly two types: stud link chain and studless chain (Fig. 6).

Wire

Wire rope is several strands of metal wire twisted into a helix forming a composite “rope,” in a pattern known as “laid rope.” Wire rope for offshore mooring purpose consists of multiple strands, or spiral strand that is round strands as

they have an assembly of layers of wires laid helically over a center with at least one layer of wires being laid in the opposite direction to that of the outer layer.

Because of its lighter weight in comparison to chain, wire rope offers a higher restoring force at the same given pretension because of the less-steep catenary shape of lighter mooring lines. The wire rope also has lower tensile stiffness in comparison to chain which would reduce the dynamic amplification of line tension. It is noted that for the same breaking strength, the wire rope also has cost advantage over the chain (Figs. 7 and 8).

For top and bottom segments of the mooring line, the chain is still preferred for handling and connection to vessel and anchor. Wire rope is widely used as the middle segment in a mooring line, making it a “chain-wire-chain” design. For mobile drilling unit moorings, the line configuration can simply be “wire-chain,” as wire ropes are typically deployed directly from winches on deck where there are no top chains.

Polyester

However, synthetic fiber rope has the lightest weight of all three. For vessels stationed in deep or ultra-deep water, the polyester rope has been increasingly favored over wire rope due to its much lighter weight and lower tensile stiffness. Polyester rope is not only highly competitive in cost, but also offers longer fatigue life than chain and wire rope. “Chain-polyester-chain” design has become a standard makeup for mooring systems in ultra-deep water. Apart from the lower unit weight, polyester also offers lower tensile stiffness which is highly beneficial to mooring

line dynamic tension in harsh environment (Fig. 9).

Anchor

The mooring system relies on the strength of the anchors. The holding capacity of anchors depends on anchor embedment depth and the soil properties. The mooring lines run from the vessel to the anchors on the seafloor. Anchor types include: drag embedment, suction, and vertical load.

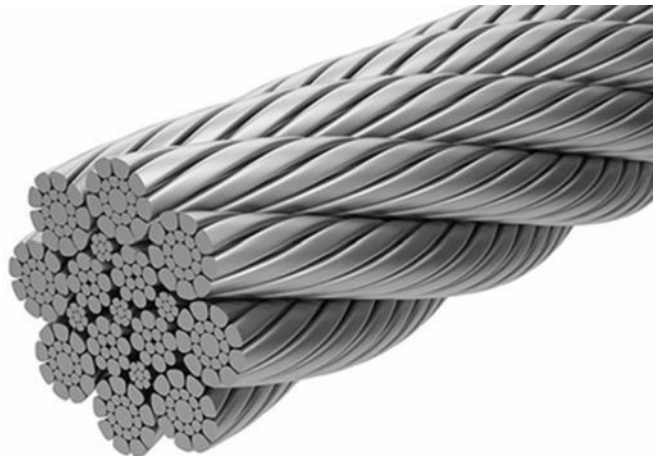
A drag embedment anchor (DEA) is the most utilized anchor for moorings. The drag anchor is dragged along the seabed until it reaches the required depth. As it penetrates the seabed, it uses soil resistance to hold the anchor in place. The drag embedment anchor is mainly used for catenary moorings, where the mooring line arrives on the seabed horizontally. It does not perform well under vertical forces.

Suction piles are the predominant mooring and foundation system used for deep water development projects worldwide. Tubular piles are driven into the seabed and a pump sucks out the water from the top of the tubular, which pulls the pile further into the seabed. Suction piles can be used in sand, clay, and mud soils, but not gravel, as water can flow through the ground during installation, making suction difficult. Once the pile is in position, the friction between the pile and the soil holds it in place. It can resist both vertical and horizontal forces.



Spread Mooring System, Fig. 7 Spiral strand wire rope. (Courtesy of SIRTEF)

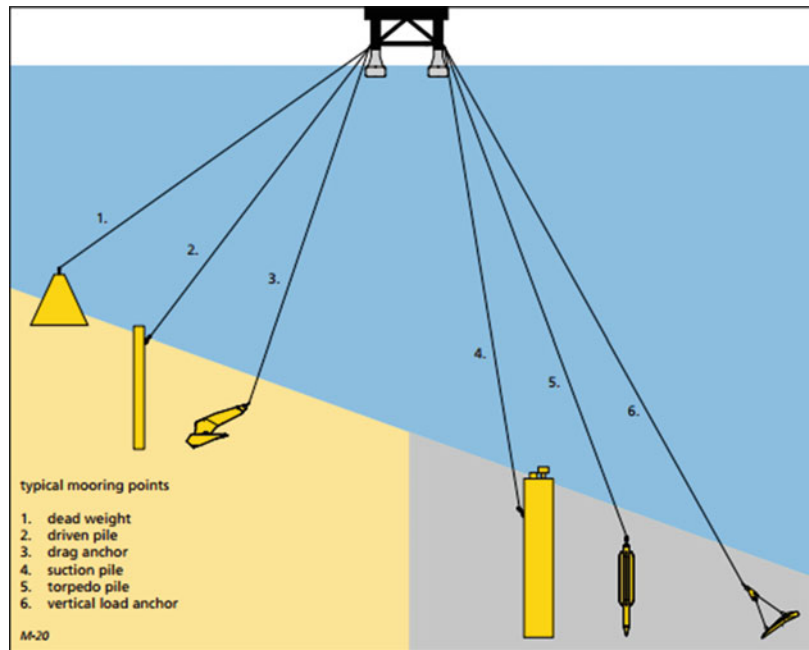
Spread Mooring System, Fig. 8 Multistrand wire rope. (Courtesy of Bridon)



Spread Mooring System,
Fig. 9 Polyester rope.
 (Courtesy of Lankhost)



Spread Mooring System,
Fig. 10 Various types of
 anchors. (Courtesy of
 Vryhof Anchor)



Vertical load anchors are similar to drag anchors as they are installed in the same way. However, the vertical load anchor can withstand both horizontal and vertical mooring forces. It is used primarily in taut-leg mooring systems, where the mooring line arrives at an angle the seabed (Fig. 10).

Connectors

As the mooring line is usually made of a number of segments with different materials or even the same material, connectors are used to connect the mooring line segments. There are different types

of connectors for different purposes. The commonly used ones include D-shackles for connection the chain segments, D-shackle in association of wire rope termination for connecting the chain to wire rope, and H-link for connecting the chain to polyester segment.

Buoy and Clump Weight

Sometimes, mooring legs are fitted with buoys and clump weights for various design objectives. The buoys can be deployed along the mooring line to lift the line from seabed to avoid say a seabed

pipeline, or near the surface to alter the line catenary shape to achieve improved restoring load characteristics.

Clump weights are typically deployed along the seabed touchdown area to improve the restoring load thus reducing floater offset.

Onboard Mooring Equipment

Mooring line connects to the floater via onboard mooring equipment which includes: fairlead, chainstopper, tensioner, and chain lock. The mooring line typically with chain as top segment goes through the fairlead or chain hawse to terminate at the chainstopper. The chainstopper is a device that allows the chain to be pulled through and locks the chain as soon as pulling in operation stops. Behind the chain stopper, the float is usually fitted with tensioners (typically on the deck) for the winch in or winch out of the mooring line. Chain locker is the chamber for the storage of excess chain.

For some MODUs, the top segment uses wire rope instead of chain. In such a case, the wire rope is pulled through the fairlead directly to onboard winch. The wire is stored on drums which can be rotated to winch in or winch out the wire rope.

Key Applications

Nowadays, the spread mooring system is widely used for offshore station-keeping of drilling and

production platforms, and vessels for designated offshore operations. According to the nature and duration of the offshore operation, the application of the spread mooring system can be divided into the following three categories (Barltrop 1998; API 2018):

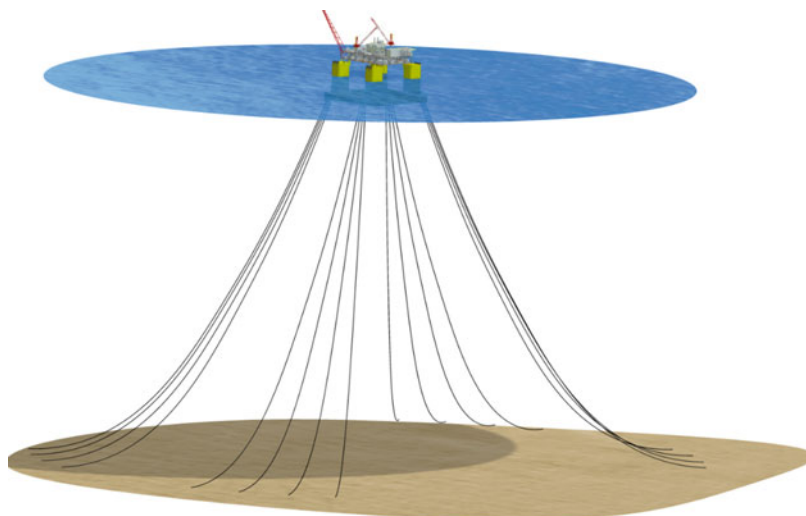
1. Mobile mooring system – Suitable for pipe laying vessels, crane vessels, logistics supply vessels, etc. When these vessels are in operation, they need to continuously move the anchors along predetermined route. The movement of the anchors is done by one or two auxiliary ships.
2. Temporary mooring system – Suitable for drilling platforms, drilling ships, drilling operations in certain areas for durations from several weeks to several months.
3. Permanent mooring system – Suitable for a variety of different types of floating structures. Depending on the design life of the oil field, it is necessary to maintain station-keeping in a fixed location for several years to several decades.

The majority of the mobile offshore drilling units and floating production systems use the spread mooring system for station-keeping purpose. For example:

1. A drilling semisubmersible typically uses 8 or 12 mooring lines in four groups connecting the four outward columns to the seabed.

Spread Mooring System,

Fig. 11 Spread mooring system for production semisubmersible. (Courtesy of COTEC)



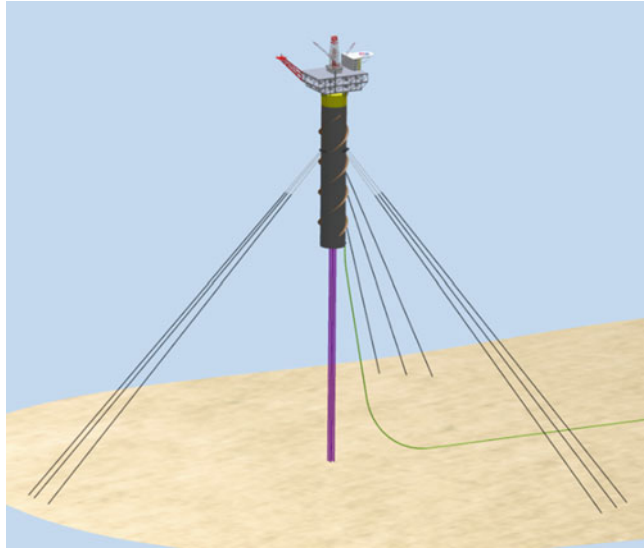
2. A drillship typically uses four groups of mooring lines connecting the bow and stern of the vessel to the seabed.
3. A production semisubmersible typically uses 8 or 16 mooring lines in four groups connecting the four outward columns to the seabed.
4. A production SPAR typically uses 9 or 12 mooring lines in three groups connecting the outer shell of the platform to the seabed (Figs. 11 and 12).

It is noted that FPSOs typically use single point moorings to allow the vessel to weather vane to minimize the environmental force, but for benign environment or for rounded shape FPSOs, the spread mooring system can be used.

There are other types of offshore systems that also use spread mooring for station-keeping, e.g., CALM offloading buoy, offshore wind turbine, etc. (Figs. 13 and 14).

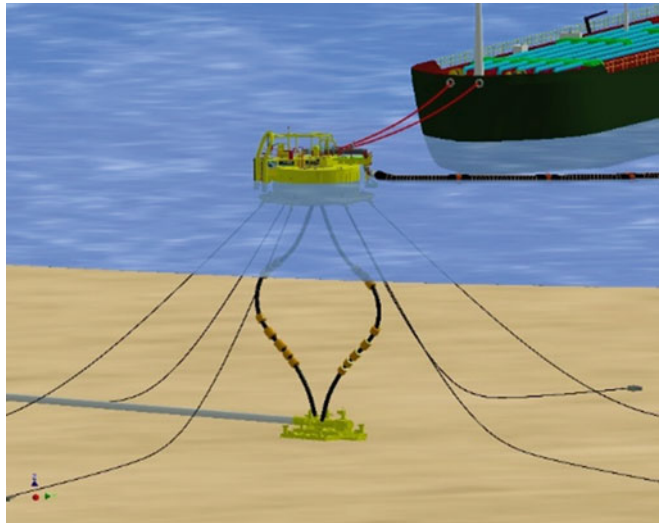
Spread Mooring System,

Fig. 12 Spread mooring system for SPAR platform.
(Courtesy of COTEC)



Spread Mooring System,

Fig. 13 Spread mooring system for CALM Buoy.
(Courtesy of COTEC)





Spread Mooring System, Fig. 14 Spread mooring system for offshore wind turbine. (Courtesy of COTEC)

Cross-References

- ▶ [Catenary Mooring](#)
- ▶ [Single-Point Mooring](#)
- ▶ [Taut Mooring](#)
- ▶ [Turret Mooring](#)

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Spread Moorings

- ▶ [Single-Point Mooring](#)

SPS, Subsea Production System

- ▶ [Moored Ship in Ice](#)

SRD(Soil Resistance During the Driving)

- ▶ [Offshore Pile Driving](#)

Stability at Large Angles

- ▶ [AUV/ROV/HOV Stability](#)

Stability in Beam Sea and Wind

- ▶ [Dead Ship Condition](#)

Stability of Damaged Ship

- ▶ [Damage Stability](#)

Stable Equilibrium

- ▶ [AUV/ROV/HOV Stability](#)

Static Analysis Method

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Synonyms

Catenary mooring lines; Taut mooring lines

Definition

Static analysis is a method to find the configuration and tension of mooring line under static equilibrium position. The static load means the forces not changed with time. The mooring line is supposed to be a completely flexible line without bending stiffness in the static analysis (American Petroleum Institute 2005).

Introduction

The catenary theory is usually used to make a static analysis. The classical catenary theory (Faltinsen 1993) and Garrett's rod theory (Garrett 1982) are both under the assumption of nonstretch in the axial direction. The lump-mass method (Korteraayama 1978) also is an effective method which translates the coordinates to assemble the global stiffness matrix. Most of the theories are limited to the assumptions that the axial deformation is small. It is convenient to perform an analysis using the above assumptions, because the chain catenary provides the restoring force due to gravitational effect.

The elastic deformation should be considered for the taut line made of synthetic fibers. The restoring force in the taut line is obtained by axial deformation, and the axial strain is relatively large compared with the chain line. The deformed configuration and reference configuration should be distinguished in the taut line when an analysis is performed. In the following, the elastic catenary theory is introduced.

Elastic Catenary Theory

The catenary method calculates the equilibrium position of the line, with ignoring the effects of bending and torsional stiffness of the line or its end terminations. This method also ignores contact forces between the line and any solid shapes in the model, but it could include all other effects, including weight, buoyancy, axial elasticity, current drag, and seabed touchdown and friction. The catenary position is often quite close to the true equilibrium position, especially when bend stiffness is not a major influence.

It simplifies the static analysis to neglect the effect of elasticity. However, in extreme conditions elasticity should be accounted for. The following demonstrate the elastic catenary theory to include the stretch of the mooring line.

An infinitesimal segment is used to create the equilibrium equations (Fig. 1). The original arc length is ds , and the weight force per length is w . When the cable is stretched, the arc length is described with $d\tilde{s}$, the weight force per length is expressed as $\tilde{w}$. The equilibrium equation is built in horizontal direction, then

$$dH = 0 \quad (1)$$

It means the horizontal force is constant on every point of the line.

Then the equilibrium equations are created in the tangent direction and the normal direction, we get

$$dT = w \sin \varphi ds \quad (2)$$

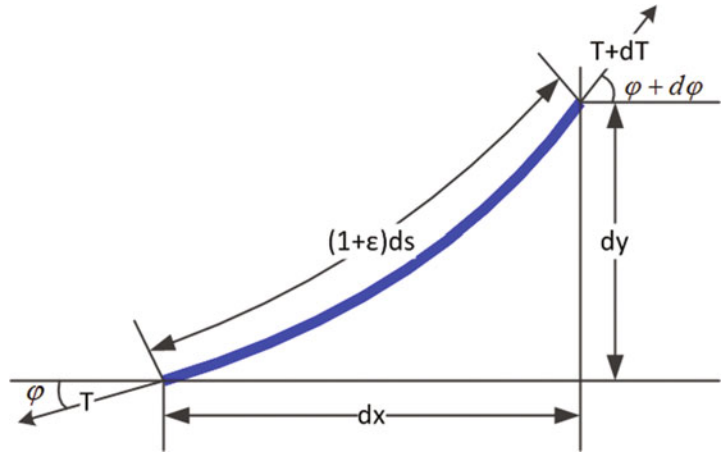
$$Td\varphi = w \cos \varphi ds \quad (3)$$

When the cable is deformed, the total mass is constant, so

$$wds = \tilde{w}d\tilde{s} \quad (4)$$

The equilibrium equations Eqs. (2), (3), and (4) become to be

$$dT = \tilde{w} \sin \varphi d\tilde{s} \quad (5)$$

Static Analysis Method,**Fig. 1** Infinitesimal segment of the extensible cable

$$Td\varphi = \tilde{w} \cos \varphi d\tilde{s} \quad (6)$$

From Eqs. (5) and (6), the following equation is obtained

$$\frac{dT}{T} = \tan \varphi d\varphi \quad (7)$$

The integration over the segment is performed on Eq. (7), each part of the integration is

$$\int_{T_0}^T \frac{1}{T} dT = \ln T - \ln T_0 \quad (8)$$

$$\int_{\varphi_0}^{\varphi} \tan \varphi d\varphi = -(\ln \cos \varphi - \ln \cos \varphi_0) \quad (9)$$

where T_0 is the tension force on the lower point, φ_0 is the angle between the tangent direction and the horizontal direction on the lower point. From Eqs. (7), (8), and (9), the relationship between the tension and the angle is

$$T = T_0 \frac{\cos \varphi_0}{\cos \varphi} \quad (10)$$

In order to get the horizontal and the vertical distances, the following geometrical relationships are used

$$d\tilde{s}^2 = dx^2 + dy^2 \quad (11)$$

$$dx = \cos \varphi d\tilde{s} \quad (12)$$

$$dy = \sin \varphi d\tilde{s} \quad (13)$$

At the same time, the relationship between the original and the deformed arc length is indicated by the Hooke's law

$$d\tilde{s} = \left(1 + \frac{T}{EA}\right) ds \quad (14)$$

Now the projection of the infinitesimal arc length in the horizontal direction is obtained by Eqs. (12) and (14)

$$dx = \cos \varphi \left(1 + \frac{T}{EA}\right) ds \quad (15)$$

Subtracting Eq. (3) from Eq. (15), we get

$$dx = \cos \varphi \left(1 + \frac{T}{EA}\right) \frac{T}{w \cos \varphi} d\varphi \quad (16)$$

With the help of Eq. (10), Eq. (16) becomes

$$\begin{aligned} dx &= \cos \varphi \left(1 + T_0 \frac{\cos \varphi_0}{\cos \varphi} \frac{1}{EA}\right) T_0 \frac{\cos \varphi_0}{\cos \varphi} \frac{1}{w \cos \varphi} d\varphi \\ &= \left(\frac{T_0 \cos \varphi_0}{\cos \varphi} \frac{1}{w} + \left(\frac{T_0 \cos \varphi_0}{\cos \varphi} \right)^2 \frac{1}{EA w} \right) d\varphi \\ &= \left(\frac{H_0}{w \cos \varphi} + \frac{H_0^2}{w EA \cos^2 \varphi} \right) d\varphi \end{aligned} \quad (17)$$

For the project of the infinitesimal arc length in the vertical direction, the same process is

performed with the help of Eqs. (3), (10), (13), and (14), we get

$$\begin{aligned}
 dy &= \sin \varphi d\tilde{s} \\
 &= \sin \varphi \left(1 + \frac{T}{EA} \right) ds \\
 &= \sin \varphi \left(1 + \frac{T}{EA} \right) \frac{T}{w \cos \varphi} d\varphi \\
 &= \sin \varphi \left(1 + T_0 \frac{\cos \varphi_0}{\cos \varphi} \frac{1}{EA} \right) T_0 \frac{\cos \varphi_0}{\cos \varphi} \frac{1}{w \cos \varphi} d\varphi \\
 &= \left(\frac{T_0 \cos \varphi_0 \sin \varphi}{w \cos^2 \varphi} + \sin \varphi \frac{(T_0 \cos \varphi_0)^2}{\cos^3 \varphi} \frac{1}{wEA} \right) d\varphi \\
 &= \left(\frac{H_0 \sin \varphi}{w \cos^2 \varphi} + \frac{H_0^2 \tan \varphi}{wEA \cos^2 \varphi} \right) d\varphi
 \end{aligned} \quad (18)$$

The horizontal distance is calculated with the integration over the cable on Eq. (17)

$$\int_{x_0}^x dx = \int_{\varphi_0}^{\varphi} \left(\frac{H_0}{w \cos \varphi} + \frac{H_0^2}{wEA \cos^2 \varphi} \right) d\varphi \quad (19)$$

$$\begin{aligned}
 x - x_0 &= \frac{H}{w} \left[\ln \left(\frac{1}{\cos \varphi} + \tan \varphi \right) \right. \\
 &\quad \left. - \ln \left(\frac{1}{\cos \varphi_0} + \tan \varphi_0 \right) \right] \\
 &\quad + \frac{H^2}{wEA} (\tan \varphi - \tan \varphi_0)
 \end{aligned} \quad (20)$$

Here the following equations are used in order to get a simplified formulation. Define

$$z = \ln \left(\frac{1}{\cos \varphi} + \tan \varphi \right) \quad (21)$$

then the following equation can be got easily

$$\begin{aligned}
 \sinh z &= \frac{e^y - e^{-y}}{2} \\
 &= \frac{\left(\frac{1}{\cos \varphi} + \tan \varphi \right) - \left(\frac{1}{\cos \varphi} - \tan \varphi \right)}{2} \\
 &= \tan \varphi
 \end{aligned} \quad (22)$$

Meanwhile, the relationship between the angle and the tension in the catenary line is

$$\tan \varphi = \frac{V}{H} \quad (23)$$

where V is the vertical force, H is the horizontal force.

From Eqs. (22), (23), we get

$$\sinh z = \frac{V}{H} \quad (24)$$

With the help of Eq. (1) and Eq. (24), Eq. (20) becomes

$$\begin{aligned}
 x - x_0 &= \frac{H}{w} \left[\operatorname{arsh} \frac{V}{H} - \operatorname{arsh} \frac{V_0}{H} \right] \\
 &\quad + \frac{H(V - V_0)}{wEA}
 \end{aligned} \quad (25)$$

The vertical distance is also calculated with the same process

$$\int_{y_0}^y dy = \int_{\varphi_0}^{\varphi} \left(\frac{H_0 \sin \varphi}{w \cos^2 \varphi} + \frac{H_0^2 \tan \varphi}{wEA \cos^2 \varphi} \right) d\varphi \quad (26)$$

$$\begin{aligned}
 y - y_0 &= \frac{H}{w} \left(\frac{1}{\cos \varphi} - \frac{1}{\cos \varphi_0} \right) + \frac{H^2}{2wEA} (\tan \varphi - \tan \varphi_0) \\
 &= \frac{H}{w} \left(\sqrt{1 + \left(\frac{V}{H} \right)^2} - \sqrt{1 + \left(\frac{V_0}{H} \right)^2} \right) + \frac{V^2 - V_0^2}{2wEA}
 \end{aligned} \quad (27)$$

Put the coordinate system on the lower point of the catenary line. If there is no vertical force on the lower end of the catenary line, the boundary conditions on the lower point are $x_0 = 0$, $y_0 = 0$, $\varphi_0 = 0$, $V_0 = 0$, the boundary conditions on the upper point is $V = wL$, Eq. (25) and Eq. (27) become to be

$$x = \frac{H}{w} \left(\operatorname{arsh} \frac{wL}{H} \right) + \frac{HL}{EA} \quad (28)$$

$$y = \frac{H}{w} \left(\sqrt{1 + \left(\frac{wL}{H} \right)^2} - 1 \right) + \frac{wL^2}{2EA} \quad (29)$$

So the dimensionless form of the extensible catenary theory is

$$\frac{x}{L} = \frac{H}{wL} \left(\operatorname{arsh} \frac{wL}{H} \right) + \frac{H}{EA} \quad (30)$$

$$\frac{y}{L} = \sqrt{1 + \left(\frac{H}{wL} \right)^2} - \frac{H}{wL} + \frac{wL}{2EA} \quad (31)$$

The catenary algorithm is robust and efficient for most realistic cases but it cannot handle cases where the mooring line is in compression. Since the bending stiffness and other effects are ignored in this method, thus, the compression means the mooring line is slack and no unique solution. The catenary method is iterative and may fail to converge. It may be possible to solve this by adjusting the catenary convergence parameters.

Cross-References

- ▶ [Catenary Mooring Lines](#)
- ▶ [Dynamic Analysis Method](#)
- ▶ [Mooring System](#)
- ▶ [Taut Mooring Lines](#)

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Station Keeping System

- ▶ [Mooring Anchor](#)
- ▶ [Mooring Connector](#)
- ▶ [Single-Point Mooring](#)

Station-Holding

- ▶ [Station-Keeping System for VLFS](#)

Station-Keeping System

- ▶ [Design of Mooring System](#)
- ▶ [Spread Mooring System](#)

Station-Keeping System for VLFS

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Synonyms

[Dynamic positioning system](#); [Hydroelasticity](#); [Mega-Float](#); [Mobile offshore base \(MOB\)](#); [Mooring system](#); [Nonuniform distribution of incident waves](#); [Position-holding](#); [Positioning](#); [Position-keeping](#); [Station-holding](#); [Uneven seabed](#)

Definition

The station-keeping system for very large floating structures (VLFS) is a group of interacting or interrelated entities that maintain VLFS in the desired position or keep these relatively static with respect to other floating structures in the presence of wind, waves, and currents. The station-keeping system for VLFS is commonly classified into two categories, namely the mooring system and the dynamic positioning (DP) system. The appropriate type of station-keeping system needs to be selected to create a balance among functionality, cost, and requirements of the VLFS, and this may be

achieved through selection, survey, research, design, verification, and development. The ultimate objective of the station-keeping system for VLFS is to improve the stability and safety of the structures, and fulfill functionality requirements and provide long-term services.

Fundamental Scientific Principles

As a permanent or semipermanent engineering project, VLFS have major characteristics that differ from typical offshore structures, such as long service life, reasonable maintenance expenses, and favorable fatigue resistance (Lamas-Pardo et al. 2015). Mooring and DP systems are two main methods used to keep VLFS in position under the effects of wind, waves, and currents.

Globally, VLFS are currently at the design stage, except for the Mega-Float in Tokyo Bay, the only manufactured VLFS in existence. Although VLFS projects are incomplete, these have inspired research on VLFS behavioral problems and station-keeping system design.

Mooring System for VLFS

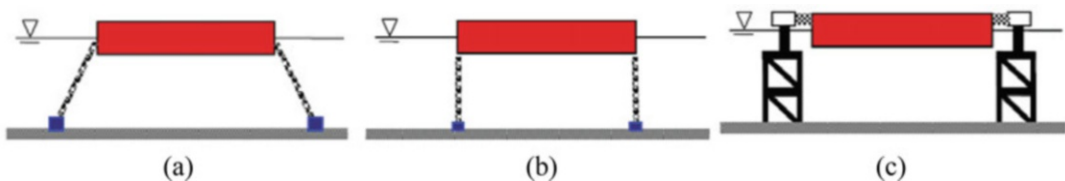
The mooring system is a widely used station-keeping technique for floating structures in moderate water depth, as used in VLFS projects such as Mega-Float in Japan (Shimada and Miyajima 2002; Sato and Inoue 2003). Mooring types for VLFS normally include mooring lines (e.g., cables and tension legs), rubber fender-dolphin, and pier wall (Fig. 1).

A mooring system, comprising several types of lines [e.g., catenary mooring line, taut mooring line, and tension leg mooring line (Journée and Massie 2001)], is a common station-keeping technique used for VLFS, in which restoring forces are

provided by the weight or elasticity of the mooring line. The mooring system for VLFS is functionally similar to that used for conventional offshore structures and has large-scale, special functionality owing to unique inherent features including (a) the high number and strong tension forces of mooring lines (few mooring lines cannot maintain station-keeping, and this leads to design difficulty and rising costs); (b) for modular VLFS, the mooring system should work with connectors mainly utilized to keep relative motions of adjacent modules as small as possible; (c) the elastic deformation of VLFS gives rise to large perturbations of certain mooring lines; and (d) asymmetrical mooring systems need to be adopted when VLFS are deployed near islands and reefs, where incident wave and bathymetry may be nonhomogeneous owing to the large scale of VLFS.

The description above implies that the design of mooring systems for VLFS is challenging due to the large dimension, complex characteristics of the sea surface, and uneven seabed conditions. Specific issues relating to the design of mooring systems for VLFS include the following:

- Nonuniform distribution of incident waves acting on different modules of VLFS leads to rugged motion responses along the VLFS, increasing variability in mooring line tensions.
- The effect of the elastic deformation may become obvious as the dimension of VLFS increases, resulting in a coupled response on mooring line tension.
- The complex seabed profile and significantly varying water depth always exist owing to the large dimension of VLFS; asymmetry mooring schemes attached with logically distributed buoys and clump weights may be utilized to achieve adequate mooring line shapes.



Station-Keeping System for VLFS, Fig. 1 Various types of mooring systems for VLFS: (a) cable/chain, (b) tension leg, and (c) rubber fender-dolphin (Wang and Tay 2011)

DP System for VLFS

In deep ocean situations, where VLFS such as mobile offshore bases (MOB) serve as floating airports, designing suitable mooring systems is challenging since the runway must be consistently aligned into the wind for planes to land with maximum control, and the cost of mooring lines increases substantially with the increasing water depth (Girard et al. 2003a). In such situations, the DP system is employed for station-keeping as this system has the advantages of rapid maneuvering and evacuation, favorable adaptation to environmental conditions, and sound station-keeping abilities regardless of water depth and seabed conditions (Fig. 2).

DP system development for VLFS is based on that for typical offshore structures and consists of four subsystems, namely the power system, thruster system, control system, and position reference system but with more stringent safety requirements. Inspired by emerging floating nuclear power plant concepts, the power system of DPS for VLFS can emulate this technology to supply sustainable power to the thrust system and other facilities. The thruster system is subjected to strict safety checks and fatigue analysis prior to installation to ensure sufficient thrust throughout the operational life of the VLFS. The control system must include manual autoposition modes that are integrated with assembly and emergency disassembly modes for MOB comprising multiple modules (Girard and Hedrick 2000). Essentially, the upper-level control law should be strictly examined with high robustness, fault tolerance, and risk avoidance, while the low-level control

law needs to distribute the commanded thrust force to all available thrusters of the VLFS in a manner that ensures optimization of energy, reduction of thrust force error, and adaptation to thruster failure/saturation. The position reference system is the “eye” of the DP system and needs to establish a credible system using Global Positioning Systems (GPS), underwater acoustic positioning systems, and laser positioning systems. An emergency situation in which all the positioning reference systems fail needs to be adequately addressed.

To avoid intensive hydroelastic loads of the VLFS, MOB dedicated to operating in the open sea commonly adopt modular design. Each module needs to be able to perform long-term station-keeping at sea. The modules can be assembled and maintain an absolute position in the global coordinate system and relative position to adjacent modules. Specific issues relating to DPS design for VLFS include the following:

- The DPS needs to maintain the overall position and orientation of all modules to be aligned with dominant wind, waves, and local currents in the predetermined operational site.
- Relative positions between modules should be properly constrained to avoid collisions and disjunctions, and suppress connector loads in a tolerant range to accomplish VLFS functionality.
- In emergency conditions, modules can be quickly disassembled and kept in the safe distance between each other to guarantee the safety of staff and equipment on the module. In harsher environments, modules can transfer to the nearest safe zones in an efficient and organized manner.
- The DPS can efficiently mobilize modules into the operational configuration.



Station-Keeping System for VLFS, Fig. 2 Scaled MOB module equipped with a dynamic positioning system (Girard et al. 2003a)

Key Applications

VLFS can be classified into two types based on geometry, i.e., the pontoon type and the semisubmersible type. Three projects, namely Mega-Float, MOB, and multimodule VLFS, are

presented to describe the applications of different station-keeping systems.

Station-Keeping System for Mega-Float

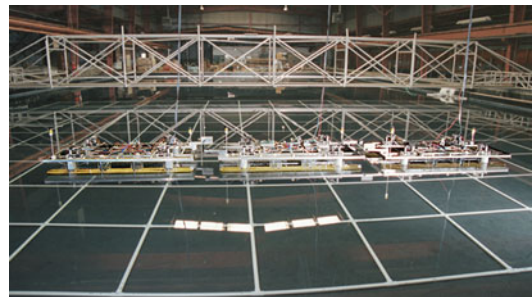
Mega-Float is a large pontoon-type VLFS designed for a floating airport deployed in protected waters (Isobe 1999). Figure 3 shows the design concept of Mega-Float utilizing the rubber fender-dolphin mooring system to restrain horizontal motions of the float. The fenders are attached to Mega-Float to have 0.1 m clearances between fenders and dolphins in the zero-load condition. Several studies have investigated the mooring feasibility of Mega-Float in various conditions such as slowly varying drift motion in irregular waves (Namba et al. 2000) and nonlinear mooring forces in the time domain (Shimada and Miyajima 2002). The sea environment in Tokyo Bay is relatively mild, and water is shallow. Kato et al. (2002) performed a quantitative risk analysis of multiple mooring dolphins for Mega-Float and found that the mooring design method was nearly appropriate if the yield strength of the dolphins was at least 2.5 times the design load.

Shimada and Miyajima (2002) conducted a sea scale model experiment of Mega-Float with a six-fender-dolphin mooring system. Restoring loads provided by deformation of the fender were measured and horizontal motion responses of Mega-Float analyzed during the sea tests. The rubber fender-dolphins could effectively restrain the motions of Mega-Float and had the lowest construction and maintenance costs.

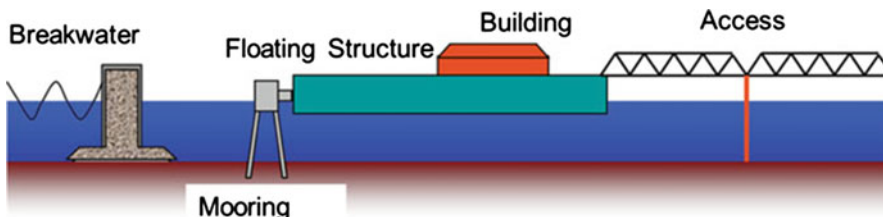
Station-Keeping System for MOB

In a deep ocean environment, Mega-Float may not be the best choice in terms of hydrodynamic

properties and costs of mooring systems. MOB may be more suitable due to the superior performances in waves (Palo 2005). MOB consists of several modules that are maneuvered into position and joined to form a single base. The US Navy supported a substantial research effort throughout the twentieth century to develop a suitable design for this concept. Mooring lines are not considered suitable for MOB, since the runway must be consistently aligned into the wind for planes to land with maximum control. At the beginning of the twenty-first century, a team from the University of California Berkeley developed an automated multimodule dynamic positioning (MMDP) control system for MOB and a simulation template for uniform support of DP control systems during tests and evaluations (Girard et al. 2003a). Rapidly forming or breaking connections between MOB components and reorientation of the entire MOB string in the direction of the wind can be achieved with the help of the control system (Girard and Hedrick 2000). Figure 4 illustrates the concept of the connected multimodule



Station-Keeping System for VLFS, Fig. 4 Simulation with three MOB modules in Berkeley's tank test (Girard et al. 2003b)



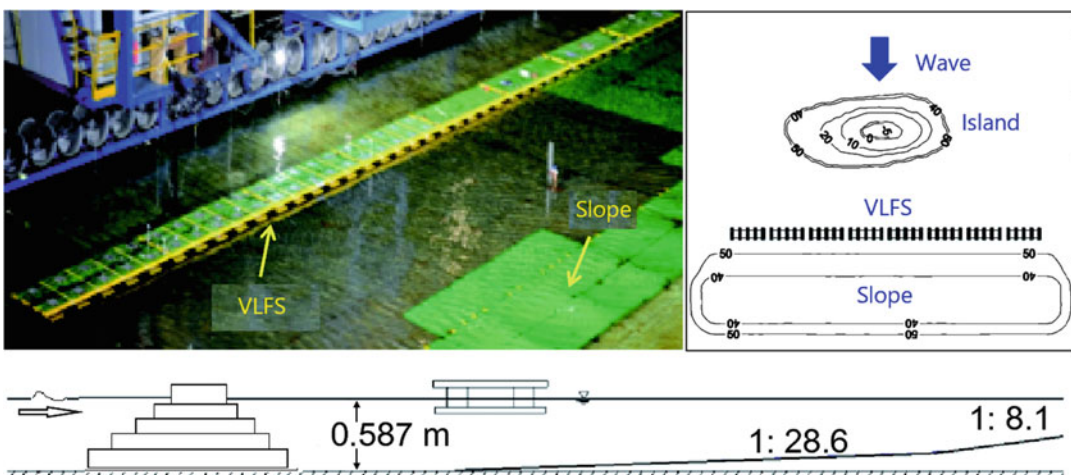
Station-Keeping System for VLFS, Fig. 3 Mega-Float with rubber fender-dolphin mooring system (Fujikubo and Suzuki 2015)

DP-derived MOB. During a 4-year program, several technical contributions were made to the MMDP: A complete system including the control, power, and thruster system was designed and developed based on a conventional PID controller; a corresponding model test was conducted to investigate and validate the control strategy of the MMDP; and a fundamental study was conducted to examine the mathematical model and real-time numerical simulations. The MOB program ended in 2000 since this was considered a less cost-effective solution than existing solutions such as a combination of aircraft carriers and support vessels for addressing logistical issues (Lamas-Pardo et al. 2015).

Station-Keeping System for Multimodule VLFS

In recent years, institutes and universities in China invested a great deal of effort in investigating the feasibility of multimodule VLFS near islands and reefs. Unlike MOB, mooring systems are employed with a comprehensive consideration of cost and safety requirements. Studies in China mainly focus on four aspects, i.e., hydroelasticity (Wu et al. 1993), effects of complex geographic environments on mooring systems (Cui et al. 2001), influences of nonuniform incident waves (Ding et al. 2017), and various types of mooring systems (Wang et al. 2017).

Based on the hydroelastic theory proposed by Wu (1984), a numerical method was established by Fu et al. (2007) to consider the coupled effects of the mooring system and hydroelasticity, and this was validated by experiments. Wu et al. (2017) and Ni et al. (2018) subsequently compared the mooring tensions between the flexible and rigid models of the module numerically and found that effects of the elastic deformation of the VLFS on the mooring tension can be significant in severe water conditions. A wave basin model test (Fig. 5) revealed that nonuniform distribution of the incident waves acting on different modules of VLFS leads to rugged motion configurations along the VLFS, increasing variety of the mooring line tensions (Wei et al. 2018; Ding et al. 2019). Apart from the coupled effects of hydroelasticity and mooring, various types of mooring systems have been investigated, including catenary mooring systems, taut mooring systems, and catenary-taut-tendon hybrid mooring systems (Wang et al. 2017, 2018). An optimization method for designing mooring systems for VLFS was proposed based on Genetic Algorithm and a vessel-mooring coupled model (Liang et al. 2019). Experimental evaluation of a mooring system simplification methodology for reducing mooring lines in a VLFS model testing was carried out (Liang et al. 2020). Results from a recent at-sea experiment of a two-module small-scale VLFS may be forthcoming.



Station-Keeping System for VLFS, Fig. 5 A wave basin test considering nonuniform incident waves, nonuniform seabed, multimodule coupling, and mooring system (Wu et al. 2018)

Future Challenges

In this entry, a summary of research and applications of station-keeping systems for VLFS emphasizing specific considerations that are far more complicated than those for conventional offshore structures is presented. Although significant progress has been made in the study of station-keeping systems for VLFS, to meet the challenge of analyzing and predicting complex ocean environment-induced motion responses, a great deal of work needs to be undertaken, particularly in the following areas:

1. The large dimension of VLFS alters the mooring system design input for traditional oil platforms, including nonuniform sea environment conditions and seabed conditions along the lengthy direction of the VLFS, elastic deformation of the VLFS, very large connector loads induced by asynchronous motion responses of adjacent modules, and mooring line deployment and dynamic response caused by seabed effects. Anchor-soil interactions can greatly challenge numerical modeling, prediction and experimental validation of the station-keeping ability of VLFS, and the mooring system design from an engineering perspective.
2. As a permanent station-keeping system for VLFS, mooring line tension, mooring chain ductile failure, and anchor-soil interactions need to be monitored in real time. Comprehensive methods combining statistic tools, probability theory, machine learning, and big data technology need to be developed to analyze monitoring results, predict operational risk over a short period of time and remaining lifetime of the mooring chain over a long period of time, and undertake temporary and permanent procedures to guarantee operational safety of the mooring system.
3. MOBs are likely to encounter more severe and complex environmental loads in the deep sea than VLFS near islands and reefs. High-level controller methods dedicated to tackling fast docking, emergency evacuation, and independent standby positioning should be developed in cases of thruster failure, DGPS signal failure, and connector failure. Local position reference systems with high reliability need to be developed to back up the global position reference system.
4. MOBs intended to serve in the deep sea are usually the semisubmersible form that can reduce wave loads. However, the relatively small water-plane area inherently leads to discernible coupling characteristics in the dynamics and kinematics between the horizontal and vertical planes. A positioning controller integrated with roll and pitch motion suppression needs to be developed to maintain the relative position and attitude of the modules, reducing the connector loads in all degrees of freedom (DoFs) and fulfilling the functionality of the VLFS.
5. The station-keeping system for VLFS needs to be easily operable, have abundant functionality, high robustness, and favorable compatibility, and be capable of integrating with artificial intelligence, green power, carbon dioxide emission reduction and have an environment-friendly interface to the ocean space and seabed.

Cross-References

- [Connectors of VLFS](#)
- [Dynamic Positioning System](#)
- [Hydroelasticity](#)
- [Mega-Float](#)
- [Mooring System](#)

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Steel Catenary Risers

► Pipeline Soil Interactions

Steel Pipelines and Risers

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Synonyms

Blowout preventer (BOP); External tieback connector (ETBC); Floating production storage and transportation systems (FPSO); Internal tieback connector (ITBC); Material takeoff (MTO); Tension leg platforms (TLP); Top tensioned risers (TTRs); Vortex-induced motions (VIMs); Vortex-induced vibrations (VIV)

Definition

Pipelines and risers are used for a number of purposes in the development of offshore hydrocarbon resources. These include the export (transportation) pipelines, flow lines to transfer product from a platform to export lines, water injection or chemical injection flow lines, Flow lines to transfer product between platforms, subsea manifolds and satellite wells, and pipeline bundles.

Design Stages

Submarine pipelines, steel catenary risers, and top tensioned risers are included in the book. The design of pipelines and risers is usually performed in three stages, namely:

1. Conceptual engineering

Conceptual engineering is normally:

- To establish technical feasibility and constraints on the system design and construction
- To eliminate nonviable options
- To identify the required information for the forthcoming design and construction
- To allow basic cost and scheduling exercises to be performed
- To identify interfaces with other systems planned or currently in existence

2. Preliminary engineering

The system concept is fixed for the primary design of the pipeline. This will include:

- To verify the sizing of the pipeline
- Determining the pipeline grade and wall thickness
- Verifying the pipeline against design and code requirements for installation, commissioning, and operation

3. Detail engineering

The detailed engineering phase is, as the description suggests, the development of the design to a point where the technical input for all procurement and construction tendering can be defined in sufficient detail.

The primary objectives can be summarized as follows:

- Route optimization
- Selection of wall thickness and coating

- Confirming code requirements on strength, vortex-induced vibrations (VIV), on-bottom stability, global buckling, and installation
- Confirming the design and/or performing additional design as defined in the preliminary engineering
- Development of the design and drawings in sufficient detail for the subsea scope. This may include pipelines, tie-ins, crossings, span corrections, risers, shore approaches, and subsea structures
- Preparing detailed alignment sheets based on most recent survey data
- Preparation of specifications, typically covering materials, cost applications, construction activities (i.e., pipelay, survey, welding, riser installations, spool piece installation, subsea tie-ins, subsea structure installation), and commissioning (i.e., flooding, pigging, hydrotest, cleaning, drying)
- Preparing material takeoff (MTO) and compiling necessary requisition information for the procurement of materials
- Preparing design data and other information required for the certification authorities

Design Process

The object of the design process for a pipeline is to determine, based on given operating parameters, the optimum pipeline size parameters.

Submarine Pipeline

Submarine pipelines have become important offshore oil and gas field production facilities and are the “artery” of offshore oil and gas production systems. Since the first submarine pipeline was laid in 1954, the length of submarine pipelines laid in the world has exceeded more than 100,000 km. Submarine pipelines are used for transporting three main substances: gas, oil, and water. Submarine gas and oil pipelines fall into three groups: intra-field pipelines, which are used to bring the oil or gas from wellheads to a point within the operating field for collection, processing, and onward transport; export

pipelines, which transport the gas and oil to land; and transport pipelines, which have no necessary connection with the operating field but transport gas or oil between two places on land. The last category is often included with the export pipelines.

Process Design

Submarine pipeline process design: Determine the submarine pipeline transportation process scheme according to the basic data including oil and gas field development scale, physical properties of oil and gas, natural conditions, as well as the overall development layout; through simulation and calculation in different operating modes, determine the best design and running data and ensure economic and reasonable running of pipelines.

Structure Design

Submarine oil and gas pipeline structure design technology: Calculate and analyze a submarine pipeline according to the route information on the submarine pipeline and the environmental conditions of the sea area including wind, wave and current, soil and geology conditions, internal fluid, platform displacement data, etc., thus ensuring that the submarine pipeline runs safely in its whole life cycle. Conventional calculation and analysis generally involve pipeline stability, on-bottom strength, riser strength, crossing, installation, etc.

Anticorrosion Design

The anticorrosion design technology for submarine pipelines is the key technology for ensuring long period safe operation of submarine pipelines in severe corrosive environment. For long-distance oil and gas pipelines or water transportation pipelines with various pipe diameters and operating conditions, select a correct and effective internal and external anticorrosion scheme; determine reasonable pipeline materials, coating systems, patching materials, and corrosion inhibitors; and take some measures like corrosion monitoring device design and other ways to ensure long-term effective and stable running of pipelines.

Construction and Installation Technology with Pipe Laying Ship Method

The construction and installation technology with pipe laying ship method includes process design technology for pipe laying operation line, pipeline behavior mechanics analysis technology, and mooring analysis technology. Simulate a submarine pipeline into a series of continuous beam elements using international standards and specifications and the elastic finite element method; simulate the tightener, support roller, and submarine boundary condition into rigid or elastic supports. The pipeline forms an S curve between the pipe laying ship and the mud landing point, and the curve is divided into upper bending section and lower bending section.

Construction with Drag Method

The construction technology with drag method: Bind a flotation buoy on a pipeline and make it float in water; then drag the pipeline to the pipeline design route using the tools such as ship and others; finally place the pipeline to the seabed. According to the state of the pipeline during towing in water, the drag method is divided into bottom drag method, off-bottom drag method, floating drag method, etc.

Submarine Pipeline Construction Complete Technology

The submarine pipeline construction complete technology includes horizontal interface docking technology, riser expansion bend installation technology, elastic laying technology, etc.

Submarine transport pipelines are used mainly for the transport of gas and are located predominantly around the Mediterranean and the Baltic and North Seas. Many have been created since 2000. In the Mediterranean, the earliest gas pipeline was the Trans-Mediterranean Pipeline, built in 1983 to link Algeria and the Italian mainland, via Sicily. This was followed in 1996 by the Maghreb-Europe Pipeline to link Morocco and Spain across the Strait of Gibraltar. Subsequent Mediterranean pipelines are the Greenstream Pipeline, built in 2004 between Libya and Sicily; the interconnector built in 2007 between Greece and Turkey; the link completed in 2008 between

Arish in Egypt and Ashkelon in Israel (which has been out of service since 2012); and the Medgaz Pipeline built in 2011 between Algeria and Spain. Further north, a link was built between Scotland and Northern Ireland in the United Kingdom in 1996. An interconnector was built between Belgium and the United Kingdom in 1998. The Balgzand/Bacton Line (BBL) connected the Netherlands and the United Kingdom in 2006. Finally, the Nord Stream Pipeline was completed in 2011 and 2012 through the Baltic, between Vyborg in the Russian Federation and Kiel in Germany. This is the longest gas transport pipeline in the world (1,222 km in length). Issues about its environmental impact bulked large in the negotiations leading to its construction, and particular problems were encountered over munitions dumped in the Baltic at the end of the Second World War (see Chap. 24 (Solid Waste Disposal)).

0.2 There are also a number of gas pipelines linking Norwegian gas production to its export markets. The Norwegian upstream gas transportation system has been developed from the 1970s and continues to develop, to cater for the transportation of natural gas produced on the Norwegian continental shelf. Norwegian domestic consumption of natural gas is limited. Almost all the gas produced is exported (101,000 million standard cubic meters) to European gas markets through landing terminals in Belgium, France, Germany, and the United Kingdom. The pipeline network in 2014 forms a 7,980-km integrated transportation system, transporting gas from nearly 60 offshore fields and 3 large gas processing plants on the Norwegian mainland, to European gas markets. The latest main addition to the system is the Langeled Pipeline, opened in 2007, which goes from the onshore processing plant in Norway for the Ormen Lange gas field to the United Kingdom, via a riser platform at the Sleipner field.

Outside Western Europe and the Mediterranean, there is a gas pipeline linking the Russian Federation and Turkey across the southeastern corner of the Black Sea and one linking the island of Sakhalin to the mainland of the Russian Federation in the North-West Pacific. Oil transport pipelines exist between Indonesia and Singapore

across the Strait of Malacca and, in China, linking the island of Hainan to Hong Kong. Generally, these submarine transport pipelines have been built and financed by oil and gas operators (including national oil and gas companies), sometimes in consortiums with national gas distribution undertakings.

Steel Riser

General

A riser system is essentially conductor pipe connection floaters on the surface and the wellheads at the seabed. There are essentially two kinds of risers, namely, rigid risers and flexible risers. A hybrid riser is the combination of these two.

The riser system of a production unit is to perform multiple functions, both in the drilling and production phases. The functions performed by a riser system include:

- Production/injection
- Export/import or circulate fluids
- Drilling
- Completion and workover

A typical riser system is mainly composed of:

- Conduit (riser body)
- Interface with floater and wellhead
- Components
- Auxiliary

The riser system must be arranged so that the external loading is kept within acceptable limits with regard to:

- Stress and sectional forces
- VIV and suppression
- Wave fatigue
- Interference

The riser should be as short as possible in order to reduce material and installation costs, but it must have sufficient flexibility to allow for large excursions of the floater.

Catenary and Top Tensioned Risers

In shallow water, it has been practiced to use top tensioned risers (TTRs), but as design for larger water depth, it has been accounted that the need for new design practice has increased. The ordinary TTR is very sensitive to the heave movements due to wave and current loads because the rotation at the top and bottom connections is limited. The heave movement also requires top tensioned equipment to compensate for the lack of tension. If the top tension is reduced, it will cause large bending moment along the riser especially if the riser is located in environment with strong current. If the effective tension becomes negative (i.e., compression), then Euler buckling will occur.

The catenary riser is self-compensated for the heave movement, i.e., the riser is lifted or lowered on the seabed. The catenary riser still needs a ball joint to allow for rotation induced by waves, current, and vessel motion, at the upper end connection.

The catenary riser is sensitive to environmental loads, i.e., wave and current, due to the normally low effective tension in the riser. The fatigue damage induced by vortex-induced vibration (VIV) can be fatal to the riser. Use of the VIV suppression devices such as helical strakes and fairing can reduce the vibration by 70–90%.

Steel Catenary Riser (SCR)

In recent years, oil and gas development and exploration activities in deep-water development have become more and more frequent, and the water depth has reached deep-water and even ultra-deep-water compared to previous years. With the increasing application of water depths, the technical equipment for deep-water development has also been facing new challenges, and the key technical equipment involved mainly includes the upper offshore platforms, and riser systems have also been developed in this challenge time and time again. From the beginnings of compliant platforms, tension leg platforms (TLP), single column platforms (Spar) and semi-submersible platforms to today's floating production systems and floating production storage and transportation systems (FPSO), the offshore platforms are better suited to the harsh marine environment. Among the currently

available riser systems, flexible risers, although they can comply with the drifting and lifting and sinking movements of floating bodies, are more costly and have some technical problems that have not been solved and are not adapted to the high temperature and pressure environment of offshore oil and gas extraction. This is when the steel suspended flow line riser was considered the most effective solution to emerge. It emerged in the mid-1990s and has been developed by many researchers and scholars and is now being used in the Gulf of Mexico, the North Sea, Camps Bay, and West Africa. This entry focuses on the basics of steel suspended flow line risers and the prospects for their application.

Structural Features and Advantages of SCR

SCR is called steel catenary riser, which is a combination of subsea pipeline and riser, where one end is connected to the upper floating platform and the other end is connected to the lower wellhead, without the connection of subsea stress joints, which greatly reduces the construction difficulty of the overall structure. There are some advantages of SCR, such as low cost, no upper top tension compensation, high tolerance to float drift and lifting and sinking movements, and suitability for high temperature and pressure environments. These features have led to the replacement of flexible and top tensioned risers as the riser system of choice for deep-water oil and gas resources development. Since 1994, when Shell laid the world's first steel catenary riser in the Gulf of Mexico at a water depth of 872 m, SCR has attracted widespread attention from the engineering and academic communities and is gradually gaining favors in the field of deep-water development.

Since the introduction of the steel catenary riser in the Gulf of Mexico, various new types of steel catenary risers have been introduced and developed to meet the needs of different water depths, and several different forms of steel catenary risers have emerged – simple catenary risers, steep wave risers, lazy-wave catenary risers, shaped steel catenary risers, mini-lazy-wave risers, and bottom weighted risers – and can be seen from Fig. 3. The simple catenary riser has a limited length of catenary available

for hull movement and is therefore mainly suitable for platforms with small hull movements; the other risers have greater compliance with hull translational movements than the simple catenary riser and can be more easily adapted to slow drifting movements with large floating hull structures (Fig. 2) and FPSOs (Fig. 1). The lazy-wave catenary riser and steep-wave catenary risers are raised by the addition of buoyancy sections.

The unique structural form of the steel catenary riser also poses new challenges for its design, manufacture, installation, and safe service. Among the factors controlling the design and safe service of the riser are the fatigue life of the top joints and touchdown points and the interaction between the flow line section and the seabed. Current research has shown that when a floating body moves in response to wind, waves, and currents, the overhanging section and the flow line section move simultaneously with the floating

body, causing the touchdown points to change along the axis and causing the flow line section to interact with the seabed. The fatigue damage at the touchdown point is mainly caused by the floating body movement and vortex-induced vibration, the stiffness of the seabed has a greater impact on the fatigue damage at the touchdown point, and the greater the seabed stiffness, the more serious the fatigue damage caused by the interaction between the riser and the seabed. Fatigue damage at the top is mainly caused by waves.

The Main Steps of SCR Design

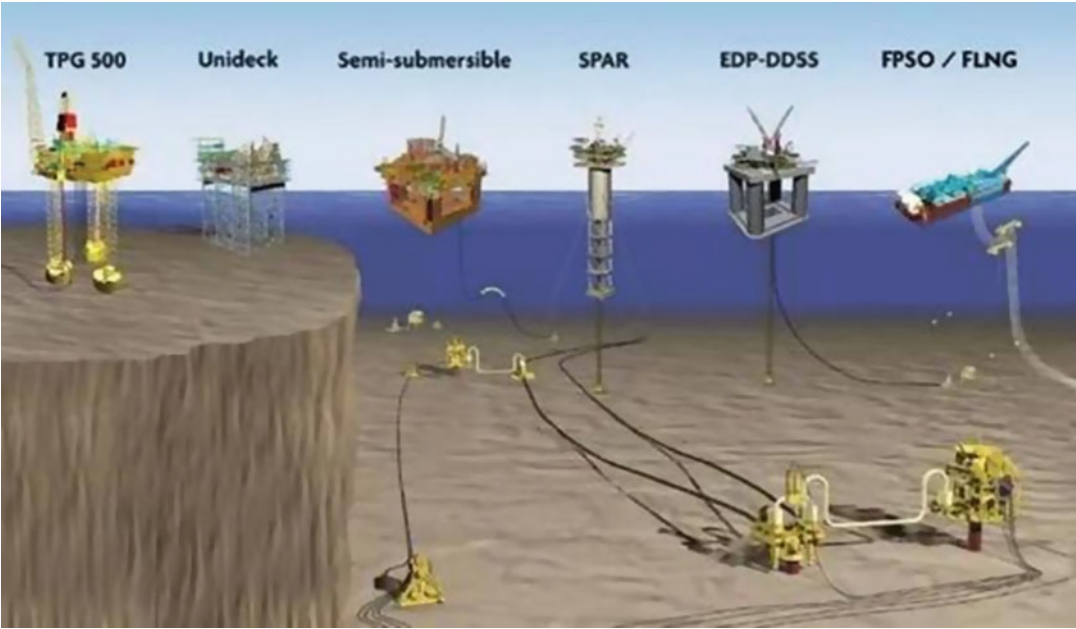
The main design steps for the SCR include:

1. **Determine the required dimensions of the riser.**

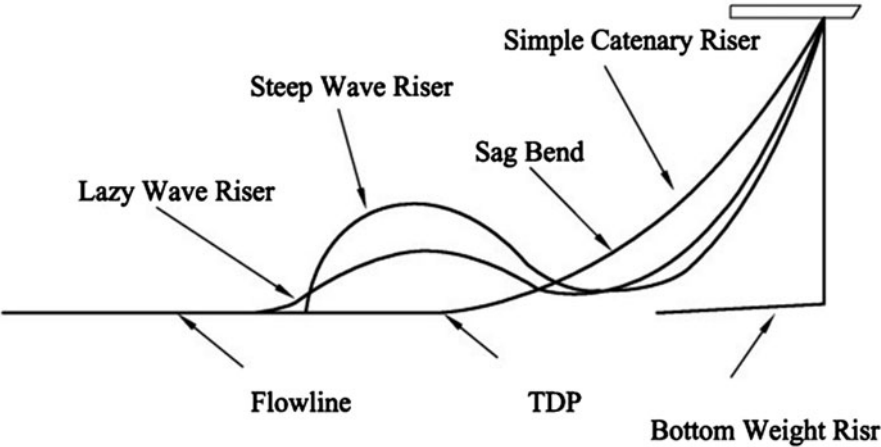
The dimensions of the riser need to be determined mainly include wall thickness, inner and outer riser diameter, riser length, top suspension angle, etc.



Steel Pipelines and Risers, Fig. 1 SCR with FPSO



Steel Pipelines and Risers, Fig. 2 SCR with different platform



Steel Pipelines and Risers, Fig. 3 Configurations of SCR

2. Static load design.

Static load design is mainly used to ensure that the stress in the critical area is within the allowable stress range when loaded with the load specified in the code; the main codes used here are API RP 2RD.

3. Dynamic load design.

The SCR motion response has natural non-linear characteristics, and it is easier to capture

the ultimate response values associated with wave forces and hull motion using time domain simulations than using frequency domain simulations.

4. Vortex-induced vibration computational analysis.

This section determines the sizing of vortex-induced vibration suppression devices.

5. **Fatigue analysis.**

Fatigue damage includes fatigue damage caused by vortex-induced vibration and fatigue damage caused by soil-riser interaction. The calculation of fatigue damage is determined based on the S-N curve specified in the code, partly applied to the code such as API RP 2A (1993).

6. **Flexible joint correlation analysis.**

This part mainly analyzes the ultimate stress at the upper point.

7. **Installation load analysis.**

This is used to analyze the installation load checks that occur when the joint is installed only at sea.

Analysis Methods for SCR

There are three main types of SCR analysis: numerical simulations, model tests, and field test monitoring analysis.

1. **Numerical simulation**

It is very difficult to use purely theoretical methods to analyse the riser with complex conditions, which include the direction of the catenary not only interact with the seawater,

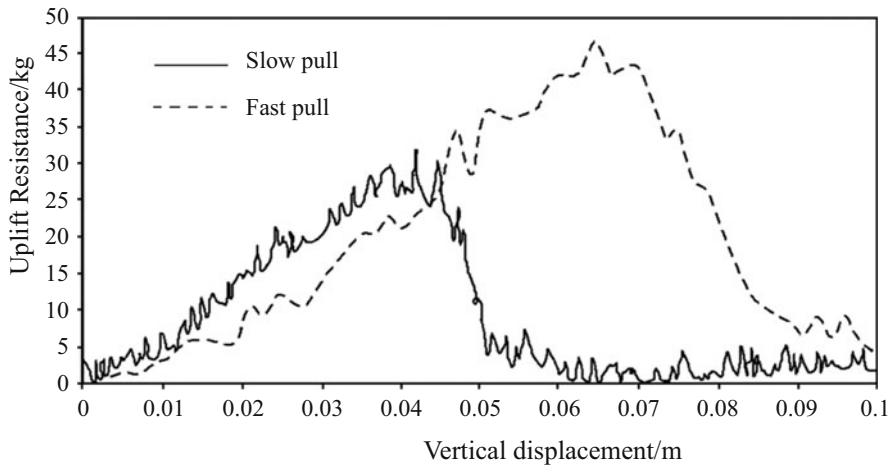
but also in the area of the touchdown point will also interact with the seabed soil.

2. **Model tests**

Model tests are generally divided into scaled-down model tests and full-scale model tests (Fig. 4). In the scaled-down model, the riser and pipe diameter data are significantly different from the actual model, which can reveal the dynamic response of the actual riser in some respects. The full-scale model test can simulate the real dynamic response of the riser more accurately; the full-scale simulations of the STRIDE JIP and CARISIMA JIP have provided much valuable information for the study of the mechanism of SCR riser-soil interactions; Fig. 5 shows the experimentally obtained curve of the effect of the riser pullout speed on the suction force. Lots of related tests have been carried out on this basis, but it is more costly than the scaled-down model. There is still some discrepancy and simplification between the conditions considered and the actual sea conditions, so a more suitable test model should be selected when choosing the model test, considering all circumstances.



Steel Pipelines and Risers, Fig. 4 Full-scale model test



Steel Pipelines and Risers, Fig. 5 CARISMA JIP model of uplift resistance

3. Field test monitoring and analysis

The field test monitoring analysis is based on the existing installation of the riser, some monitoring instruments are installed in the field riser to measure the dynamic response data of the riser, and based on these data, the analysis can provide reference significance for the subsequent numerical simulation.

Key Technologies for SCR

The key technologies of SCR are mainly focused on the following aspects:

1. The effect of floating body motion

SCR is mainly used to connect floating platforms to subsea wellheads, and floating structures generate large motion responses under the action of wind and wave currents. These responses mainly include first-order high-frequency responses and second-order low-frequency responses. The first-order motion of the floating body is an oscillation in the equilibrium position, which has a small amplitude and high frequency, causing a low-stress fatigue cycle in the riser, which forms the bulk of the high perimeter fatigue damage at the top of the riser. The second-order slow drift motion of the floating body consists of two main components, one being the large drift caused by wind and current and the other

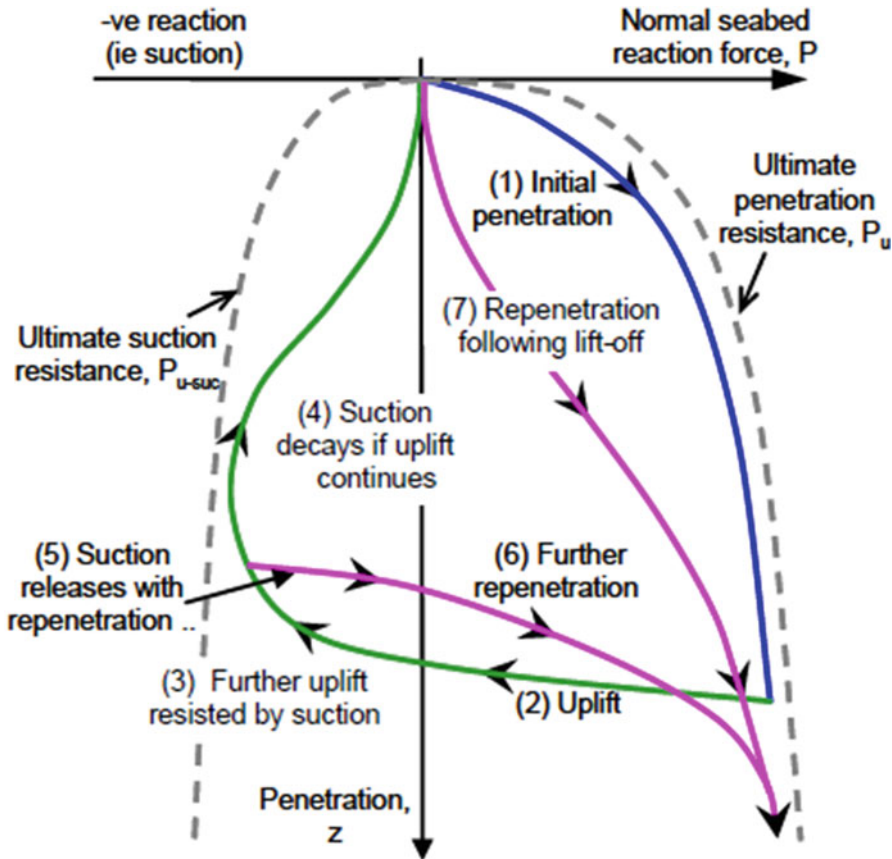
being the small amplitude oscillations caused by waves. In addition, the motion of the floating body can have an important effect on the motion of the riser touchdown points.

2. Interaction with the seabed soil

As the SCR moves with the floating body, its flow line segments interact with the seabed and are the most important parameter affecting the fatigue cycle at the touchdown point. Under the repeated action of the riser, trenches will be formed in the seabed, the width of which is generally 2 to 3 times the diameter of the riser and the depth of which is generally 0.5 to 1 times the diameter of the riser. The formation of the trench has a greater impact on the movement of the riser. When a substantial drift of the floating body occurred under the storm, the pull-out motion of the flowline and out of the plane movement will be subject to the resistance of the trench, causing the riser local stress increases. The riser tension is large when the suspension catenary is in a taut state, and the impact of the trench is particularly serious. Seabed stiffness model can be divided in two major categories just as shown in Figs. 6 and 7.

3. Vortex-induced vibration (VIV)

The fatigue damage caused by vortex-induced vibration is mainly concentrated in the static touchdown point; the nature of the



Steel Pipelines and Risers, Fig. 6 The RQ pipe-soil model

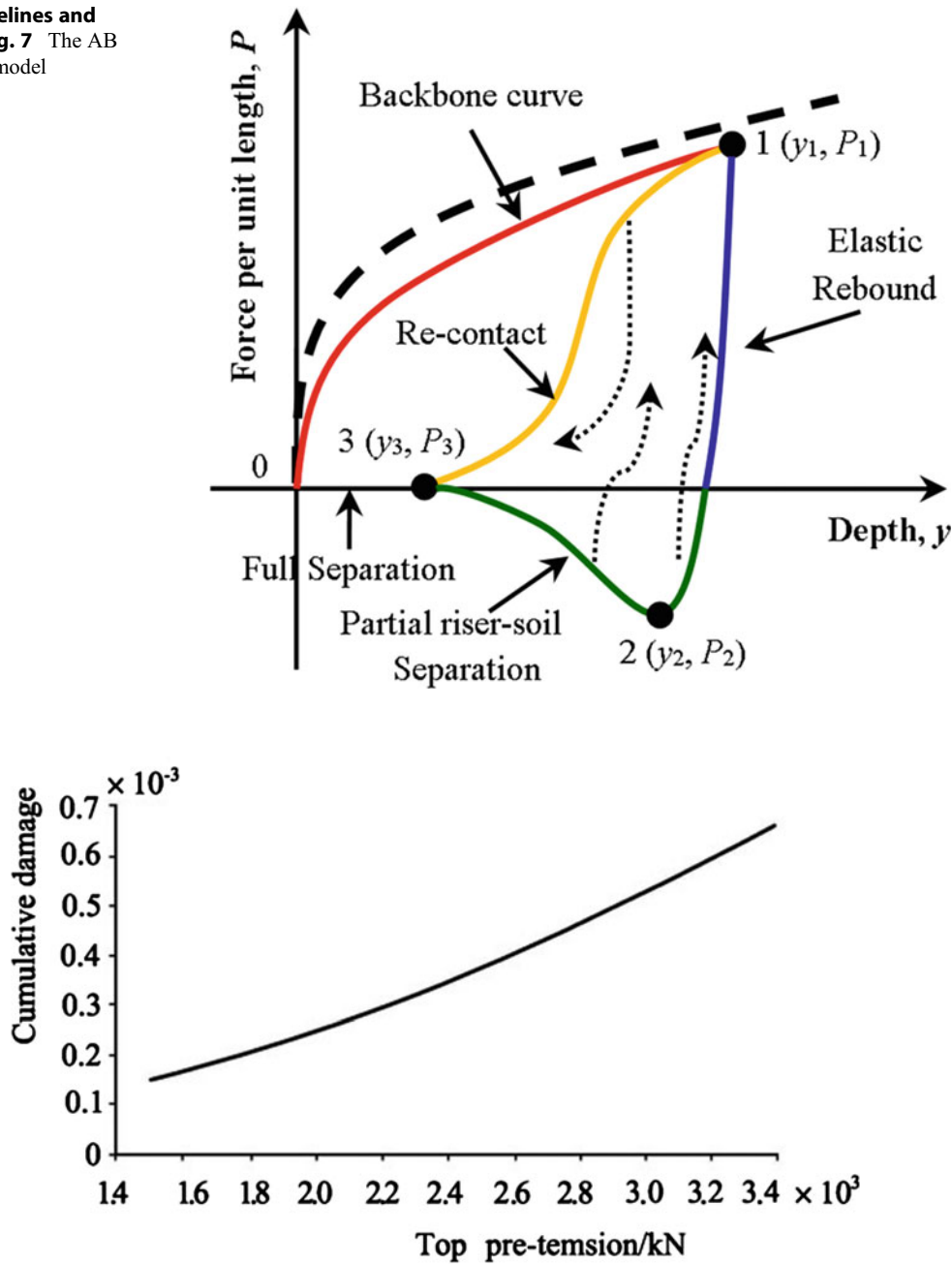
damage is high stress cycle of low circumference fatigue damage. The main influencing factors include the lifting and sinking motion of the floating body and the pulsating component of the second-order response, the subsea stiffness, and the dynamic characteristics of the eddy forces. VIV in steel catenary risers can be effectively controlled by welding spiral reinforcement plates to the first half of the catenary section. Studies have shown that a moderate rib height can effectively control the separation of the boundary layer and prevent alternating vortex shedding; the optimum height is 0.1 to 0.2 times the riser diameter. Otherwise, the drag force will be greatly increased. A moderate pitch can effectively change the position of the boundary layer separation and interfere with its axial correlation, typically five times the tube diameter.

A larger pitch will likewise lead to a substantial increase in drag force. The VIV suppression device can effectively reduce the fatigue damage caused by VIV.

4. Fatigue damage and fatigue life prediction

Fatigue life is the control parameter of the steel catenary riser design, its influence of many factors, so that the fatigue life prediction problem has become a difficult point in the design of the steel catenary riser. The influencing factors that have been identified are as follows: wave forces, flow, first-order response of the floating body, second-order response, and lift and sink motion. These influences cause fatigue damage at different locations in the steel catenary riser. Studies have shown that the top fatigue damage caused by second-order motion is much greater than first-order motion, shown in Figs. 8 and 9. By reducing

Steel Pipelines and Risers, Fig. 7 The AB pipe-soil model

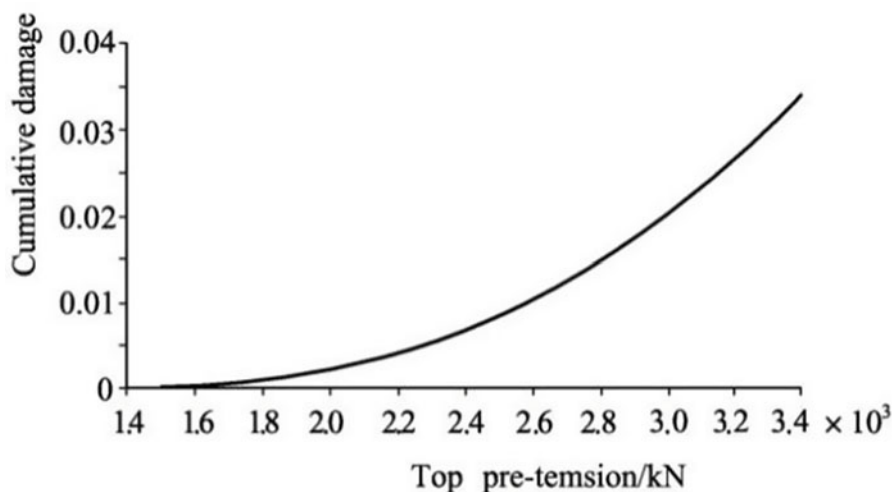


Steel Pipelines and Risers, Fig. 8 Fatigue damage caused by first-order motion of vessel

the stiffness of the flexible joint, the fatigue damage at the top can be reduced.

The steel catenary riser introduced in this entry has become the first choice for deep-water oil and

gas development due to its low cost, high temperature, and high-pressure medium environment suitable for deep-water oilfield and strong adaptability to floating body movement. Foreign research and engineering experience show that,



Steel Pipelines and Risers, Fig. 9 Fatigue damage caused by first-order motion of vessel

with the help of buoyancy wave and riser tower structure, the applicable water depth of steel catenary riser and the tolerance of floating body movement can be further improved, which becomes a cost-effective solution for deep-water and very deep-water riser systems.

Top Tensioned Riser (TTR)

Top tensioned risers (TTRs) are dry tree riser solutions typically suited for a closely clustered well reservoir, wherein they provide surface access to subsea wells, thus allowing workover and reentry drilling from the host vessel during life of field. TTRs are challenging due to their complex and highly iterative design, weight control, and multiple complex interfaces between numerous equipment packages. The key design issues of TTRs are focused around the parameters that influence the riser strength, fatigue, and clearance response as well as their configuration through the host vessel wallaby.

TTRs are vertical riser systems that require externally applied tension to maintain their structural stability. The top tension is also the primary method used to control the response of the riser. These riser systems are in tension over a part or all of their length. The required tension is provided by tensioner systems, buoyancy modules, or a combination of both. As implied by the name,

the external tension is applied near the top sections of these riser systems.

These risers are composed of several tubulars that are the extension of the well's casing and tubing strings. Some of the TTRs are used with fixed platforms or jackups and may look like free-standing risers in that no subsea wellheads are used. The top tension for these risers is required to provide additional stability to the riser system. The external tension for these risers is generally provided by nonmotion-compensating mechanical and/or hydraulic tensioners.

Subsea or marine wellheads are used with riser systems deployed from floating platforms. The subsea wellhead is normally considered to be the top of the well's high-pressure housing and is generally located about 3–5 m (10–15 ft) above the seafloor. The subsea wellheads are required to allow for the disconnection of the riser from the well. This disconnection capability is needed for emergency considerations and operational flexibility.

Historical Background

TTRs have an established track record for deep-water applications in the Gulf of Mexico (GoM), West Africa, and offshore Indonesia. TTRs have been in service on floating production systems from as early as 1984 when the Hutton tension

leg platform (TLP) was installed in the UK North Sea, in 486 ft. water depth. Since then, more than 30 dry tree production facilities have been installed using TTRs, with either a TLP or spar as the host vessel. The first TLP installation in the GoM was Shell Auger in 1994. While this development was in a water depth of only 2,860 ft., the technology has now been successfully extended into water depths exceeding 5,000 ft. The first spar installation in the GoM was Neptune in 1986 in 1,930 ft. water depth. The deepest spar installed to date is the Shell Perdido Spar in 8,000 ft. water depth in the GoM.

TTRs can be utilized for either production or drilling activities, and the key system components for the production and drilling TTR are described in following sections.

Production TTR

A production TTR consists of a section of vertical pipe supported at the vessel by individual tensioner systems and connected to the subsea wellhead via a tieback connector (Fig. 10).

The tensioner system provides global stability, controls the motion response of the riser, and ensures that positive effective tension is maintained at the base of the riser under all loading conditions.

One option to tension the TTR is using a hydropneumatic tensioner. Such a tensioner system may have either four (4) or six (6) cylinders/rods that are mounted into the TLP deck structure via various interfaces. The cylinder rod slides up and down relative to the cylinder barrel and provides the stroke for the tensioner system and maintains near constant tension. The cylinders and rods interface with the riser string via a tension ring that typically incorporates a concentric large pitch thread profile that is complementary to the threads machined onto the tension joint. The threaded section is incorporated to provide space out capability to accommodate riser installation stack-up tolerances. The tension joint interfaces with the tension ring and centralizer guide roller system. The centralizer guide roller at the tree deck elevation acts as the riser centralization point and provides lateral support. The roller guides are designed to accommodate lateral

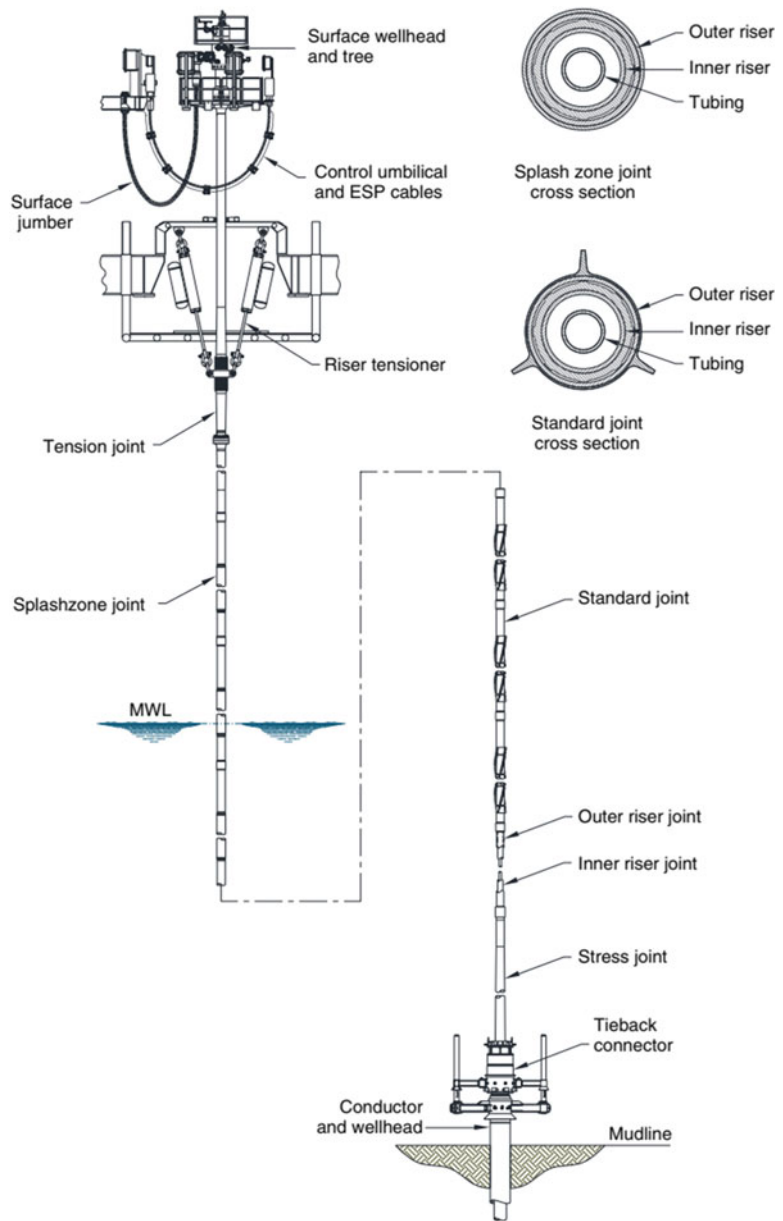
loads imparted by the riser in all environmental conditions. Top tension may also be provided by air- or nitrogen-filled buoyancy cans attached to the upper section of the TTR. In contrast to the hydropneumatic tensioners, buoyancy cans provide tension passively and minimize the interaction between the host vessel and TTRs.

Production TTRs are typically dual casing risers that have outer and inner riser strings. Production tubing is run through the inner riser to transfer production fluids to the production vessel. The production tubing is entirely supported from the surface tree body, with no hanger at the mudline.

The inner production riser is supported from the surface wellhead with a casing hanger and connects in the subsea wellhead with an internal tieback connector (ITBC). The ITBC provides the interface between the inner riser and the subsea wellhead assembly or the external tieback connector (ETBC). The ITBC must resist the tension overpull/preload that will be applied to the inner riser prior to setting the adjustable casing hanger in the surface wellhead. The ITBC contains a lockdown sub, which when rotated by a running tool, locks into the subsea wellhead housing or ETBC. The ITBC interface with the production inner riser joints is via a premium casing thread. The annulus between the inner riser and outer casing may be filled with a thermal barrier gel to maintain production fluid temperatures during operation, while the annulus between the inner casing and tubing is filled with brine for most operations.

Outer riser joints for the production risers extend from the tension joint to the stress joint assembly above the subsea wellhead. Outer riser pipe can be straked to suppress vortex-induced vibration (VIV) motion under current loads. The lower stress joint connects the riser to the subsea casing system at the subsea wellhead via the ETBC. The ETBC provides the interface between the outer riser and the subsea wellhead and must accommodate the tension and high bending loads imposed by the riser. The ETBC may be integral or flanged to the lower stress joint. The stress joint is used to manage the high bending stresses and fatigue loading at this localized section adjacent to the point of fixity at the seabed.

Steel Pipelines and Risers, Fig. 10 Typical production top tensioned riser schematic. (© 2H Offshore Inc.)



On a TLP, a guide wire system may be utilized during riser installation to direct the production TTR to the target wellhead and ensure no interference with adjacent structures.

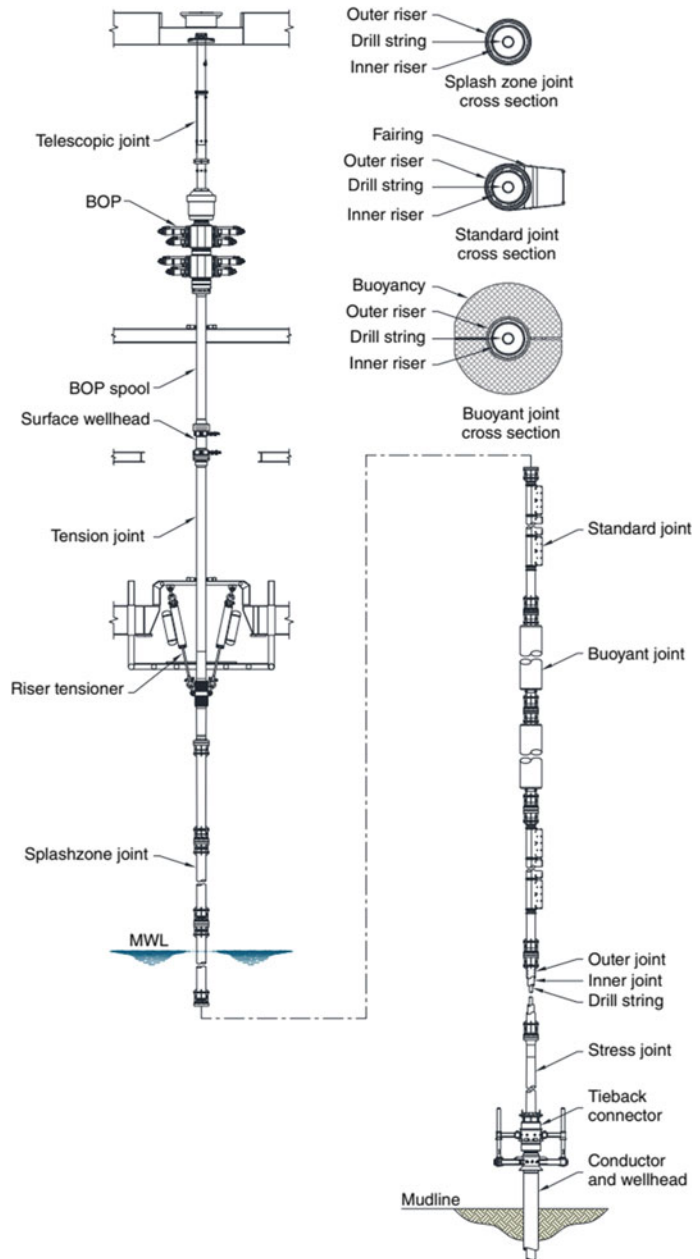
Drilling TTR

Drilling risers (Fig. 11) deployed from host vessels include a surface blowout preventer (BOP) stack to contain the pressure in the wellbore.

Therefore, the riser pipe needs to be designed to accommodate the incidental wellbore pressure. Drilling operations may be conducted through a single casing outer riser or through a drilling inner riser for dual casing configurations.

Similar to the production TTRs, the drilling TTR is supported on the host vessel by a tensioner system, which may be standardized to be identical

Steel Pipelines and Risers, Fig. 11 Typical drilling top tensioned riser schematic. (© 2H Offshore Inc.)



to the production tensioner or customized to accommodate the higher tension requirements for the drilling TTR. Non-standardized tensioner systems may include additional tensioner cylinders at the tensioner deck or a supplementary tensioning system at a separate deck level to meet the top tension requirements for the drilling TTR.

Outer riser joints may be seam-welded pipe sections with connectors welded at either end to form a continuous riser joint. The exterior of the outer riser joints is coated with thermal-sprayed aluminum (TSA) for corrosion protection. Thick-wall outer riser joints that are neoprene coated may be used in the splash zone to provide extra corrosion and impact protection. Buoyancy

modules may be attached to a section of the drilling outer riser, below the high current region, to reduce the loads on the tensioner system.

Similar to production TTRs, precision machined forgings, with profiles optimized to accommodate the high strength and fatigue loads at TTR interfaces with the vessel and the mudline, are located at the upper and lower ends of the drilling riser in the form of the tension joint and stress joint assemblies, respectively. The stress joint assembly connects and seals to the high-pressure housing on the subsea wellhead via the ETBC. Similar to production TTRs, fairings and strakes may be utilized as VIV suppression devices to reduce fatigue damage caused by riser VIV.

The drilling inner riser, if used, is installed inside the drilling outer riser and is suspended with a casing hanger that lands inside the drilling surface wellhead. At the base of the drilling inner riser is an ITBC that latches inside the subsea wellhead housing. The drilling inner riser is made concentric to the drilling outer riser by the use of centralizers.

TTR VIV and VIM Response

In the absence of suppression measures, VIV of TTRs may be a significant contributor to fatigue damage and may dictate the design of such risers. Suppression devices such as strakes and fairings are used in high current environments to mitigate VIV. Fairings offer the additional benefit of reducing drag on the TTR.

The fatigue response of the TTR to vortex-induced motions (VIMs) of the hull also needs to be considered during design. The hull VIM response is independent of water depth and may be more onerous for TTRs deployed in shallow water depths (the large ratio of the VIM amplitude to water depth results in higher cyclic bending loads at the base of the TTR).

Installation and Commissioning Considerations

The first consideration in any TTR installation campaign is the evaluation of environmental conditions to determine whether weather conditions are acceptable to complete the TTR installation.

There are many tools available to aid this evaluation and equipment to monitor real-time conditions for this assessment.

To land the riser on the subsea wellhead, a method of guiding the riser is required, which varies based on the vessel type. For dynamically controlled semisubmersible vessels, the riser can be guided by adjusting the position of the vessel using the dynamic positioning (DP) thruster system. This provides for large adjustment and good accuracy. A spar platform can guide the riser using the vessel mooring system; for TLPs, there is no means of controlling the vessel position so a dedicated guide wire system is utilized to guide the riser base to the desired subsea wellhead. A guide wire system generally has a winch or multiple winches on the vessel with wire rope connected to a structure on the subsea wellhead. A guide arm structure clamped to the riser base connects the riser to the guide wire. The riser is guided to the subsea wellhead using increased tension in the guide wire as the riser approaches the wellhead.

A TTR can be installed from different offshore platform types, but a drilling rig is always required to perform the installation work. The drilling rig uses many tools to install a TTR, and consideration of the tool interfaces with the riser equipment must be evaluated to ensure that the equipment can perform the job as intended. Riser equipment will be staged on an available deck or stacked up vertically in a pipe rack depending on riser pipe size or vessel design/duty. Riser pipe is typically lifted to the drill rig floor using the platform crane and presented to a joint handling device in the horizontal position, and the joint is then lifted to the vertical position using an elevator type design to hold the joint and position it over the rotary opening.

The joint is then lowered through the drill rig floor and landed in a securing system in the rotary with the upper connection of the joint presented upward for the connection of the next joint. The riser connections will be either a casing-type design that requires the use of makeup power tongs to torque the threaded connections to the required value or a flange-type design that requires tensioning of studded connections on

the flange to achieve a required preload. These operations are performed at the rig floor when the next joint is presented to the rotary, and subsequent joints are run repeating the same steps.

The taper stress joint with ETBC is designed to connect to the subsea wellhead or subsea equipment stacked on the wellhead. Standard outer riser joints make up the majority of the riser length. Once all standard outer joints are connected, provision can be made to perform a space out exercise either by running more joints or by lowering the riser on drill pipe to the subsea wellhead; this operation confirms the length between the vessel and subsea wellhead for comparison to calculated values. Once elevations are confirmed, the riser installation is completed by running a selected number of pup joints to correctly space out the riser followed by the tension joint.

With the tension joint connected and the full riser string hanging from the drill rig, the riser can be lowered and connected to the subsea wellhead. Typically, connection is made using an ROV for visual monitoring and hydraulic actuation of the connection system. With the riser connected to the subsea wellhead, the riser can be tensioned to the required top tension for engagement of the tensioner system. Once engaged, the riser tension can be handed from the rig to the tensioner system by increasing the tension through the tensioner system and decreasing the hook load.

With the outer riser fully supported on the tensioners, the riser is ready to begin surface equipment installation. For a single casing drilling riser, typically a wellhead spool and BOP system are attached to the top of the tension joint and tested to confirm surface equipment and riser integrity before drilling operations commence.

For dual riser systems, a surface wellhead spool is attached to the top of the tension joint prior to running inner riser joints. Inner riser joints are installed using the same method as outer joints, and the first inner joint is capable of latching and sealing into a profile within the subsea wellhead to isolate the inner riser from the outer annulus. Once all inner riser joints are run and spaced out correctly to match the outer riser length, the inner riser can be tensioned to the

required value and latched and sealed into a profile within the surface wellhead. Some adjustment of tension through the tensioners may be required to support the additional load of the inner riser.

For a drilling riser, a BOP system can be attached to the top of the wellhead and tested to confirm riser integrity before drilling operations commence. For production risers with surface equipment, a surface tree is installed on top of the wellhead and a BOP system connected to the top of the tree. With the BOP system in place, the riser has well control barriers to begin running tubing downhole and perform well completion operations to bring the well online.

Additional installation activities required before full production are the installation of control lines to the riser system through a control umbilical system supported between topsides and the top of the riser. A flexible jumper is also required to convey fluid between tree and topside production manifold.

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Steel Pretreatment

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Definition

Marine steel, through the process of rolling transportation and stacking, often rusts, deforms, and is covered with various oil stains and impurities, and the direct usage of which will result in not only the reduction of steel performances but also poor processing accuracy and low production speed. Therefore, the open-air stacked steel must go through the steel pretreatment procedures in order to be used for hull components processing (Liu et al., 2011).

Introduction

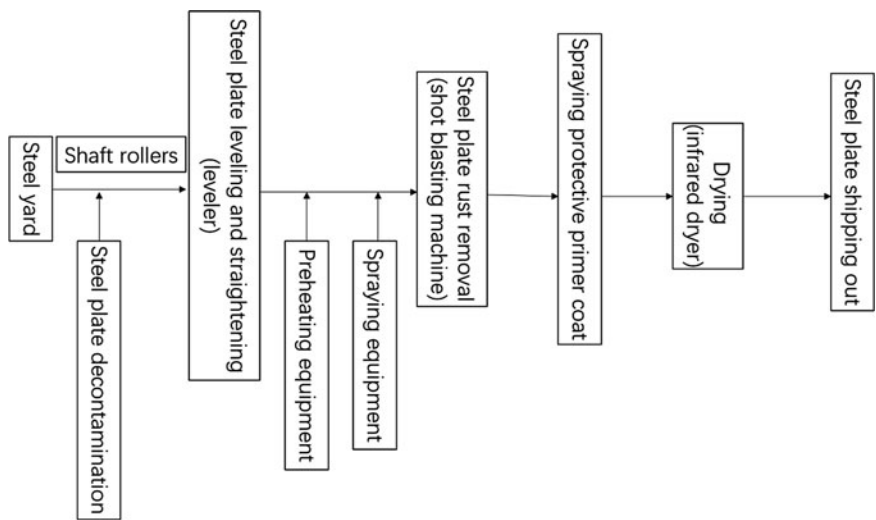
Steel Pretreatment Assembly Line

The steel pretreatment automatic assembly line refers to the automatic operation assembly line composed of steel transportation, leveling and

straightening, rust removal, primer coat spraying, drying, and other procedures, which is categorized into two types: steel plate pretreatment assembly line and profile pretreatment assembly line. There are also cases of the treatment of both steel plate and profile on the same production line. This introduction is mainly about the steel plate pretreatment assembly line (Xu, 2005).

The technological process and main equipment of the steel plate pretreatment assembly line are shown in Fig. 1:

- (1) In the steel yard, the steel is first lifted onto the roller table by a crane or an automatic loading truck, as shown in the Fig. 2.
- (2) The roll table passes the steel to a multi-roll leveler and flattens the plate, as shown in the Fig. 3.
- (3) Some steel plate pretreatment assembly lines are equipped with preheating devices and spraying devices. The leveled steel plate is sent to the preheating device by the conveying table roller, and the temperature of the steel plate reaches 40 °C ~ 60 °C. The purpose of preheating is to remove the moisture on the surface of the steel plate and loosen the oxide scale and rust spots for later removal and to improve the adhesion of the paint film. The



Steel Pretreatment, Fig. 1 Technological process and major equipment of steel pretreatment assembly line

purpose of the spraying device is to remove salt from the surface of the steel plate.

- (4) The steel plate enters the procedure of impeller blasting rust removal, with the shot blasting device automatically throwing shots to both sides of the steel plate (the



Steel Pretreatment, Fig. 2 The roller table



Steel Pretreatment, Fig. 3 The multi-roll leveler



Steel Pretreatment, Fig. 4 The shot blasting machine

shots can be recycled and reused), and hot air removing the dust on the surface of the steel plate. Figures 3 and 4 respectively show the steel plate before entering the shot blasting machine and the steel plate after rust removal.

- (5) After rust removal and cleaning, the steel plate enters the spraying chamber where the maintenance primer coat is sprayed. Spraying operations are carried out in a semi-enclosed spray chamber, as shown in Fig. 5. To adapt to steel plates of different widths, the round-trip distance of the spray gun can be adjusted. In order to maintain continuous coating and a certain film thickness, the conveying speed of the roller should be in line with the spraying speed of the spray gun, which can be adjusted through electrical control.
- (6) After leaving the spraying chamber, the steel plate is dried in the drying chamber. The film



Steel Pretreatment, Fig. 5 The spraying chamber





Steel Pretreatment, Fig. 6 The drying process



Steel Pretreatment, Fig. 7 The processing workshop

drying methods include infrared rays, far infrared rays, and electric heating, all of which require a ventilation device to facilitate the evaporation of the paint solvent and to speed up the drying process, as shown in Fig. 6.

- (7) After being dried, the plate leaves the drying chamber and is sent out of the pretreatment assembly line through the high-speed roller table and into the processing workshop for the marking off and processing after passing the quality inspection, as shown in Fig. 7.

It should be noted that since the rust removal chamber and spray chamber are filled with iron dust and spray, to maintain a good working environment and to prevent the diffusion of harmful gases into the open air, special attention should be paid to dust collection, ventilation, and explosion prevention, and the corresponding environment protection

measures and the fire and explosion prevention measures should be taken (Huang, 2013).

Steel Leveling and Straightening

The steel plates and profiles used in shipbuilding, due to the uneven rolling and uneven cooling contractions after rolling, and various factors during transportation and storage, often have waves, local concavities, and distortions. These deformations will affect the normal marking-off, gas cutting, and other machining procedures, and reduce machining accuracy (Shu et al., 2013). In addition, these deformations may also cause additional stress or component instability during welding, thus affecting the strength of components. Therefore, the steel plates should be leveled and straightened to eliminate these deformations. In the shipyard, the deformations of the steel plates are usually leveled and straightened by a multi-roll leveler, and the deformations of profiles by a profile leveler.

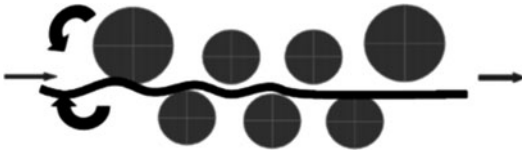
Steel Plate Leveling and Straightening

The deformation of a steel plate is caused by that some fibers are turning shorter or longer than others. Therefore, plate leveling and straightening is actually shortening the longer fibers or lengthening the shorter fibers so that these fibers are at the same length with their surrounding ones, but in reality, the method of lengthening fibers is often adopted because of the difficulty to compress fibers.

A leveler commonly used in a shipyard is composed of two rows of working rollers, which often consist of 5–11 working rollers, as shown in Fig. 8. The lower row is the row of active shaft rollers, which is fixed on the machine body by bearings and cannot be adjusted at all, and the rotation of which is driven by the motor through the speed reducer. The upper row is the row of driven shaft rollers. The vertical gap between the upper and lower rollers can be adjusted by the manual screw or the electric adjusting device for leveling steel plates of various thicknesses. When being leveled, the plate gets in between with the rotation of the rollers and is subjected to multiple alternating small curvature bendings from opposite directions between the upper and lower



Steel Pretreatment, Fig. 8 The leveler machine



Steel Pretreatment, Fig. 9 The leveling machine with the upper and lower rollers parallel

rollers, where plastic deformations are generated as the bending stress exceeds the yield limit of the material. Thus, shorter fibers are lengthened and the plate is leveled.

Multi-roll levelers, according to the different arrangements of the rollers and the positions of the adjustable rollers, can be divided into two types: multi-roll levelers with the upper and lower rollers parallel and separately adjustable (Fig. 9), and the leveling machines with the upper and lower rollers unparallel (Fig. 10).

Profile Leveling and Straightening

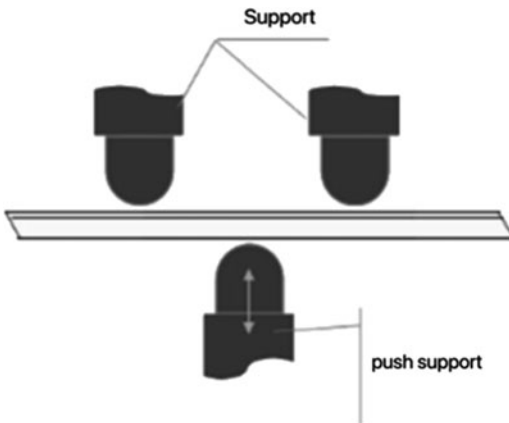
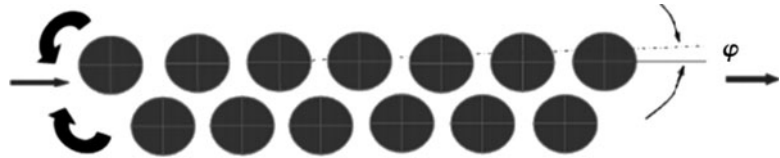
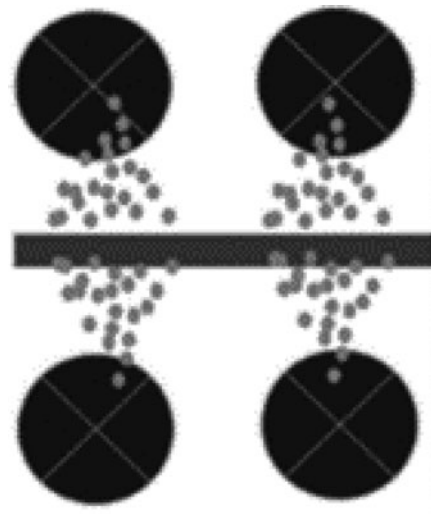
The profile is mainly leveled and straightened by the profile straightener. Fig. 11 shows the working principles of the profile straightening machine, which consists of two supports and a push support. The two supports have no power transmission, but the distance between them can be adjusted according to the need. The push support is installed on a sliding block, which is capable of horizontal reciprocating movements driven by a

motor through a speed reducer. During profile straightening, the deformation section of the profile is set between the two supports and is deformed in the opposite direction by the force of the pushing support, and thus the profile is straightened.

In the absence of special profile leveling and straightening equipment, small profiles can be leveled and straightened by manual tapping on platforms or round piers; large profiles can be leveled and straightened using flame heating and water cooling method or with hydraulic press machines, and it is necessary to configure compression modules in accordance with the profile shape when using hydraulic press machines (Kennedy et al., 2004).

Surface Cleaning of Steel

Steel surface cleaning refers to the procedure of removing the oxide scale, rust, and other impurities on steel surface (generally referred to as rust removal), and then spraying a protective primer coat. The common methods of steel surface cleaning involve manual rust removal, mechanical rust removal (rust removal of shot blasting and rust removal of impeller blasting), chemical rust removal, hydro blasting rust removal, etc. At present, the widely adopted methods in steel pretreatment are rust removal of impeller blasting and chemical rust removal.

Steel Pretreatment,**Fig. 10** The leveling machine with the upper and lower rollers unparallel**Steel Pretreatment, Fig. 11** The profile straightening machine**Steel Pretreatment, Fig. 12** Basic form of shot blasting machines**Rust Removal of Impeller Blasting**

Rust removal of impeller blasting is one of the rust removal methods in which iron shots or other abrasives are flung onto the surface of steel at high speed to remove oxide scale and rust spots.

The shot blasting machines in use now are mostly horizontal, as shown in Fig. 12, and the steel can be directly transmitted into them through the raceway so that automatic assembly lines can be set up easily. When horizontal shot blasting machines are used to remove rust, the iron shots are easily spread on the upper surface of the steel plate, so it is necessary to set up high-pressure blowers or mechanical scrapers to momentarily remove the iron shots on steel plates.

Generally, shot blasting machines are equipped with a shot recycling system to facilitate the recycling of shots, and a ventilation and dust removal device to reduce their environmental pollution (Nguyen and Oterkus 2020).

Chemical Rust Removal

Chemical rust removal usually refers to a multi-step pickling process, with the basic process as

follows. After being degreased and washed, the steel is placed in the pickling tank, where the oxide scales and rust spots are removed. After water washing, the residual acid is neutralized with lye to prevent corrosion of the steel, and then the steel is rinsed with water. In chemical rust removal, hydrochloric acid, sulfuric acid, phosphoric acid, or a mixture of them is often used as a rust removal solution.

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STL, Submerged Turret Loading Concept

- [Moored Ship in Ice](#)

Strapdown Inertial Navigation System (SINS)

- [Integrated Navigation](#)
- [Optical Compass](#)

Strap-down Inertial Navigation System (SINS)

- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)

Structural Analysis and Design of VLFS

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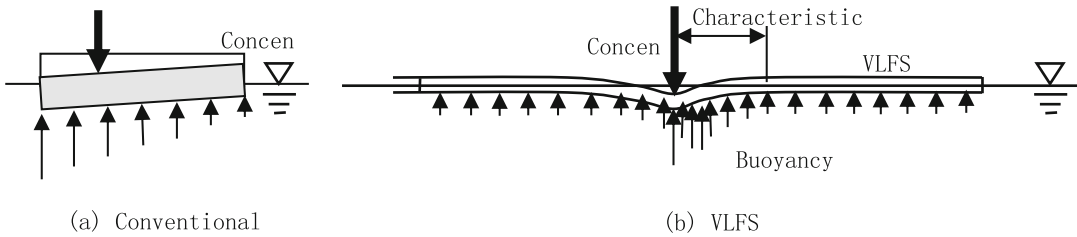
Definition

The dominant load to very large floating structure (VLFS) is the hydrodynamic load by ocean waves. The wave loads on VLFS may be altered due to the structural deformation of VLFS, which is the so-called hydroelasticity effect. It is important to proceed the structural design of VLFS while performing and using the hydroelasticity analysis in parallel. The structural design is thus combined with analysis. This article illustrates how the hydroelasticity analysis of different levels may be incorporated into the structural design of VLFS, by classifying the process into initial design stage, scantling stage, and evaluation stage.

Scientific Fundamentals

Key Concept

VLFS has large dimensions in horizontal directions compared with that in depth-wise. The structure is flexible to deform under wave actions, and the magnitude of the deformation can be equivalent to or exceed that of the rigid body motions. The flexible deformation characterizes the response of VLFS. Figure 1 shows a schematic comparison of the response between a conventional rigid ship and VLFS subjected to a point load (Suzuki and Yoshida 1995). The rigid ship bears the force and moment due to the point load by the increase of the translational and angular displacements, while VLFS distributes the load along the structure by the deformation mechanism; VLFS deforms and interacts with the surrounding static/dynamic pressure fields. This makes the hydroelasticity theory play the role. In the structural analysis and design of VLFS, how to



Structural Analysis and Design of VLFS, Fig. 1 Rigid body motion vs flexible deformation to a point load. (Reproduction from Suzuki 1995)

consider the hydroelastic effect at as early stage as possible is the key issue (Suzuki et al. 2006).

Structural Analysis Method

Primary, Secondary, and Tertiary Response

Discussion starts with the structural analysis of conventional ship structure. The structural stress in a ship may be classed into “primary,” “secondary,” and “tertiary” stresses. These structural stress types may be evaluated separately and superposed together to give a picture of the stress distribution within the structure. The analysis method and the structural model depend on the stress levels to be considered. The primary stress is the longitudinal stress due to hull girder deformations when the ship hull girder is regarded as a ship beam. The longitudinal stress may be understood and analyzed by adopting a beam theory. The load to the beam model may be evaluated by the rigid body motion analysis together with hydrodynamic theories. The so-called strip theory is used to this end. The hydrostatic/hydrostatic pressure and inertial loads which are balanced among themselves can be evaluated and given to the beam model to find the sectional loads such as vertical bending moment.

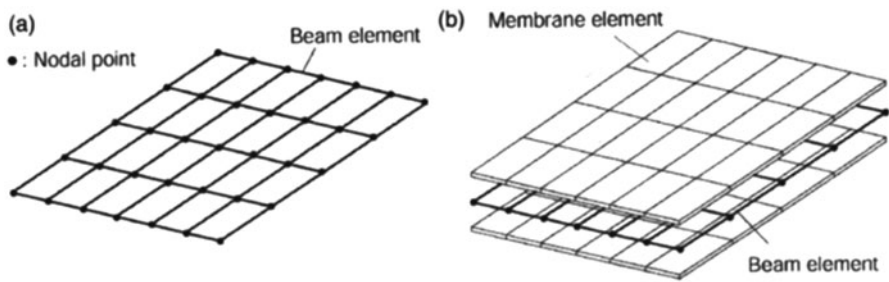
The secondary stress is the stress due to the transverse deformation when the structure is subjected to loads such as static/dynamic pressure, internal pressure, etc. For example, the double bottom structure may deform to the pressure as a whole. This type of stress may be analyzed by using a so-called hold model with the pressure applied on the structure. The hold model is usually a three-dimensional finite element (FE) model including the transverse structural members such

as girders/floors, transverse frames, double bottom, etc. Usually, the shell and beam elements are used for the modeling, and the membrane stress is only referenced. The tertiary stress is the local bending stress due to the local pressure. The local pressure may include static/dynamic/internal/external ones such as panting, slamming, heavy concentrated weight, etc. The local stress is applied to the local model (e.g., stiffened plate model).

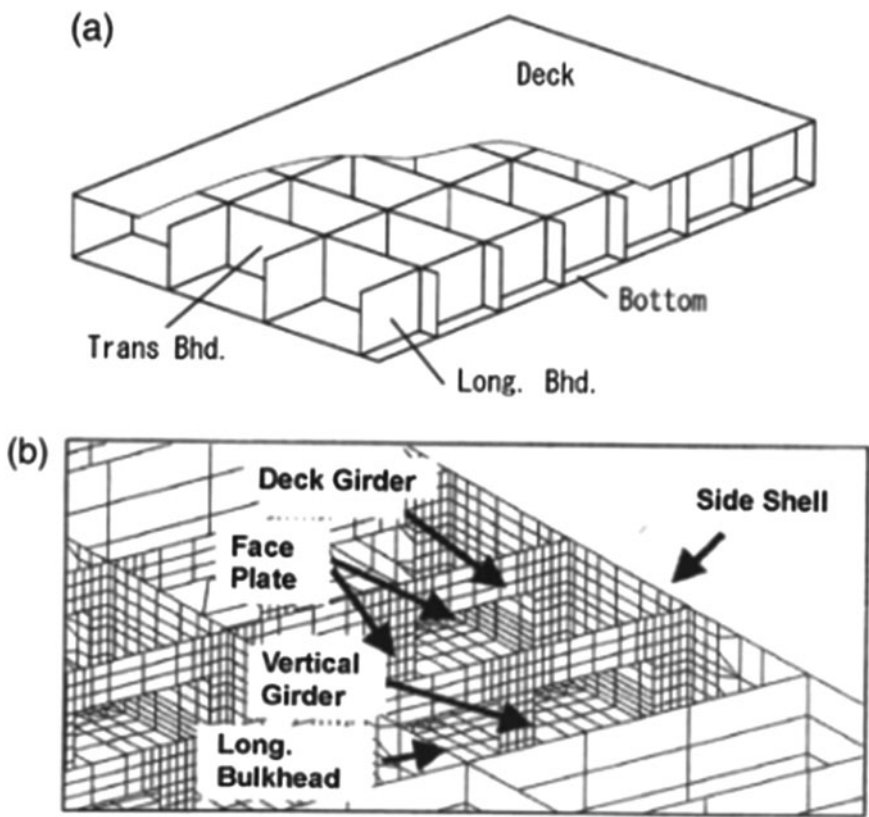
Structural Analysis of VLFS

When one regards that the above classification applies to the analysis of VLFS, too, the only one big difference from that of ship structure is found to be the evaluation of the hull girder (primary) stress of VLFS. The hydroelasticity theory should be applied to the prediction of the primary stress in VLFS (Suzuki 2006; Iijima 2018). The structural model for the hydroelasticity analysis can be a beam grillage which model the whole VLFS (Fig. 2, left). When one considers about the large horizontal plane of VLFS, it is also an idea to use the plate model based on plate theory (Fujikubo and Yao 2001). However, at least, the anisotropic plate model is recommended to be used for the structural analysis to account for the anisotropic properties of the structure; the structure may exhibit different deformations between the longitudinal and transverse bending, due to the difference of girder arrangements in longitudinal and transverse directions, respectively.

Further, the sandwich grillage model (Fig. 2, right) is advantageous over the beam grillage model in a point that it can represent the torsional deformation correctly. The shell FE model (Fig. 3) is the other choice. It should be pointed out,



Structural Analysis and Design of VLFS, Fig. 2 Grillage model and sandwich grillage model. (Taken from Fujikubo and Yao 2001)



Structural Analysis and Design of VLFS, Fig. 3 Shell model. (Taken from Seto et al. 2003)

however, that the detailed modeling consumes time both in terms of model preparation and analysis, and detailed information is not available at the early design stage. The pressure for the evaluation of the secondary stress is also imported from the hydroelasticity analysis which is used for the evaluation of the primary stress in VLFS.

Table 1 compares the structural analysis between ship structure and VLFS. This procedure may be termed as “two-step” method in which the hydroelasticity analysis is performed firstly; then primary and secondary stress are evaluated. It is similar to the “two-step” method for ship structure, in which the rigid body motion analysis with

Structural Analysis and Design of VLFS, Table 1 Structural stress level, structural model, and applied load used for the direct analysis and design: ship structure vs VLFS

	Ship structure	VLFS
Load analysis	Rigid body motion + hydrodynamic theory, e.g., strip theory	Flexible body deformation + hydroelasticity theory
Primary stress	Beam model under the pressure/inertia evaluated by using the load analysis	Hydroelasticity theory + beam assumption
Secondary stress	Three-dimensional shell FE model under the pressure/inertia evaluated by using the load analysis	Three-dimensional shell FE model under the pressure/inertia evaluated by using the load analysis
Tertiary stress	Local structural model under the local load	Local structural model under the local load

hydrodynamics is performed firstly; then subsequent analyses are made (Oka et al. 2002).

There is the other method: “one-step” method (Seto 2003). In this method, the primary stress and secondary stress are evaluated simultaneously. The shell FE model may be used seamlessly between the primary stress analysis and the secondary stress analysis. The primary and secondary stress may be simultaneously evaluated, then. The one step method should be useful at the late design stage for confirming the stress level more accurately.

Structural Design of VLFS

Structural Design Formula for VLFS Taking Account of Hydroelasticity

Objectives The design process inevitably follows the so-called design spiral for optimization. When the structural design of ship is revisited, one may notice that design loads are usually used, instead of the direct motion-load analysis. The designer can easily know the design hull girder loads once the main particulars of the ship is determined, which facilitates the convergence of the design spiral. For the structural design of VLFS, it is important to consider the hydroelastic effect even at the preliminary design stage, as early as possible in place of using design loads for VLFS. The hydroelastic behavior can be evaluated only after the main particulars and structural design scantlings are fixed (because it is a fluid-structure interaction problem), and the structural data necessary for the hydroelasticity analysis have become available. It is thus required to develop such a formula to roughly estimate the hydroelastic effect based on

several design parameters such as size of structure, stiffness of structure, etc.

Theoretical Background When one considers VLFS with uniform mass, stiffness, and shape, the VLFS may be modeled to a uniform beam on an elastic foundation. This simple model applies to both semi-submersible type (e.g., MOB) and pontoon type (Mega-Float) VLFS. The hydrodynamic interactions among the floaters are neglected, herein. The objective of setting up this simple model is to derive the parametric dependencies. When it is subjected to a regular wave with the circular frequency ω and with the wave height H , the governing equation in Eq. (1) is given from the force equilibrium among loads on a strip with unit length. The vertical deflection w is taken upward and the x axis along the VLFS with the center of the origin at the midship. The variable k denotes the wave number of the incident wave.

$$(m + m_a) \frac{\partial^2 w}{\partial t^2} + c \frac{\partial w}{\partial t} + EI \frac{\partial^4 w}{\partial x^4} + k_c w = F_z \cos(kx - \omega t) \quad (1)$$

where m is the mass per unit length, m_a the added mass per unit length, c the damping force coefficient per unit length, EI the bending rigidity, k_c the restoring force coefficient per unit length, and F_z the force amplitude per unit length which is proportional to the wave height H . The value of these variables can be regarded constant here by following the assumption of the uniformity even though they could be a function of time and space in reality.

The solution to Eq. (1) may be given by a sum of a particular solution and homogeneous solutions (Suzuki and Yoshida 1995). The homogeneous solutions to Eq. (1) give a large contribution to the behavior near the both ends, however, decay exponentially towards the inner part of the structure from the ends. This implies that the effect of disturbance at the ends decays quickly with the distance from the disturbance. The disturbance is distributed over a length. This length is called “characteristic length” and discussed in the next paragraph. The particular solution is then more relevant when discussing the design parameters. The analytical solution by this approach gives a good insight into the hydroelastic behavior of a VLFS.

The particular solution w_p to Eq. (1) is given in Eq. (2). The damping term is dropped here for simplicity.

$$w_p(x) = \frac{F_z}{-(m + m_a)\omega^2 + EI k^4 + k_c} \cos(kx - \omega t) \quad (2)$$

Characteristic Length and Maximum VBM The length over which the effect of disturbance decays and averaged is defined as a characteristic length λ_s (see Fig. 1) and given by a simple formula in Eq. (3).

$$\lambda_s = 2\pi \sqrt[4]{EI/k_c} \quad (3)$$

For a frequency region lower than a heave natural frequency given as $\sqrt{k_c/(m + m_a)}$, the inertia term or the first term in the denominator of Eq. (2) is negligible; then the vertical bending moment (VBM) obtained from Eq. (2) reduces to Eq. (4).

$$\begin{aligned} M &\approx -EI \frac{\partial^2 w_p}{\partial x^2} = EI \frac{k^2}{EI k^4 + k_c} F_z \cos(kx - \omega t) \\ &= EI \frac{1}{\left(k \sqrt{EI/k_c}\right)^2 + 1/k^2} \frac{F_z}{k_c} \cos(kx - \omega t) \\ &\leq 2\pi^2 EI \frac{1}{\lambda_s^2} \frac{F_z}{k_c} \approx \pi^2 EI \frac{H}{\lambda_s^2} \end{aligned} \quad (4)$$

In the above, the vertical force amplitude F_z is approximated rather conservatively to a product

of the restoring force coefficient k_c and the wave amplitude $H/2$ (times the breadth) in the low-frequency region. The vertical bending moment takes a maximum value determined by the wave height H , the bending rigidity EI , and the characteristic length λ_s , and it is attained when the incident wave length $2\pi/k$ is equal to the characteristic length λ_s , irrespective of the length of the VLFS. It is interesting to note that the characteristic length which firstly has been introduced as the length over which the disturbance is distributed is also related to the maximum vertical bending moment in VLFS. Equation (4) is applied to the prediction of VBM in VLFS which has the length sufficiently larger than the characteristic length.

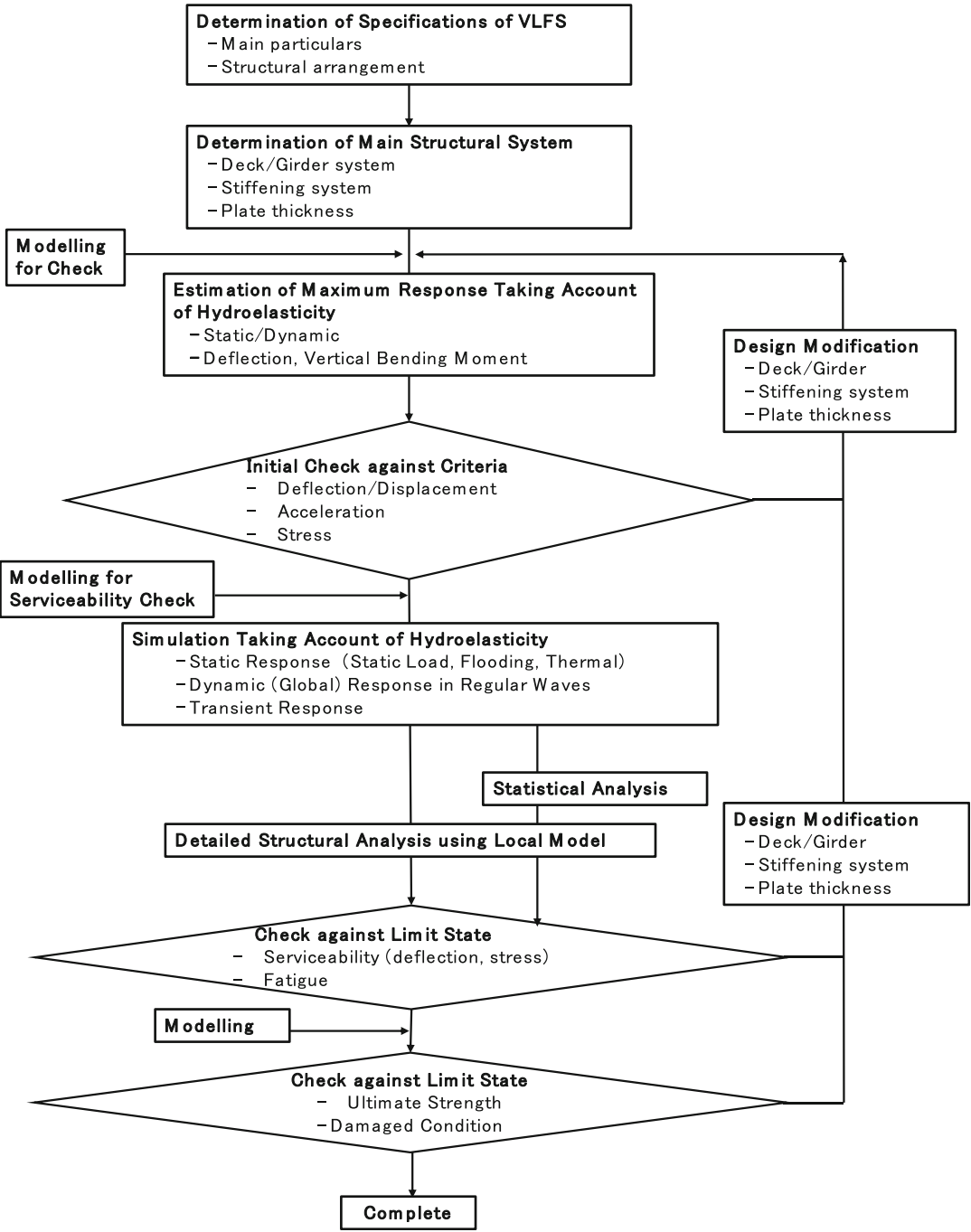
Structural Design Flow of VLFS

The structural design flow of VLFS can be summarized as follows (see Fig. 4):

- (i) Initial planning
- (ii) Structural arrangement of large structural members, e.g., floors and girders, with a consideration of Eq. (4), or hydroelasticity analysis using plate model
- (iii) Structural modeling using grillage or sandwich grillage model
- (iv) Hydroelasticity analysis (simulation)
- (v) Initial check performance and stress, initial design stage
- (vi) Scantling of structural members, e.g., longitudinal stiffeners, frames, and transverse members
- (vii) Structural modeling including all the members
- (viii) Detailed analysis for transverse members taking account of hydroelasticity. One-step (hydroelasticity analysis) or two-step method (static analysis)
- (ix) Check performance and stress, scantling check stage
- (x) Safety check and evaluation (ultimate strength, Tsunami, damaged condition, etc.)

The items ii–iv and vi–viii are looped until the design converges.

What is special about the design of VLFS is that the numerical analysis based on



Structural Analysis and Design of VLFS, Fig. 4 Design flow for VLFS (Reproduction from JASNAOE Committee 2004)

hydroelasticity theory is required for all the stages of design. One may call it “Design-by-Analysis,” while that for ship is “Design-by-Rule.” The

hydroelasticity analysis is used both in initial design check and final design check. At the final safety evaluation stage, safety margin to the

extreme loads (e.g., extreme weather, tsunami) and redundancy of the system (damaged condition) must be checked. The safety margin may be evaluated by performing nonlinear structural analysis such as ultimate strength analysis. Buckling, yielding, and rupture of the structural members should be considered.

Sometimes, the final safety evaluation may be done with the use of reliability theory, or probability of failure. The target safety level of VLFS is another issue. Some studies are done in the past and may be consulted with Suzuki (2001).

Cross-References

- [Floating Bridge](#)
- [Hydroelasticity Theory](#)
- [Mega-Float](#)
- [Very Large Floating Structures \(VLFS\): Overview](#)

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Structural Characteristics of Polar Engineering

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Synonyms

[Structural mechanics of polar engineering](#)

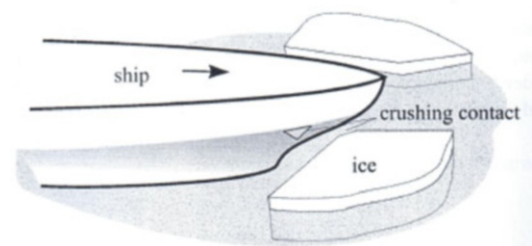
Definition

Structural characteristics of polar engineering mainly refer to the deformation, stress, and strain of the polar structure under the load of sea ice. According to the polar structural characteristics, the structural strength may be determined so as to prevent them from damage or accidents.

Scientific Fundamentals

Design Scenario and Development of Design Loads

The design scenario forming the basis of the ice loads for plating and framing design is a glancing collision on the shoulders of the bow, see Figs. 1 and 2 (Daley 2000). In this scenario, the ship is assumed to be moving forward at the design



Structural Characteristics of Polar Engineering, Fig. 1 Design scenario – glancing collision on shoulder (Daley 2000)

speed, striking an angular ice edge. During the collision, the ship penetrates the ice and bounces off. The ship speed, ice thickness, and ice strength are assumed to be class dependent. The maximum force can be calculated by equating the normal kinetic energy with the energy used to crush the ice. The ice crushing force cannot exceed the force required to fail the ice in bending. The force limit due to bending is determined by the angles, ice strength, and also the thickness.

The rule scenario is strictly valid only for the bow region and for the stern of double-acting ships. In order to produce a balanced structural design, loads on other hull areas are set as a proportion of the bow area by using empirical hull area factors (Kendrick et al. 2007). The loads on other hull areas are not strongly dependent of the bow angles, and so bow loads are normalized using a “standard” set of bow angles when applied elsewhere.

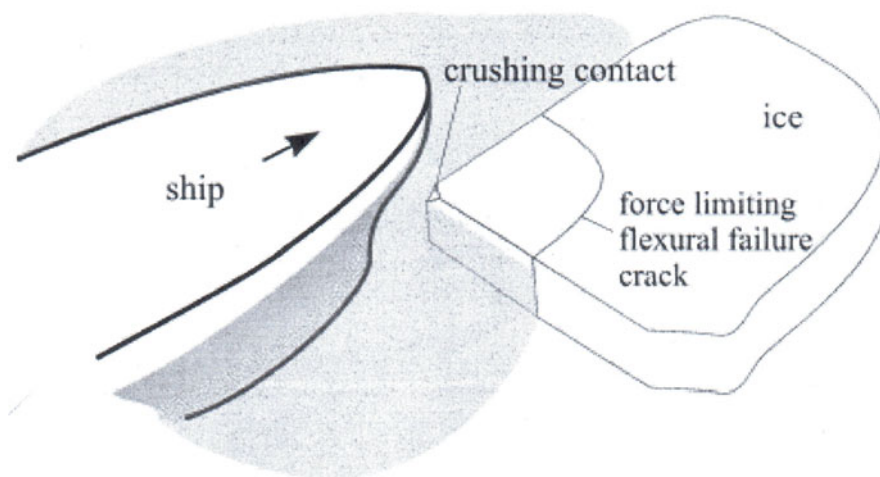
Another operational scenario involves a collision on the midbody. Two cases are considered here. In the first case, the vessel gains a lateral velocity due to an impact at the bow (Daley and Kendrick 2008; see Fig. 3 below). The midbody impact may occur immediately after the bow collision or be somewhat delayed, depending on the amount of sea room between the ship and the second impact.

In this scenario, the impact velocity in the midbody will be a function of the initial bow impact magnitude and location, and of the time lag between the first and second impacts, depending on the channel width or distance between ice floes. Here the second impact is assumed to take place at the mid-ship so that the influence of the yaw (turning) velocity can be ignored.

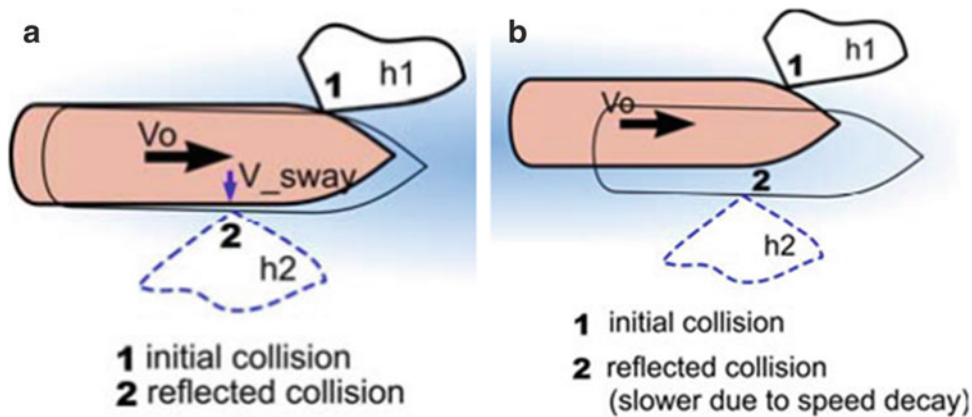
A second case of glancing impact results from turning in a channel (Daley and Kendrick 2008; see Fig. 4 below). In this case the impact velocity depends on the turning rate, and will be higher aft of amidships. For vessels with a better turning ability, for instance, those equipped with azimuth thrusters, special attention should be given for this scenario.

The discussion hereafter will focus on the case of collision with a reflected impact but the results and conclusions can also be applied to the case of turning collision.

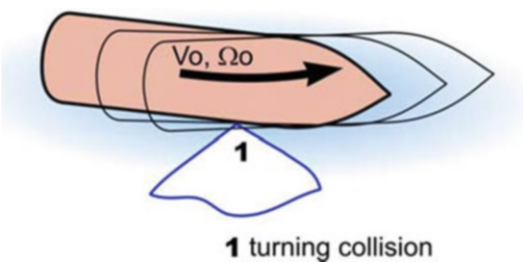
The design loads are obtained in several steps. First, the total load is found as the minimum of the crushing and flexural limiting loads for the design ice. Then, the patch over which this load is applied is determined and idealized. Finally, the distribution of load within the patch is modified to account for the local loading peaks.



Structural Characteristics of Polar Engineering, Fig. 2 Design scenario – flexural failure during glancing collision (Daley 2000)



Structural Characteristics of Polar Engineering, Fig. 3 Bow-midbody glancing sequence – (a) immediate, (b) delayed (Daley and Kendrick 2008)



Structural Characteristics of Polar Engineering, Fig. 4 Midbody glancing during a turn (Daley and Kendrick 2008)

Structural Requirements for Polar Class Ships

Hull Areas

The hull of polar class ships is divided into areas reflecting the magnitude of the loads that are expected to act upon them. In the longitudinal direction, there are four regions: bow, bow intermediate, midbody, and stern. The bow intermediate, midbody, and stern regions are further divided in the vertical direction into the bottom, lower, and icebelt regions (IACS Req. 2011; see Fig. 5 below).

The upper and lower ice waterlines on which the design of the vessel has been based is to be indicated in the classification certificate. The upper ice waterline (UIWL) is defined by the maximum draughts fore, amidships, and aft. The lower ice waterline (LIWL) is to be defined by the minimum draughts fore, amidships, and aft.

The lower ice waterline is determined with due regard to the vessel’s ice-going capability in the ballast loading condition (e.g., propeller submergence).

At no time is the boundary between the bow and bow intermediate regions to be forward of the intersection point of the line of the stem and the ship baseline. The aft boundary of the bow region need not be more than 0.45 L aft of the forward perpendicular (FP). The boundary between the bottom and lower regions is to be taken at the point where the shell is inclined by 7° from the horizontal plane. If a ship is intended to operate astern in ice regions, the design of aft section of the ship should also satisfy the bow and bow intermediate hull area requirements.

Shell Plate

The required minimum shell plate thickness, t , is given by:

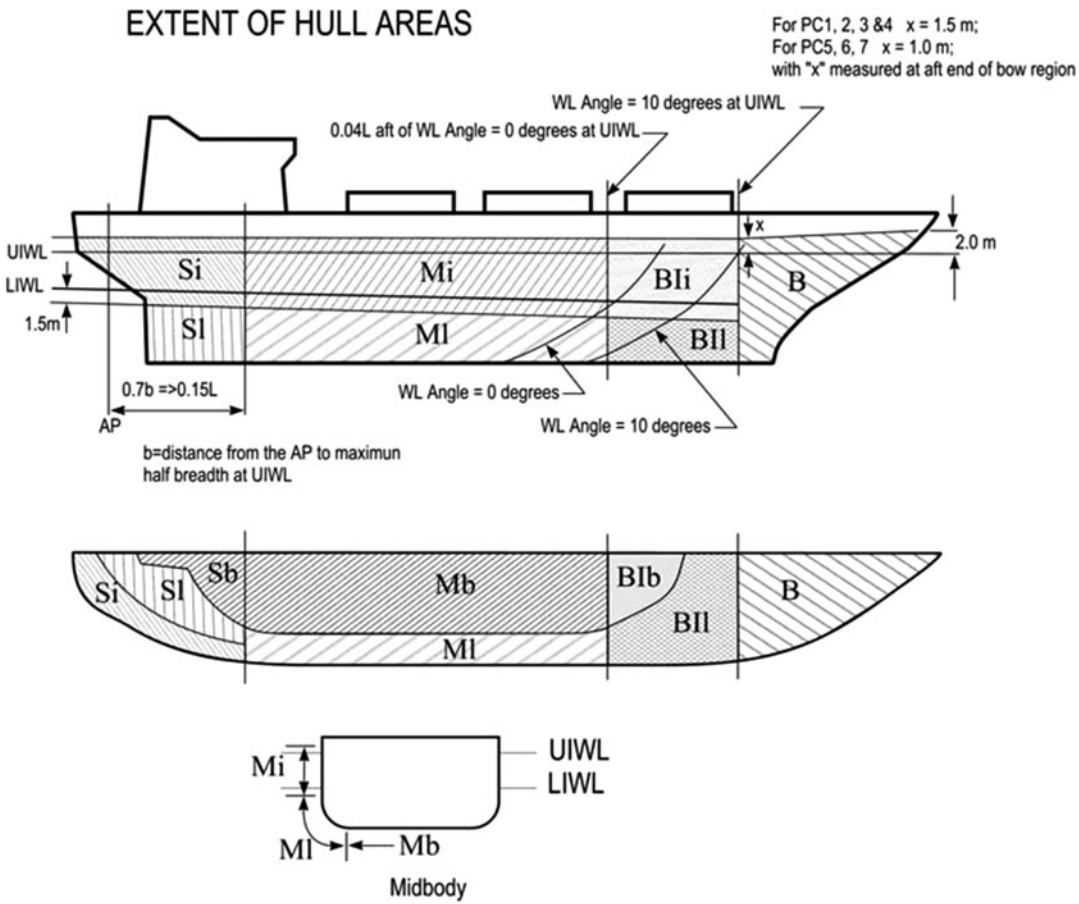
$$t = t_{net} + t_s \text{ [mm]} \tag{1}$$

where

t_{net} = plate thickness required to resist ice loads
 t_s = corrosion and abrasion allowance
 t_{net} , depends on the orientation of the framing (IACS Req. 2011; see Fig. 6 below).

Framing

Framing members of polar class ships are to be designed to withstand the ice loads. The term



Structural Characteristics of Polar Engineering, Fig. 5 Hull area extents (IACS Req. 2011)

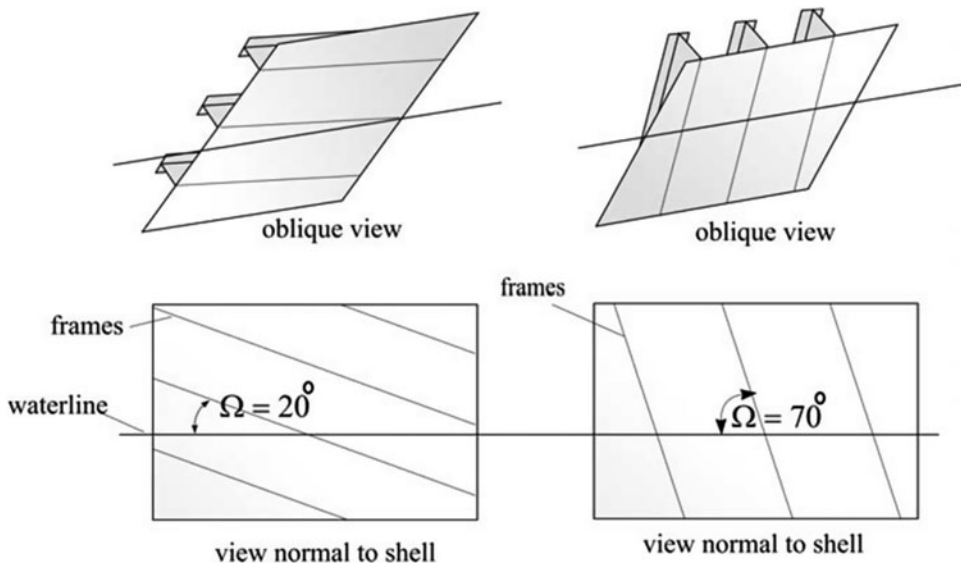
“framing member” refers to transverse and longitudinal local frames, load-carrying stringers, and web frames in the areas of the hull exposed to ice pressure (IACS Req. 2011; see Fig. 5), where the load-distributing stringers have been fitted; the arrangement and scantlings of these are to be meet the requirements of each member society.

The strength of a framing member is dependent of the fixity that is provided at its supports. Fixity can be assumed where framing members are either continuous through the support or attached to a supporting section with a connection bracket. In other cases, simple support is to be assumed unless the connection can be demonstrated to provide significant rotational restraint. Fixity is to be ensured at the support of any framing which terminates within an ice-strengthened area.

The details of framing member intersection with other framing members, including plated structures, as well as the details for securing the ends of framing members at supporting sections, are to be in accordance with the requirements of each member society.

The design span of a framing member is to be determined on the basis of its molded length. If brackets are fitted, the design span may be reduced in accordance with the usual practice of each member society. Brackets are to be configured to ensure stability in the elastic and post-yield response regions.

When calculating the section modulus and the shear area of a framing member, net thicknesses of the web, flange (if fitted), and attached shell plating are to be used. The shear area of a framing



Structural Characteristics of Polar Engineering, Fig. 6 Shell framing angle (IACS Req. 2011)

member may include that material contained over the full depth of the member, i.e., web area including portion of flange, if fitted, but excluding attached shell plating.

The local frames in transversely framed side structures and in bottom structures (i.e., hull areas BIb, Mb, and Sb) are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism.

Side longitudinals are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism.

Web frames and load-carrying stringers are to be designed to withstand the ice load patch as defined in I2.3. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimized.

Plated Structures

Plated structures are those stiffened plate elements in contact with the hull and subject to ice loads.

These requirements are applicable to an inboard extent which is the lesser of:

- (i) web height of adjacent parallel web frame or stringer
- (ii) 2.5 times the depth of framing that intersects the plated structure

The thickness of the plating and the scantlings of attached stiffeners are to be such that the degree of end fixity necessary for the shell framing is ensured. The stability of the plated structure is to adequately withstand the ice loads.

Longitudinal Strength

Ice loads need only be combined with still water loads. The combined stresses are to be compared against permissible bending and shear stresses at different locations along the ship's length. In addition, sufficient local buckling strength is also to be verified.

The strength criteria provided in Table 1 are to be satisfied (IACS Req. 2011). The design stress is not to exceed the permissible stress.

Local Details

For the purpose of transferring ice-induced loads to supporting structure (bending moments and

Structural Characteristics of Polar Engineering, Table 1 Longitudinal strength criteria (IACS Req. 2011)

Failure mode	Applied stress	Permissible stress when $\sigma_y / \sigma_u \leq 0.7$	Permissible stress when $\sigma_y / \sigma_u > 0.7$
Tension	σ_a	$\eta \sigma_y$	$\eta \mathbf{0.41} (\sigma_u + \sigma_y)$
Shear	τ_a	$\eta \sigma_y / (3)^{0.5}$	$\eta \mathbf{0.41} (\sigma_u + \sigma_y) / (3)^{0.5}$
Buckling	σ_a	σ_c for plating and for web plating of stiffeners $\sigma_c / \mathbf{1.1}$ for stiffeners	
	τ_a	τ_c	

where

σ_a = applied vertical bending stress [N/mm²]

τ_a = applied vertical shear stress [N/mm²]

σ_y = minimum upper yield stress of the material [N/mm²]

σ_u = ultimate tensile strength of material [N/mm²]

σ_c = critical buckling stress in compression [N/mm²]

τ_c = critical buckling stress in shear [N/mm²]

$\eta = 0.8$

shear forces), local design details are to comply with the requirements of each member society.

The loads carried by a member in way of cutouts are not to cause instability. Where necessary, the structure is to be stiffened.

Key Applications

Stiffness Curves Used for Structural Design and Verification of a Typical Icebreaker

To properly understand the structural interaction of icebreaker grillage section components, LR has used nonlinear finite element methods to compute the characteristic stiffness curve well into plasticity. The characteristic stiffness curve is considered representative of the effective structural interaction of the section components and has been found to relate directly to the section design methodology (elastic or plastic). Pearson et al. (2015) analyzed a set of existing icebreaking vessels and presented a simple acceptance criterion that requires the structure to carry at least 150% of the design load before reaching a plastic strain of 2.5%.

An Acceptance Criterion for Evaluation of Primary Structures of Polar Class Ships

Ville et al. (2020) proposed a robust and simple-to-use assessment methodology and acceptance criteria. The proposed method provides means to ensure proper structural hierarchy. Appropriate

safety margins are described through the plastic response of the local frame and the primary structural member. Allowable plastic deformation caused by the design ice load is determined by relating the permanent set of the structure to newbuild vessel production tolerances. An example analysis comparing the proposed method to existing linear and nonlinear methods is presented. The proposed method ensures that the structure remains serviceable throughout the lifetime of the vessel and provides a tool to make the primary structures of polar class vessels lighter and safer by filling the gap of missing detailed acceptance criteria for nonlinear analysis in the PC rules.

Guidelines for Transverse Web Frame Design in Polar Class Ships

Edward (2018) presents guidance on how to develop accurate finite element models for polar class ships and provides guidance on appropriate acceptance criteria for the plastic design/behavior of transverse web frames. Some design considerations should be taken into account to ensure the design will be sufficient in accidental overload scenarios. Several geometric models were created to study how different design factors influence the response of the transverse structure. One of the models created was that of a PC3 icebreaking side shell structure. For most overload scenarios, the solver will require both material and geometric nonlinear capabilities. Depending on how the

load is modeled, an indenter may be used which requires nonlinear boundary capabilities in the form of contact.

A New Ice-Resistant Jacket Platform Based on Field Monitoring

Offshore structures in ice-covered waters are exposed to the actions of drifting ice floes. The ice sheets may push over a weak platform or generate significant loads on it. The loads affect not only the routine operation of an offshore platform such as drilling and production activities, but also the safety and serviceability of the structure. In order to reduce the dynamic ice load acting on the platform, the offshore structure's conventional design should be changed.

Firstly, the global ice load acting on structure can be reduced by reducing the number of legs exposed to ice action, and reducing diameter of legs to lower the projection area of ice excitation. On the other hand, the dynamic ice load acting on structure might be reduced by changing the failure mode of ice floes. Conventional steel jacket platforms always use cylinder leg at water level, and ice crushing failure on cylinder exerts the highest loads. Installing a conical component at water level is an effective solution to change the failure mode of ice from crushing to bending, and to reduce the magnitude of ice load significantly (Wang et al. 2012).

In order to make use of the research result, a newly designed platform, JZ20-2NW, was deployed in 2005. On this platform (Wang et al.

2012; see Fig. 7 below), new conceptual designs were used, including single-leg design and icebreaking cone constructed at water level.

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Structural Characteristics of Polar Engineering,
Fig. 7 The JZ20-2NW platform

Structural Design

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Synonyms

Conceptual design; Construction design; Detailed design; Preliminary design; Structural strength

Definition

Structural design refers to the creative work that transforms the design tasks and requirements of the structural system into technical documents that can be implemented. The designer needs to analyze, calculate, draw, and test all the components of the structural system according to the actual original conditions and finally present all the drawings for production and the relative technical documents. This process usually involves constant revisions to pursue perfection, which brings together the creativity and inventiveness of the designers (Yu 2010).

Scientific Fundamentals

The Main Contents of Structural Design

The contents of structural design can be summarized as follows (Yu 2010):

1. Structural layout design. The locations and deployment of components and equipments of the structural system should be proper and co-ordination.
2. Selection of structural shapes and arrangement of the bearing members. When subjected to working loads, the structures demand reasonable forms, which do not only have good mechanical properties to meet the requirements about strength and stiffness but are as light as possible.
3. Determining the sectional dimensions of the structures according to task assignments and the specifications.
4. The selection of structural materials. The material selection of marine engineering needs comprehensive considerations. Metal, composite materials and non-metallic materials are commonly used in marine structures, among which steel, titanium alloy, composites, and ceramics are most widely used.
5. The selection of components composing structural system and the ordering requirements, as well as corresponding technical requirements and standards are supposed to be decided in structural design.
6. The strength, stiffness of the structures, and corresponding tests. Structural strength and stiffness are the capability of structures to withstand external loads, namely, the ability to resist fracture, failure, and deformation or displacement. In addition to theoretical calculation, mechanical properties of structures must be verified through relevant tests.
7. Interface design. Interfaces include the connections between components, equipments in the structural system, and the connections among different systems. Interface design deals with a lot of coordination work throughout the whole process.
8. The design and layout of maintenance channels. Convenient repairments and efficient maintenances of the structures will be based on reasonable layouts of maintenance channels.
9. Forming a set of drawings that can be used to produce as well as all the technological files in the design process.

The overall project design encompasses conceptual design, preliminary design, technical design (detailed design), and construction design in successive order, and the emphases of structural design vary with different design stages (Yu 2010; Zhu 1992).

In conceptual design, a variety of structural design schemes are usually made, each of which will be analyzed to be compared according to design project description requests, and finally a reasonable design scheme is chosen after comprehensive considerations and assessments about feasibility and economy.

In the preliminary design, the structural design focuses on fully understanding and analyzing the original conditions such as the design tasks and the overall design data, analyzing the loads, service conditions, production conditions, determining the principal dimensions of the structures as well as various interface relations, the structural configuration, the arrangement of the bearing components, and completing the overall design drawing of the structures.

In the detailed design, main endeavor will be done in the following aspects. Carrying out

structural analyses of components, especially, fatigue, durability, and damage tolerance are also analyzed and accessed for the reused parts and components. During the design process, necessary tests should be carried out to check the properties of material or the performance of the structures. A complete set of engineering drawings for submitting and production, together with corresponding technical documents for this stage, should be completed.

Construction design combines the technical documents of detailed design and the production capacity of manufacturers to produce working drawings, craft equipments, documents about equipments acceptance, installation, and testing. Structural design is not always included in the construction design (Zhu 1992).

All the work above will be demonstrated and evaluated according to the timeline.

Original Conditions of Structural Design

It is a prerequisite for structural design to understand and analyze the original conditions sufficiently. The main contents of the original conditions are as follows (Yu 2010):

1. The structural design task descriptions and the data related to the overall requirements; the theoretical demands to the structures in the shape, size, properties, etc.; and the specifications and installation requirements of the payloads inside the structural system (if any).
2. The working environments of the structures. The working environments can range from storage, transportation, operation/service, to deployment or recycling environment, leading to various loads, to all of which the structures must be adapted.
3. The interface relations inside and outside of the structural system, which will pose restrictions and requirements on structures. The contours, dimensions, and weights of the structures should meet the requirements about the interface relations, space layout, and performance.
4. The production conditions of the manufacturers. In order to design structures better, the designer should know the production capacity and assembly equipments of manufacturers,

the technical level and processing experience of technicians and workers, and the possibility of adopting new materials and new technology.

5. The cost of the structures. The more complicated and higher performance of the structural system, the higher it costs. Structural design aims to achieve balance between economy, safety, reliability, and high performance.

Loading Analysis in Structural Design

Loading analysis is one of the main bases for structural design. There are various kinds of loads that affect marine structures, e.g., loads due to working environments (the wave induced loads, the hydrodynamic loads, the sloshing loads, the impact loads, and so on (Wang 2013)), gravity, loads during ground operation, loads during the launching and recovery period, etc.

The maximum working loads that a structure bears under normal service conditions is p_w , and the corresponding stress is σ_w . It is not allowed for structure that either any deformation impeding working in order under the service load or harmful residual deformation after unloading occurs. Therefore, the structural design should ensure that σ_w of the main structural components should not exceed the ratio limit of the material σ_p , that is (Zhang 2011),

$$\sigma_w \leq \sigma_p \quad (1)$$

The Calculation Load Method

The calculation load method is widely used in the actual structural design (Yu 2010). In which the calculation load is obtained by multiplying the load by the safety factor f ($f > 1.0$).

$$p_c = p_w f \quad (2)$$

And the strength criterion is:

$$p_u \geq p_c \text{ or } [\sigma] \geq \sigma_{c, \max} \quad (3)$$

Where p_u is the ultimate load of the structural material, $[\sigma]$ is the ultimate stress of a component, which can be the tensile stress σ_b or the critical stress σ_{cr} , depending on the load being

compression or tension. If $[\sigma]$ is not less than the maximum stress in the components under calculation load $\sigma_{c,max}$, the structure will be destroyed or lose the bearing capacity (Yu 2010).

The advantages of the calculation load method lie in two aspects. First, when adopting the calculation load method, the working stress of the structures are near the ratio limit of the material, and some components are allowed to work beyond the elastic range of the material, so the material can be fully used and the structures can be lighter. Second, it is convenient to observe the results of the tests and to compare the theoretical results with the test results (Yu 2010).

Safety Factor

The safety factor is associated with the calculation reliability, the materials, the manufacturing level, the maintenance, etc. The safety factor is greater than 1.0, providing safety margin in cases that the actual loads exceeding the service load and that the uncertainties in the materials and design procedure are inevitable. If the safety factor is too large, the structure system will be larger and its performance can be decreased. If it is the opposite situation, the structure may be too risky to fail during service. The safety factor should not be determined only to ensure that the structure has enough strength, stiffness, and stability, but the structure mass is not too large. The method for determining the safety factor is generally as follows (Yu 2010; Zhu 1992):

$$f = f_1 f_2 f_3 f_4 \cdot f_i \cdots \quad (4)$$

f_1 is the ratio of the strength limit to the proportion limit of material, in order that the structure does not have permanent deformation under the working load, and it is the main factor determining the safety factor.

f_2 is concerned with fatigue and permanent deformation of materials, and it is generally greater than 1 except for some disposable equipments.

f_3 mainly focuses on the influences of such errors as the calculation errors of the loads, the deviations of the material performances, the

errors of the assembly, and the errors of the measurements.

f_4 is presented to guarantee the safety of the ground crew.

And f_i is the i_{th} impact parameter to the safety factor. The factors in Eq. (4) are always greater than 1.0.

It is more feasible for the safety factor to be determined by probabilistic methods as the factors are all random. The standard safety factor is selected on the basis of a large number of data accumulations from various structures over decades.

Stress Analysis

The procedure of stress analysis is introduced in Bai (2003). Stress analysis plays an important part in structural design, predicting various behaviors of structures under service loads and working conditions. The different modes of failure are not likely to arise at the same time; in more cases, the failure occurs by one or two mechanisms. It is rather difficult to design a complicated structure of high performance merely using simple qualitative calculation, and thus the strength analysis is indispensable in structural design.

Nowadays, the major classification societies all over the world, such as ABS (ABS 2010), CCS (CCS 2013), LR (LR 1989), BV (BV 1989), DNV (DNV 1988), GL (GL 2009), RS (RS 2004), NK (NK), have provided evaluation, calculation methods, and other technical service associated for marine structures, which are one of the main kinds of business lines of the classification societies.

In the conventional strength analysis, there are always design calculation and checking calculation (Yu 2010). The design calculation is carried out in the earlier stage of structural design, and the purpose is to obtain the basic section sizes of the main force elements. The methods of design calculation should be simple and of certain accuracy. Theoretical formulas, empirical or semi-empirical formulas in elastic mechanics, structural mechanics, and material mechanics are often employed in design calculation methods, and analytical solutions might be obtained for simple structures.

The checking calculation is implemented after determining the structural details. The purpose of the calculation is to get the strength data of the whole structural system and each component to check the results of design calculation and to decide whether the design scheme need further modifying and optimizing (Yu 2010). Therefore, the accuracy of checking calculation is very important, and finite element method and non-linear analysis programmed and executed on computers are often used to get accurate results as far as possible. The finite element analysis (FEA) was proposed by Clough (1960), and its basic idea is dividing a continuous system into a finite number of elements, establishing the matrix equation about the displacement and force of every element, and seeking the solution to the matrix equation. FEA can be applied to complicated structures with irregular geometries and is easy to be programmed and executed utilizing computers, as a result, FEA has become an important approach to structural design in variety of industries.

Tests in Structural Design

It is insufficient to accurately analyze the structures merely depending on the theoretical calculation. The tests on the real structures are necessary to check the theoretical results and identify the actual structural strength.

There are many tests during the process from design to manufacturing and operation of the structures (Yu 2010). The tests can be categorized by different classification methods, e.g., static strength tests and fatigue strength tests due to different load style, ground tests and pressure cylinder tests due to different test places, component tests and structure tests due to different tested objects, the calculation load tests and the failure tests according to different loads, etc.

The History of Structural Design

The structural design develops in the wake of improvements of science and technology.

There was recording about human diving using equipments as early as in the fourth century B.C. (Zhu 1992), when the workers or artisans might rely on their life experience and intuition

to some degree rather than accurate calculations due to lack of science and technology.

Thanks to the development of mechanics and mathematics, analogy and empirical mathematical formulas were used in the calculations of structural design. The theoretical analysis methods of rods, plates, beams, trusses, shells can be found in material mechanics and structural mechanics, such as force method, displacement method, matrix displacement method, finite element method, and so on, which laid foundation for structural strength analysis (Yu 2010). The mechanics theory and abundant design practice combine to form empirical design or semi-empirical design by scientific summary and improvement, which is traditional approach to do structural design before and even after computer technology coming into being.

However, in most cases, it is unlikely to get the analytical solutions to the analysis of complex structures using theoretical methods. Thanks to the development of computer technology, revolutionary changes have been made to structural design, forming the modern design methods (Wang 2013). In modern design, system tools are used to implement the integrated design of man-machine-environment system, to make the concepts, process, and organization of design more reasonable and to optimize the process, strategy, scheme, and data selection of design (Yu 2010). The computer technology is universally used in calculation and drawing of large-scale structures, e.g., in the structural design of modern ships and offshore platforms, subbottom tunnels, whose structures are becoming larger and more complicated. Along with the concepts of reliability design, multi-objective optimal design and risk-based assessments of structures which are expensive and time costly being recognized as urgent; computer technology is playing a more and more key role in such design work because of their high efficiency and calculation capacity, saving a lot of time and money, until now there have been a number of commercial softwares which are widely used in marine engineering (Bai 2003).

The modern design methods can improve design quality and shorten the design cycle, as a

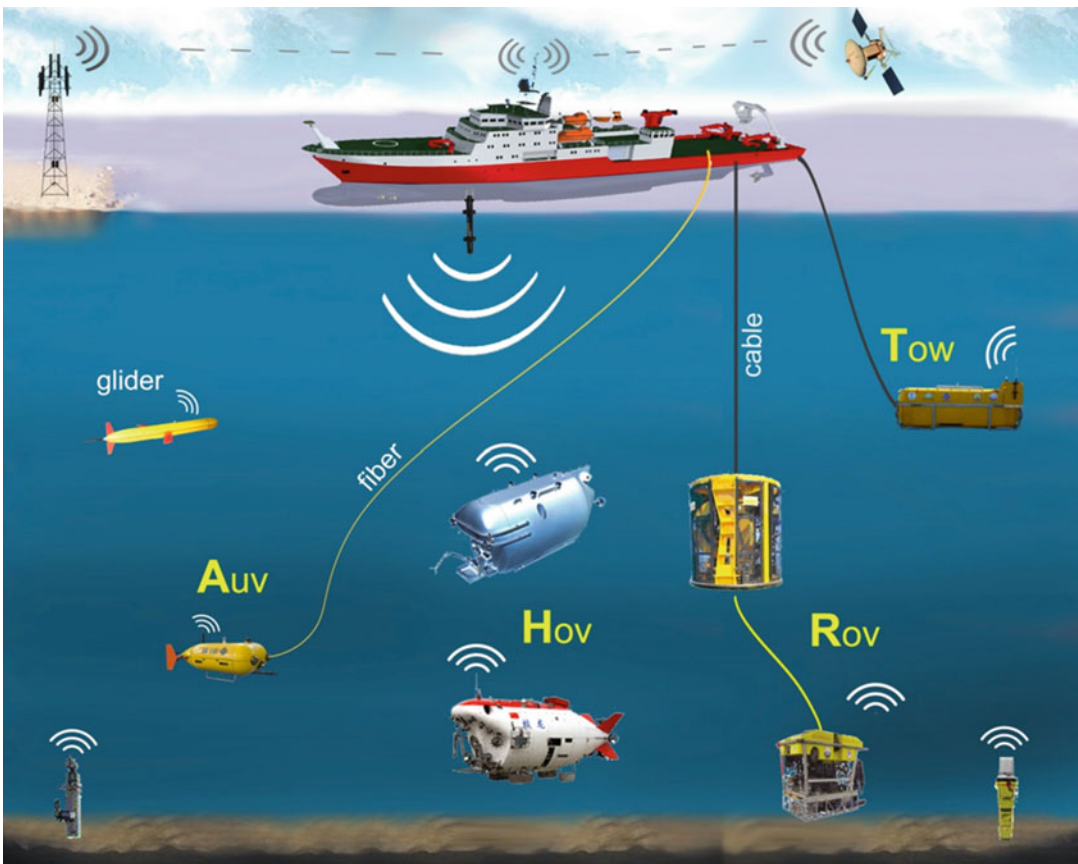
result, the efficiency is enhanced. In a manner of speaking, at present, the modern design methods are indispensable in the marine structural design.

Key Applications

Submersibles

Submersibles are important technical means for human beings to carry out deep-sea developments, see Fig. 1. They can carry various electronic devices, mechanical equipment, or even engineering technicians and scientists to reach ocean floor quickly and accurately, and conduct explorations, scientific investigations, and such work efficiently (Liu 2001). Generally, submersibles can be grouped into four kinds: human occupied vehicle (HOV), remotely operated vehicle (ROV), tethered mapping system (TMS), and

autonomous underwater vehicle (AUV). There into, New Alvin of USA, Mir I & MIRII of Russia, Nautilus of France, Shinkai6500 of Japan, and Jiaolong of China are the main operation type deep HOVs (Pan and Cui 2011), and unmanned submersibles that have reached the ocean floor in the Mariana Trench were Nereus of USA and Kaiko of Japan (Vaskov 2012). The structural system of a submersible includes pressure and non-pressure structures, and relative design methods are available in classification societies' rules. Among pressure structures, pressure vessels provide normal pressured and sealed space for equipments, apparatus, or personnel contented. The pressure vessels should be designed to be of sufficient structural strength, reliable sealing, and as light as possible. The pressure vessels are generally in the shape of cylinder, sphere, ellipsoid, or combination of shapes



Structural Design, Fig. 1 Various kinds of submersibles. (Pictured by CSSRC)

abovementioned (Liang et al. 2004). Most of the deep HOVs' manned cabins adopted the form of a single ball with multiple openings, and the strength and stability of such kind of hulls have been investigated since 1915 (Pan et al. 2012); Pan et al. have studied the ultimate strength of deep manned cabins (Pan et al. 2010, 2012; Pan and Cui 2011), and the cylindrical pressure structures are mostly used for submersibles no more than 800 m (Zhu 1992). Reliable sealing is another critical technology in pressure vessels design. Buoyancy material is also a kind of pressure structure, which provides buoyancy compensation for the submersibles. The structural design of buoyancy material needs to examine the pressure resistance performance, machinability, and water absorption properties of the material itself comprehensively. The appropriate design of buoyancy material can guarantee the buoyancy of the submersibles, increase the payload, improve the contour and size of the submersible (He et al. 2017). Ceramic matrix composites housing and buoyancy have been used in Nereus, obtaining excellent bulk density (McDonald 2014; Stachiw et al. 2006; Weston et al. 2005) which can provide positive buoyancy. The non-pressure structures which do not form closed vacuum are composed of light shells and frame mainly, and they provides a smooth shape, protecting enclosure, connection, and supporting of equipments for the submersible. The structural design is supposed to satisfy the requirements of the working environments and the task assignments, and consider the existing technological level such as processing, welding, heat treatment, composite technology of composites, and so on to control the structural deformation.

Ships and Offshore Platforms

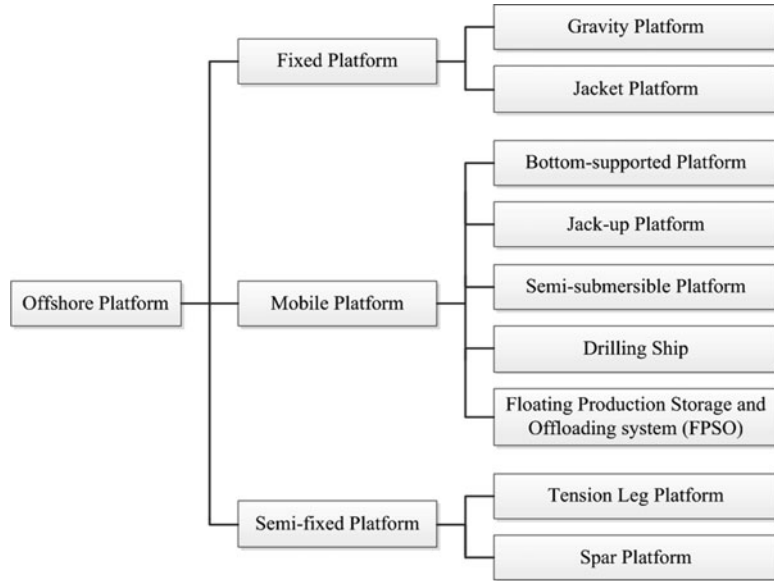
Ships and offshore platforms are important marine structures and are of significance in social economy. The main material in the construction of the marine structures is steel, which accounts for the 20–30% of the construction cost of the engineering, and the steel products include steel plate, formed steel (marine bulb bar, H-steel, angle steel, cast steel and forged steel, welding materials, and accessories (Wang 2013)). Steel plates

are mainly used to make the ship hull, angle steel and formed steel are mainly used to make the hull structures; bulb bars are mainly used for making the hull keel, and steel pipes are mainly used to make the vessels on the ships. Because of the wide application of the steel, the steel weight is about 60% of the total ship weight. Offshore platforms are large welded, trussed constructions composed of all sorts of string bars and inclined rods. High performance marine steel products are demanded to have high strength, high toughness, good weldability, corrosion resistance, and crack arrest characteristics (Yang and Su 2012).

As the utility specialization of modern ships, there are various types of ships, e.g., oil tankers, bulk carriers, container ships, et al., the scantling of which may be different from each other. With the development of marine oil and gas resources, offshore platforms are being built from shallow sea to deep sea gradually. The three motion modes offshore platforms and the main subtypes are demonstrated in Fig. 2 (Di et al. 2008), and structural forms are different more or less with types.

As for marine structures withstand very harsh working conditions, including a variety of climate conditions, wind and wave invasion, and seawater corrosion, therefore brittle fracture and fatigue damage of the structures are to be prevented, special attention should be paid to strength calculation and checking, fatigue resistance and corrosion resistance in structural design. Global or local stress analysis can be conducted using computer technologies besides the conventional methods, leading to the development of many sophisticated commercial softwares.

Welding is basic technology in such large scaled and complicated steel structures as modern ships and offshore platforms, and the technical requirements of welding are becoming higher. Due to local heating and the following uneven heat in welding, residual stress is easy to occur in welded joint areas, which makes welding parts particularly vulnerable to defects. The weld joints are more sensitive to fatigue loads, and issues such as welding materials, welding methods, welding technology, and welding inspections can all affect the structural performance; hence, the welding defects are usually reduced by

Structural Design,**Fig. 2** Different types of offshore platforms. (Drawn according to Di et al. 2008)

controlling the welding process, through selecting suitable welding material (with properties matching the parent material), preheating before welding, and heat treatment post welding (Yu 1992). In addition to the design calculation and analyses, the welding procedure is formulated, in which the rules and requirements are clearly specified so as to ensure the strength and fatigue resistance of the platform structures.

Cross-References

- [Concept Design](#)
- [Detailed Design](#)
- [Design of Submersibles](#)
- [Empirical Design](#)
- [Preliminary Design](#)
- [Structural Strength](#)
- [Technical Design](#)

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Structural Mechanics of Polar Engineering

► Structural Characteristics of Polar Engineering

Structural Strength

► Structural Design

Sub

► Submarine

Submarine

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Synonyms

[Dunker](#); [Oscar](#); [Pigboat](#); [Sub](#)

Definition

Submarine is the general name of all kinds of underwater vessels. Submarine is a kind of naval vessel that can sail and fight on the water and at a certain depth under water.

Scientific Fundamentals

Composition and Basic Type

System Composition

The submarine is mainly composed of hulls, control systems, power units, weapons systems, observation equipment, communications equipment, and navigation equipment. The equipment on board is mainly divided into five parts according to functions: observation system, communication system, navigation system, weapon system, and electromechanical system. Submarines usually use the following four types of shapes: conventional one, water-drop one, like water-drop, whale-shaped (Zhong and You 2017).

The weapon cabin includes torpedo cabin and missile cabin. Torpedo cabin of the modern submarine is generally located in the stern and used to store and launch torpedoes, missiles, and mines.

The missile cabin is located in the middle of the submarine and used to store and launch cruise missiles and ballistic missiles. The power cabin is the heart of the submarine, which is commonly close to the submarine raft and equipped with the main and auxiliary power machinery. The command cabin is the operational center of the submarine, which is located below the submarine's bridge. It has some system equipment for manipulation, observation, detection, and control. The living cabin is the place where the crew rest. It includes crew's rooms, the dining room, the conference room, and other security cabins. The lower floor of the living cabin is generally equipped with the battery, which is called the battery cabin (Zhong and You 2017).

Submarine Types

The submarine can be divided into four categories according to the displacement size: the large submarine, the medium submarine, the small submarine, and the pocket submarine (Xie 2007).

The displacement exceeding 2000 t is the large submarine, as shown in Fig. 1. This kind of submarine has greater self-sustaining power, endurance, and a large amount of weapons and equipment. On January 10, 2013, the first “Borei-class” strategic nuclear submarine named “Yury Dolgorukiy” was officially served. The submarine is defined as the medium one for the displacement between 600 t and 2000 t, which has less self-sustaining power, endurance, weapons, and

equipment than the large submarine. However, the medium submarine is more flexible and can execute the tasks in the middle sea. This submarine currently occupies a considerable proportion of conventional submarine in various countries. The displacement between 50 t and 600 t is the small submarine, which is suitable for offshore and shallow water areas. And its self-sustaining power, endurance, weapons, and equipment are all relatively small. Nevertheless, the small submarine has high flexibility and low cost advantages. Meanwhile it is convenient for mass production in wartime. The displacement below 50 t is the pocket submarine, which is used to raid the enemy's base and conduct reconnaissance activities. And the pocket submarine is usually transported by other submarines, surface ships, and helicopters, then moves by its own (Xie 2007).

The submarine is divided into open sea submarine, near sea submarine, and offshore submarine according to the navigation area. Meanwhile, this division is usually commensurate with the size of the displacement. So the open sea submarine usually refers to the large submarine, the near sea submarine refers to the medium submarine, and offshore submarine refers to the small submarine. The submarine is also divided into the attack submarine, anti-submarine, radar warning submarine, minefield submarine, and the transport submarine according to the missions (Wolfpack 2010).

Development History

The development of submarine can be traced back to the Renaissance in Europe. The first person to

Submarine, Fig. 1

Russian “Borei class”
ballistic missile submarine
trimmed on the surface
(Yu 2013)



design submarine was Da Vinci, the Italian artist. The first documented submarine researcher was an Italian, named Leonard, who proposed the theory of underwater hull structure in 1500 AD. In 1578, William Bourne, a British navy lieutenant, recorded his research on the submarine theory in his book “Inventions.” In 1620, Cornelis Drebbel, a Dutch physicist, built the first diving boat in human history (Tierie 1932), so he was called “the father of the submarine.”

The first military submarine appeared in the American Revolutionary War. David Bushnell from Yale College (now known as Yale University) built the “Turtle” (shown in Fig. 2). Only one person was allowed to operate the rudder and the propeller due to limited inner space. The “Turtle” could dive to the depth of 6 m and stay for about 30 min through the foot valve to the water tank. (Zhong and You 2017).

The early submarine was all human powered and its speed was very low. In 1863, the “Plongeur” built by French was powered by a steam engine with 59 kW (80 hp). Its speed was 2.4 knots. It was able to submerge underwater for 3 h at the depth of 12 m. In 1886, the “Nautilus” built by British was powered by the battery and had a speed of 6 knots with an endurance of about

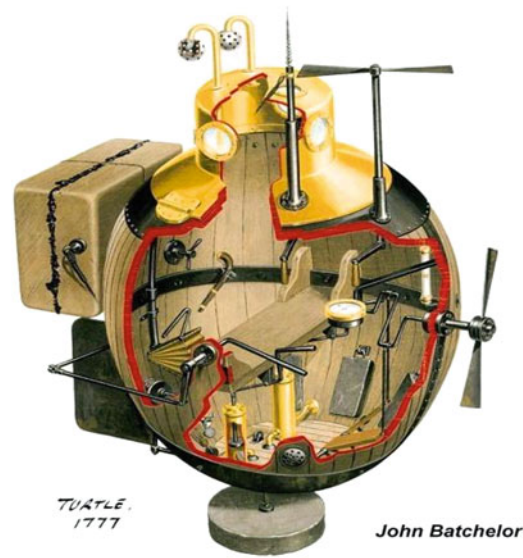
80 nautical miles. In 1880s, the United States, Britain, France, Sweden, Italy, Germany, and Russia began to study and develop new types of the submarine. It was not until 1900 that the navy of the United States began to equip the submarine (Donitz 1958).

In the late 1880s, the development of the submarine attracted the interests of more countries. In 1893, the “Gustav Zede” with a length of 45.7 m and a displacement of 266 t was launched in France. It was powered by the motor propellers. At that time, it was the most advanced one in the world (Bowers 1999).

At the end of nineteenth century, an American youth Simon Lake got the inspiration from the science fiction “20,000 Leagues under the Sea” and built the “Argonaut” based on its own power. The submarine sailed from Norfolk to New York, so it became the first one on a long voyage in human history (John and David 1986).

At the beginning of twentieth century, the equipment and performance of submarine were gradually improved with certain actual combat capabilities. The submarine with double shells was powered by the dual-propulsion and diesel-motor system. It had a displacement of several hundred tons and good seaworthiness. The endurance of it was significantly improved (Parker 2007).

At the beginning of the World War I, the submarine quickly became a new star on the sea battlefield. The number of submarine in various countries is shown in Table 1. In 1914, the “German U-9” (shown in Fig. 3) sank three British 10,000 t cruisers at the slight cost of six torpedoes in just over an hour, which strongly verified the



Submarine, Fig. 2 The diagram of the internal structure of the “Turtle” (Zhong and You 2017)

Submarine, Table 1 The number of submarines in major countries on the eve of the World War I in 1914 (Zhong and You 2017)

Nation	In service	In the construction
Britain	77	32
France	45	25
America	35	6
Germany	28	13
Russia	28	2
Italy	18	2
Japan	13	2

Submarine, Fig. 3

Germany U-boats
(Williams 2008)



submarine's combat effectiveness (Douglas 1999).

After the World War I, the major naval countries paid more attention to the construction and development of the submarine. The submarine not only acted as the defensive force, but also was assigned to various offensive tasks. The number and type of the submarine were increasing, some had more than 2000 t of displacement, 100–200 m of diving depth, 7–10 knots of underwater speed, 16–20 knots of surface speed, more than 10,000 nm of endurance, 1 or 2 months of self-sufficiency, 6–10 torpedo tubes, more than 20 torpedoes, and 1–2 guns. In the later part of the war, the submarine was equipped with radar, radar reconnaissance instruments, and self-guided torpedoes (Delgado 2006).

After the World War II, the navies of various countries in the world began to attach importance to the development of the new submarine. The use of nuclear power and strategic missiles brought the development of the submarine to a new stage. In 1955, "Nautilus," the world's first nuclear-powered submarine built by the United States (shown in Fig. 4) was in service. Its underwater speed and the time of endurance increased (Preston 2009). The emergence of ballistic missile nuclear submarines caused a fundamental change in the role of submarines. It had become a strategic nuclear strike force under the water (The Submarine Turtle 2008).

After 1980s, new conventional submarine was designed and built by Soviet (Russia), Germany, Sweden, the Netherlands, Japan, and other countries. The new Air Independent Propulsion submarine without air propulsion was developed quickly, such as the Russian "Lada-class," Germany's "212-class" and "Export 214-class," Sweden's "Gotland-class," Japan's "Soryu-class". It had gradually become outstanding representatives and development trend of nonnuclear-powered submarine in the world (Zhong and You 2017).

Key Technology

The power equipment used by the conventional powered submarine is the diesel-electric power equipment and the Air Independent Propulsion (AIP) equipment (Yu 2013).

The diesel-electric power unit is the earliest and most widely used in the development history. It is mainly composed of a diesel engine, a battery, and a main motor. Fig. 5 shows a conventional submarine using a diesel-electric power unit. The diesel engine is the main power unit of a conventional submarine for the surface navigation with the speed of 10–15 knots. The main motor is the main power unit of a conventional submarine for underwater navigation with the speed of 15–20 knots driven by the battery. The diesel-electric power unit is also equipped with economical

Submarine, Fig. 4

“Nautilus,” the world’s first nuclear powered submarine (Yan 2012)

**Submarine, Fig. 5**

German “206” conventional submarine uses diesel-electric power unit (<http://www.baidu.com/>)



electric motors as an auxiliary power with the underwater speed of 2–4 knots. However, due to the limitation of the battery power, the submarine needs to float to the snorkel for a period of time and use the diesel generator to set up battery for charging. At this time, the submarine is easily discovered and attacked. For this problem, the Air Independent Propulsion (AIP) came into being. The AIP was originally developed in Sweden and used the oxygen carried by the submarine to complete the conversion of energy, which

provided the propulsion power for submarine under the water. Comparing with ordinary conventional submarines, the devices equipped with AIP could stay underwater for at least a week. In addition, the AIP is more secluded. The operational effectiveness of the AIP is already close to nuclear submarines (Yu 2013). In most countries, the navy is unable to purchase nuclear submarines because of military spending and restrictions on combat waters, so the AIP is a good choice for them.

The nuclear-powered submarine emerged due to the great advantages of the nuclear power. And the power system of the nuclear powered submarine is composed of nuclear reactors, steam generators, pressurizers, main circulation pumps, and main steam turbines. Utilizing the nuclear power as driving force, the power of submarine has undergone a fundamental transformation. It could reach tens or hundreds of megawatts and continuously work for many years with one loading. Its endurance can reach hundreds of thousands of sea miles or even greater. Because the nuclear power unit does not rely on the air, the submarine can sail underwater for a long time and has good concealment, which has become one of the most important strategic weapon platforms in the new era. Nuclear power unit is usually used as the main power unit for nuclear submarine. Nuclear submarine is also installed auxiliary power devices, such as diesel generator sets, power storage units, and electric motors, which provide emergency power for nuclear submarines when needed (Ross 2012).

The submarine is usually equipped with a variety of weapons. At present, the submarine's weapon system mainly includes various missiles, torpedoes, and mines. Missiles are the main combat weapons of modern submarine including ballistic missiles, cruise missiles, anti-ship missiles, anti-submarine missiles, etc. The submarine carries different missiles for the different tasks. Torpedo is a traditional submarine weapon and used to attack and counter attack against boats. Strategic nuclear submarine uses intercontinental ballistic missiles as weapons. Attack nuclear submarine uses various cruise missiles and anti-submarine missiles as weapons. For the strategic nuclear submarine, the ballistic missile carried by the submarine has the range of 10,000 km and the missile can carry the nuclear warhead. So the strategic nuclear submarine is the symbol of national deterrence. So far, only the United States, Russia, Britain, France, and China have strategic nuclear submarine in the world, such as American "Ohio-class" strategic nuclear submarine, Russian "Typhoon-class" strategic nuclear submarine, French "Triomphant-class" strategic nuclear submarine, British "Vanguard-class" strategic nuclear

submarine, and Chinese "Type 094" strategic nuclear submarine. For the attack nuclear submarine, the main task of it is to search information and attack the enemy submarine, aircraft carrier combat groups, and surface warship formations, so that the strategic missile submarine, aircraft carrier combat groups, or marine mobile formations, as well as strategic campaign reconnaissance, have the safe channel. Meanwhile, the attack nuclear submarine can conduct the maritime escort and destroy the enemy traffic lines. Attack nuclear submarine and strategic missile submarine are mainly distributed in the United States, Russia, Britain, France, and China. The United States is the first country to develop attack nuclear submarine. The latest US attack nuclear submarine is the "Virginia-class" nuclear submarine. The fourth generation of Russian attack submarine is the "Type 885 Severodvinsk." The newest British nuclear submarine is the "Astute-class" attack nuclear powered one, which uses advanced floating raft damping technology. French newest attack nuclear submarine is "Amethyste-class," which is produced to meet the new operational needs of the French navy. The Chinese attack submarine is the "Type 091" and "Type 093" (Zhang 2014).

The pressure-resistant hull structure is designed to withstand the high water pressure in the deep sea for the submarine. Meanwhile, the submarine needs to wrap all or part of it with a nonpressure hull structure. Therefore, the submarine must have a smooth surface to reduce the resistance. The hull structure of modern submarine can be divided into four types: the single-shell submarine, the half-shell submarine, the double-shell submarine, and the single-double-mixed shell submarine, depending on the degree of wrapping. The single-shell submarine has only one layer of pressure-resistant shell. The main ballast tank is arranged in the pressure hull. And the submarine has less water flow holes at the outer surface, good smoothness of the hull, small wet surface area, low resistance encountered during underwater navigation, and rapid submersible speed (Yu 2013). The single-shell structures are mainly used in European and American, such as the United States "Sea wolf class SSN" attack

nuclear submarine, the British “Trafalgar-class SSN” attack nuclear submarine, the French “Rubis class” attack submarines, German “209” and “214,” and Swedish “Gotland” conventional submarines. The half-shell submarine has a layer of coating around the pressure hull as the half shell, which is a double shell area with pressure and nonpressure tanks. The bottom of the pressure hull is still exposed to the outside. Compared with the single-shell structure, the internal space and external line pattern of the half shell are improved. From the development history, the submarine with half shell structure is less. The submarine with the inner shell and the outer shell is the double-shell submarine. The inner shell is a pressure hull made of high-strength steel or alloy material, which is all wrapped by nonpressure outer shell. The main ballast water tanks, fuel tanks, fuel ballast tanks, and buoyancy control tanks of the submarines are located between the two shells (Yu 2013). The “Typhoon-class” strategic nuclear submarine with “Type 941” (shown in Fig. 6) is the largest ballistic missile submarine of the Russian navy. It is also the world’s largest volume and tonnage submarine record holder up to 2017.

The double-shell structures are used in the former Soviet Union, Russia, China, and Japan. The typical double-shell submarine is the “Oscar-class” and “Typhoon-class” nuclear submarine. The single-double-mixed shell submarine appeared in 1950s. It can reduce the displacement of submarine and the wet surface area of the hull, and increase the underwater speed and the ability

to fight at critical locations. With the rapid development of technology, the design of the shell structure has become more and more mature. Instead of the traditional single-shell and double-shell structure, the mixed-shell gradually becomes the mainstream (Yu 2013).

Key Applications

In the twenty-first century, with the continuous development of economy and the advancement of technology, the development of the submarine in various countries was in boom times, the development of submarine had created a wave of competition in quantity and quality all over the world.

On November 9, 2012, the “Tourville,” which was the first pressure-resistant shell section and the third French “Barracuda-class” attack nuclear submarine, was built by the “DCNS.” Meanwhile, there would be six “Barracuda-class” nuclear submarines to be delivered to the French Navy from 2017 to 2027, and they would replace the current four “Rubis class” nuclear submarines and two modified “Amethyste class” nuclear submarines.

Russian submarine, at the beginning of 2012, the Russian Ministry of Defence decided to sequentially build 13 “Oscar class” nuclear submarines (shown in Fig. 7) and update the existing ones.

At present, the Russian navy has equipped 27 nonstrategic attack nuclear submarines, 9 guided missile submarines, and 18 multipurpose nuclear submarines. Among the guided missile

Submarine, Fig. 6 The “Typhoon-class” ballistic missile submarine is the largest submarine in the world (Yu 2013)



Submarine, Fig. 7

Russian “Oscar-class”
nuclear submarine ([http://
www.baidu.com/](http://www.baidu.com/))



“Song-class” attack conventional power
submarine.



“Jin-class” nuclear power submarine.

Submarine, Fig. 8 Chinese submarine (<http://www.baidu.com/>)

submarines, there are eight “Oscar II” submarines and only one “Type 885 Yasen class” submarine named “Severodvinsk.” Among the multipurpose nuclear submarines, there are 11 “Akula-class” submarines with “Type 971,” 2 “Sierra Class I” submarines, 2 “Sierra Class II” submarines, and 3 “Victor-class” multipurpose nuclear submarines. “Ohio-class” nuclear power submarine has a length of 171 m, a width of 13 m, and a displacement of 18,750 t. The submarine has been in service for 30 years.

The British Navy will own 4 “Vanguard-class” strategic missile submarines and

7 “Astute-class” attack nuclear submarines, which have been planned to extend the life to around 2028. At the same time, the Trident missile’s “Vanguard-class” in service will be replaced by the new generation of nuclear submarines, which is scheduled to serve the navy in 2028 (Zhong and You 2017).

For China, there are two kinds of submarines: the attack conventional power submarine and the nuclear power submarine. And the total number of submarines built is over 130. The “Ming-class” (Type 035) and “Song-class” (Type 039) submarines are the attack conventional power ones. The



Submarine, Fig. 9 “Columbia-class” nuclear power submarine. (http://en.wikipedia.org/wiki/Main_Page)

“Ming-class” (Type 035) is the first generation of conventional submarine. The “Song-class” (Type 039) is the second generation of conventional submarine and it achieves launching an anti-ship missile from a distance underwater firstly (Fig. 8).

The “Han-class,” “Xia-class,” “Shang-class,” and “Jin-class” submarines are the nuclear power ones. The “Han-class” (Type 091) is the first generation of attack nuclear power submarines in China. The “Xia-class” (Type 092) is the first generation of ballistic missile nuclear power submarine. And the “Shang-class” (Type 093) is the second generation of attack nuclear power submarines designed and built by China, entering service in 2006, which has the fast underwater speed, good concealment, and a powerful blow. Meanwhile, the “Jin-class” (Type 094) is the second generation of the ballistic missile nuclear power submarine, which has the largest displacement in Chinese submarine and is improved greatly in concealment, sensor, and propulsion system reliability.

The US Navy decided to convert the SSGN-726, the SSGN-727, the SSGN-728, and the SSGN-729 into a cruise missile nuclear submarine carrying conventional guided missiles. The “Columbia-class SSBN,” (shown in Fig. 9) the fifth generation of the United States ballistic missile nuclear submarine, is used to replace the



Submarine, Fig. 10 “Ohio-class” nuclear power submarine. (<http://image.baidu.com/>)

“Ohio-class” nuclear power submarine (shown in Fig. 10). This submarine with a length of 171 m, a width of 13 m, and a displacement of 20,810 t will be the largest submarine built by the US Navy. The research process of this submarine is quite smooth. It was formally approved in January 2017. And the submarine construction is



Submarine, Fig. 11 “Virginia-class SSN” nuclear power attack submarine. (<http://image.baidu.com/>)

planned to start in 2021 and scheduled to come into service in 2031. The “Virginia-class” SSN submarine (shown in Fig. 11) is the nuclear powered fast attack one of the United States Navy. Considering the time in service and the standard of construction of the American nuclear powered attack submarine, it is the seventh generation. But from the perspective of the technical features and the purpose, it belongs to the fourth generation.

This submarine with multipurpose tasks is used to replace large numbers of “Los Angeles-class SSN” in service. The “Los Angeles-class SSN” has a full length of 110 m, a width of 10 m, a full-load displacement of 6927 t. Its speed is 32 knots. The “Virginia-class SSN” with anti-submarine capability has gradually become the main force for the naval warfare in twenty-first century.

Cross-References

- [Ship Electrical Design](#)
- [Ship Machinery Design](#)
- [Ship Structural Design](#)
- [Special Marine Vehicle](#)

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Submarine Pipeline

- [Pipeline Soil Interactions](#)

Submarine Rescue Bell

► Rescue Bell

Submarine Rescue Vehicle (SRV)

► Deep Submergence Rescue Vehicle (DSRV)

Submerged Depth

► AUV/ROV/HOV Hydrostatics

Submersible

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Synonyms

Autonomous and remotely operated vehicle (ARV); Autonomous underwater vehicle (AUV); Deep submergence rescue vehicle (DSRV); Human occupied vehicle (HOV); Manned submersible (MS); Remotely operated vehicle (ROV)

Definition

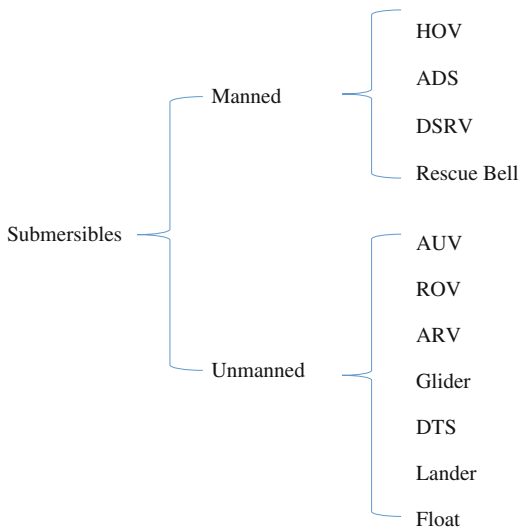
A *submersible* is a small vehicle designed to operate underwater. The term *submersible* is often used to differentiate from other underwater vehicles known as submarines, in that a submarine is a fully autonomous craft, capable of renewing its own power and breathing air, whereas a

submersible is usually supported by a surface vessel, platform, shore team, or sometimes a larger submarine. In common usage by the general public, however, the word *submarine* may be used to describe a craft that is by the technical definition actually a submersible (<https://en.wikipedia.org/wiki/Submersible>).

Scientific Fundamentals

Submersible Types

There are many types of submersibles, including both manned and unmanned vehicles. A division of various submersibles is given in Fig. 1. These include autonomous underwater vehicles (AUVs), remotely operated vehicles (ROVs), autonomous and remotely operated vehicles (ARV) or hybrid of ROV and AUV (HROV), glider, deep tow system (DTS), lander, float, human occupied vehicles (HOVs), atmospheric diving suit (ADS), deep submergence rescue vehicle (DSRV), and rescue bell. The detailed introduction to these various types of submersibles will be given in each entry of this encyclopedia, and in this entry, only the information about the general concept of submersible is given. Submersibles have many uses worldwide, such as oceanography, underwater archaeology, ocean exploration,



Submersible, Fig. 1 A division of various submersibles

adventure, equipment testing, maintenance and recovery, and underwater videography. Various underwater submersibles are the main working force for the research fleet. The users of these vehicles include offshore petroleum, telephone cable maintenance, science, surveying, salvage, and military.

Historical Development

Human curiosity for the natural world and human exploitation of ocean resources are the leading two factors for the progress of deep ocean exploration. People have dived into the sea to collect pearls from pearl oysters since very ancient times, but the depths of the oceans were beginning to be known only in the nineteenth century. The British survey ship *Challenger II* surveyed the Mariana Trench near Guam in 1951 and identified the deepest known point in the ocean at 10,912 m. This was named the Challenger Deep. However, as technology improves in accuracy and precision, the actual depth value for the Challenger Deep may vary. In the middle of the nineteenth century, British biologist, Edward Forbes, made the infamous statement that life did not exist at depths greater than 600 m. However, this statement was soon found to be wrong, and it inspired scientists to explore the oceans deeper and deeper. The existence of life in the deep sea was established in the late 1860s (MTS 2009).

Many early ideas in these two directions about the deepest ocean depth and deepest depth for life existence led primarily to the development of the human occupied vehicles (HOVs). Forman and Kohnen presented the early developments from the 1930s to 1960s and the later results from the 1960s to 2000s, respectively, of human exploration of the deep seas (MTS 2009). In the 1930s, William Beebe (1877–1962) and Otis Barton (1899–1992) initiated manned deep-sea exploration using a very small steel sphere named Bathysphere dangled in the ocean by a cable. It was the beginning of the modern era of humans entering the deep abyss in a pressure-resistant hull.

Manned observations using the Bathysphere in the deep sea rewarded the researchers with views of live marine life. However, it cannot dive to a

very deep depth due to the self-weight of the cable. Auguste Piccard (1884–1962) was the first to design a submersible without the use of steel cables. In 1952, he began building the famous bathyscaph *Trieste*. In 1958, *Trieste* was purchased by the Office of Naval Research (ONR) of the USA, and plans for Project Nekton started. The main purpose of Project Nekton was to reach the bottom of the Challenger Deep. On January 23, 1960, the bathyscaph *Trieste*, piloted by Jacques Piccard and Don Walsh, made one successful dive to the depth of 10,913 m of the Challenger Deep. However, in considerations of the length of day, sea state, and stirred cloud of sediment, *Trieste* stayed at the bottom for only 20 min. No photographs of the sea floor were taken (Piccard and Dietz 1961).

During the same period, the French navy also took great interest in building a deep manned submersible. The objective of *Archimede* was to reach 11,000 m. By 1959, construction and engineering work on *Archimede* were fully engaged. The scientific history of *Archimede* was summarized by Alan Jamieson who suggested that the reports from *Archimede* were far more valuable at the time than was realized (Jamieson 2015). However, in its whole life, it did not fulfill its original mission to reach the Challenger Deep.

Both the *Trieste* and *Archimede* submersibles are regarded as the first generation of deep submersibles to distinguish the other operational deep manned submersibles (MSs) such as Alvin of the USA (4500 m), Nautilie of France (6000 m), MIR 1 and MIR 2 of Russia (6000 m), Shinkai 6500 of Japan (6500 m), and Jiaolong of China (7000 m) (Cui 2013). The main goal in the 1950s for most deep dives was to set record depths, whereas in the 1960s, after reaching the Challenger Deep in the Mariana Trench, the development of manned submersibles focused more on the solution of certain practical and scientific problems. Now, the development of a second-generation manned submersible for full ocean depth (FOD) is the current challenge.

In the same process for some people diving deeper and deeper, other people started to explore the unmanned submersible called remotely operated vehicle (ROV). It was hard to define exactly who to credit with developing the first ROV;

however, there are two who deserve credit. The PUV (Programmed Underwater Vehicle) was a torpedo developed by Luppis-Whitehead Automobile in Austria in 1864, while the first tethered ROV, named POODLE, was developed by Dimitri Rebikoff in 1953 (http://www.rov.org/rov_history.cfm). The US Navy is also credited with advancing the technology to an operational state in its quest to develop robots to recover underwater ordnance lost during at-sea tests. They have funded most of the early ROV technology development in the 1960s into what was then named a “Cable-Controlled Underwater Recovery Vehicle” (CURV). This created the capability to perform deep-sea rescue operation and recover objects from the ocean floor, such as a nuclear bomb lost in the Mediterranean Sea after the 1966 Palomares B-52 crash and then saved the pilots of a sunken submersible off Cork, Ireland, the *Pisces* in 1973, with only minutes of air remaining. The next step in advancing the ROV technology was performed by commercial firms that saw the future in ROV support of offshore oil operations. Building on this technology base, the offshore oil and gas industry created the working class ROVs to assist in the development of offshore oil fields. Two of the first ROVs developed for offshore work were the RCV-225 and the RCV-150 developed by HydroProducts in the USA. Many other firms developed a similar line of small inspection vehicles (Christ and Wernli 2014). More than a decade after they were first introduced, ROVs became essential in the 1980s when much of the new offshore development exceeded the reach of human divers. Since then, technological development in the ROV industry has accelerated, and today ROVs are performing numerous tasks in many fields. Their tasks range from simple inspection of subsea structures, pipelines, and platforms to connecting pipelines and placing underwater manifolds. They are used extensively both in the initial construction of a subsea development and the subsequent repair and maintenance. With ROVs working as deep as 11,000 m in support of ocean exploration and research (Takagawa 1995), the technology has reached a level of cost-effectiveness that allows organizations from police departments to academic institutions to

operate vehicles that range from small inspection vehicles to deep ocean research systems (Teague et al. 2018).

Due to the cable restriction to the submersible movement, people in the 1980s started to explore the concept of an untethered remotely operated vehicle (UROV) (Griffiths 2012). This was the beginning of the development of autonomous underwater vehicle (AUV). Gwyn Griffiths (2012) provided a good account of the historical development process of AUVs, while Wynn et al. (2014) provided a good introduction about the current technology status and applications in marine geoscience.

Since the 1990s, many other new concepts of hybrid unmanned submersibles (Bowen et al. 2009), gliders (Rudnick 2015), and floats (Richardson and John 2001) were proposed and received fast development.

Key Technology in the Development of a Submersible

In the development of the various submersibles shown in Fig. 1, the most challenging equipment is the development of the FOD HOV since almost all of the technological challenges encountered in the development of unmanned submersibles are covered by the HOV.

In order to develop a more capable HOV for ocean scientists, in 1999, the deep submergence scientific community responded to a questionnaire circulated by Woods Hole Oceanographic Institution (WHOI) on improved submersible capabilities. Based on these results, WHOI developed some preliminary specifications for an improved HOV (NRC 2004):

- Increased depth capability
- Increased ascent or descent rates in order to reduce the transit time
- Increased visibility
- Increased sphere volume for personnel comfort
- Increased external science payload
- Increased battery capacity
- Automated position keeping in all axes
- Enhanced controls and monitoring for single-pilot control
- Pan and tilt video for each observer

By considering these requirements and current available technology, various persons (MTS 2009; Cui et al. 2014) have carried out a preliminary concept design of the FOD HOV, and the following key technologies have been identified.

The whole manned submersible may be divided into seven subsystems (Cui et al. 2014), and they are (1) the overall performance and general arrangement, (2) structural subsystem, (3) mechanical subsystem, (4) electric subsystem, (5) control subsystem, (6) acoustic subsystem, and (7) life-support subsystem. The technical problems to be solved for each subsystem in the development process are briefly introduced here.

The overall performance and general arrangement subsystem includes the hydrodynamic performance, general arrangement, and outfitting. In order to have at least 4–6 h bottom working time, descent and ascent times ideally need to be kept to 1–2 h (MTS 2009). Hence, the vertical descent speed needs to be about 1.5–3 m/s; however, this speed is almost an order of magnitude higher than that of a second-generation HOV. Therefore, the hydrodynamic performance has become a key technology to be solved in the development of the FOD HOV. Conceptually speaking, there are several means to increase the speed; hydrodynamical shape optimization; powered driving, flying, or gliding principle; resistance reduction by bubble generation or super-cavitation technique; or a mix of these.

The structural subsystem includes the manned cabin, small pressure spheres and cylinders, main framework, buoyancy material, composite light shells, and stabilized fins. In this subsystem, the most challenging problem is the design and manufacture of the manned cabin (Cui et al. 2015). If the traditional three-person sphere with an internal diameter of 2.1 m is designed, the required wall thickness would be 118 mm if the traditional titanium alloy Ti-6Al-4V ELI is used and this is too thick to be manufactured. Thus, higher-strength titanium alloy of 1100 MPa class or steel such as the maraging nickel steel of 1700 MPa class should be used. Now people are also exploring the use of high-strength glass or composite material (MTS 2009) for the manned cabin.

In order to be competitive against future unmanned systems with cameras evolving fast, the currently used viewports in the manned cabin are a limitation for transparency. Full-view, hemispherical, clear glass domes would be the ideal solution for transparency. Deepsearch explored the feasibility of a full glass personnel sphere, melding two areas of innovation (new material and large viewports) into one big undertaking (MTS 2009). Glass has been proven to withstand great ocean depths as evidenced by the widely used Benthos spheres and newer Vitrovex spheres. The virtues of unimpeded views are well recognized in shallower depth manned submersibles. Deep Rover, DR1002, Johnson Sea Link, SeaMagine, Deep-See, and others have proven the concept with acrylic spheres. However, due to its brittle nature, technical problems such as ease of manufacture, composition and coatings, testing protocols, hatch and penetrators, reliability, mounting in a frame, and even testing and safety need to be carefully studied. Scale model spheres should be fabricated. Pressure tests will determine seal interactions at the glass interface. Cycle testing of the scale models will be undertaken in a pressure chamber to $FOD \times 1.25$. Strategies for sphere retention will be tested as well for shock loading protection, practical serviceability, and human factors (MTS 2009).

Buoyancy material is another important structural system in a deep-sea manned submersible. Through the preliminary analysis (Cui et al. 2018), it is thought that the current solid buoyancy material of glass microballoon/epoxy resin can be optimized to the maximum collapse strength. This requires that one needs to use the buoyancy material exceeding the rule requirement in the full ocean depth manned submersible. One of the safety measures used by this exceeding the rule requirement is to shorten the life span. Yet how to determine the life of the buoyancy material scientifically and reasonably is a key question to be answered. What kind of testing methods and testing standards should be used to determine whether the buoyancy blocks in a submersible need to be replaced or continue to be used? One still needs to have an in-depth understanding of the

characteristics of buoyancy material's water absorption property.

Syntactic foam can be used but at a size and weight penalty for the design. In order to have lighter buoyancy material, one can use ceramic spheres, which have been applied in the unmanned FOD submersible *Nereus* (Bowen et al. 2009). However, one needs to solve the sympathetic failure problem (MTS 2009). The loss of the *Nereus* in May 9, 2014, further implied the significance of this issue.

The mechanical subsystem includes the main ballast water tank, the adjustable ballast system, the trim adjustment system, the propulsion system, the hydraulic system, the sampling system, the underwater emergency jettisoning system, and the emergency rescue buoy. For this subsystem, only if one wants to use a high pressure sea water pump, will there be some technology challenge in the development. However, one can use other options to realize the buoyancy adjustment function.

The electric subsystem includes electric power system such as batteries, electric power distribution system, lighting, video system, and VHF communication. The safe application of the lithium ion battery which has been used in *Shinkai6500* should be explored for the FOD HOV. The main concern is surrounded with handling, packing, charging protocols, and testing in oil-filled housings.

Observations and documentation of observations are key to the HOV capabilities required by the scientific community. The latest illumination is to apply all the lights with LED arrays which can provide twice the amount of illumination currently available from the HMI lights and also improve the beam pattern and uniformity of the illumination field. The internal video infrastructure is the foundation for present and future upgrades to imaging system components. The internal video system accommodates all external camera signals and internal computer displays and distributes them to recorders, monitors, and data overlay subsystems. The implementation of 3D HD video cameras would be an interesting option. It uses the submersible's master timing controller to synchronize cameras and lights as necessary. It includes:

- Camera command and control system: Controls all external camera and lens functions, and provides pan and tilt, zoom, and focus control. It also provides acquisition of still images or frame-grabs.
- Video routing system: The router interfaces the camera, recorder, and computer display source feeds to the monitors and recording devices in the personnel sphere.
- Recording systems: In-hull HDTV recorders with time code and auxiliary data input capabilities.
- Monitoring and display system: Consists of the displays used by the pilot and observers when controlling the external cameras.
- Synchronization and time code generation system: Provides synchronization and time signals as necessary to all imaging hardware in the personnel sphere as well as the external cameras and lighting system.

The command and control subsystem is that part of the submersible that allows the pilot (and potentially the observers) to interact with and control components external to the personnel sphere, as well as components that require integration of multiple inputs and outputs. Other tasks of the command and control system are to automate repetitive tasks and to provide assistance to the pilots and observers in safely accomplishing their tasks. For example, dive activities are simplified through automated heading, altitude, depth, and position-keeping capabilities, which will be particularly helpful during manipulation activities. In addition, it will provide more flexibility in interfacing with instruments both carried by the submersible and installed on the seafloor, as well as greater bandwidth for scientific data collection and logging. So the subsystem includes the navigational control system, the equipment control system, the information display and control system, the navigation and positioning system, and the surface monitoring system.

The acoustic subsystem includes an underwater acoustic communicator, the underwater acoustic telephone, the acoustic positioning system, high-resolution side-scan sonar, acoustic Doppler velocity log, anti-collision sonar, imaging sonar,

long-distance ultrashort baseline positioning sonar, and long baseline positioning sonar. The acoustic equipment can be ordered from commercial companies.

Until recently, live transmission from HOVs at depth was limited chiefly to through-water acoustics with a bandwidth too narrow for anything but voice and position data, and low-resolution frame grab images. The principal problem with acoustic transmission to the surface is stratification of the water column. One solution is to transmit to an acoustic receiver (modem) at depth that is connected by wire or optical fiber to the surface. Even more promising is the use of an optical fiber that links the HOV to the surface. This fiber would provide for a direct, high-bandwidth link to the surface without the problems of acoustic transmission, and with reduced drag and handling problems compared to thicker ROV tethers. High-bandwidth communications from HOVs will improve their scientific productivity by involving many more researchers in the real-time experience, just as ROVs do. Similarly, AUVs will benefit from real-time human presence in the control loop. So in the new FOD HOV, the use of a micro-optical fiber similar as that used in *Nereus* is worth to be explored (Bowen et al. 2009).

The life-support subsystem includes the oxygen provision system, the carbon dioxide absorption system, the water vapor absorption system, and the odor removal. Up to now, no manned submersible has been reported to be equipped with a toilet. That is the main reason why the manned submersible could not work longer than 12 h. If this problem can be solved, people can work as long as they like in a dive. Theoretically speaking, this is not a very difficult problem nowadays.

Key Applications

Reaching the Challenger Deep

Manned and unmanned submersibles reaching the Challenger Deep are always an important event, and up to now, only two manned and four unmanned submersibles have finished that feat. On January 23, 1960, the bathyscaph *Trieste*,

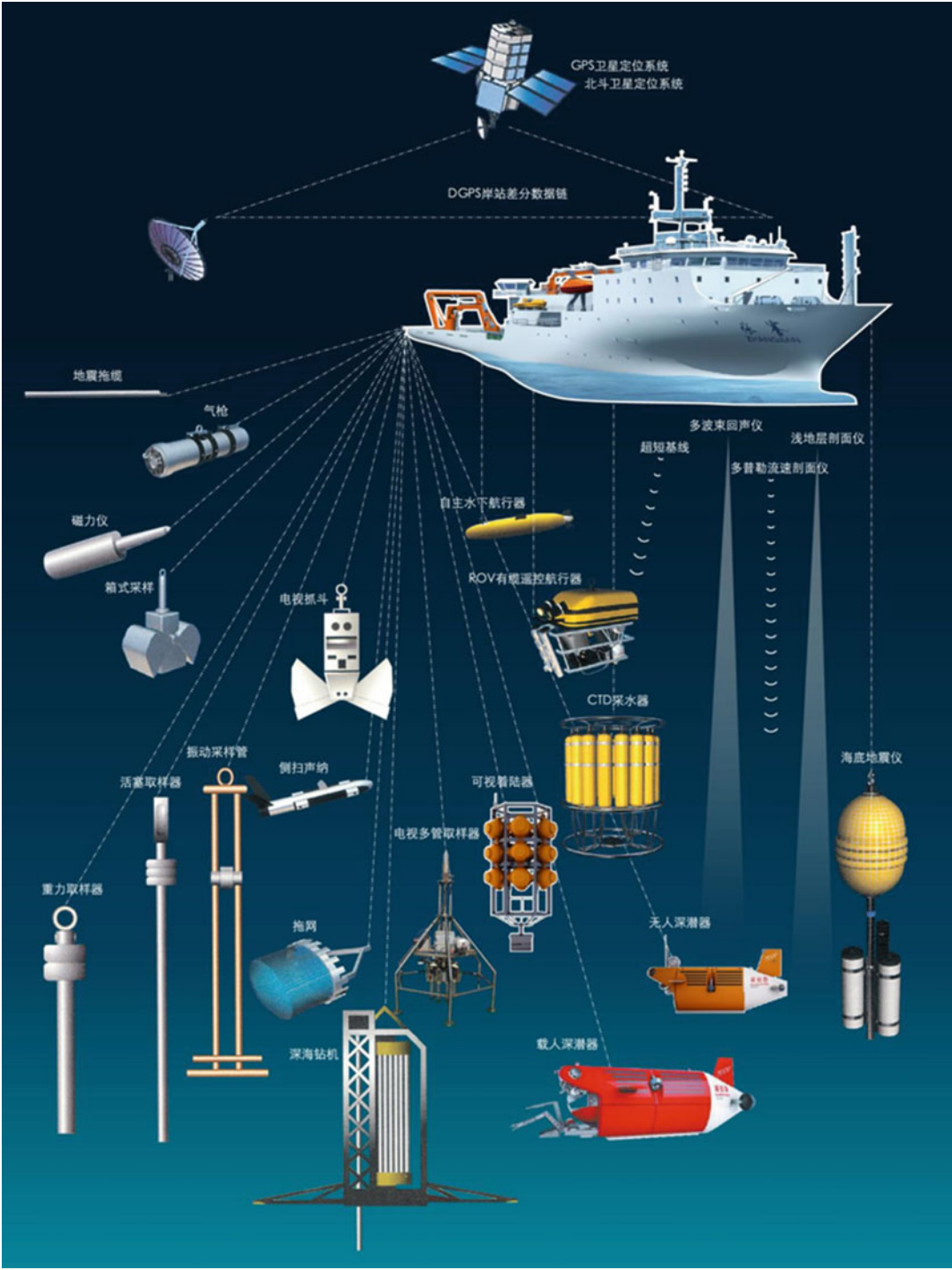
piloted by Jacques Piccard and Don Walsh, made one successful dive to the depth of 10,913 m of the Challenger Deep (Piccard and Dietz 1961). In Japan, the first full ocean depth ROV “*Kaiko*” was successfully developed in 1995 (Takagawa 1995). In addition to marking the deepest point on Earth, *Kaiko* achieved more than 20 scientific dives to trenches before being tragically “lost at sea” in 2003. Since the turn of the twenty-first century, two more unmanned vehicles have been constructed: the *ABISMO* vehicle by JAMSTEC (MTS 2009) and the *Nereus* vehicle by WHOI (Bowen et al. 2009). In 2009, the hybrid AUV/ROV vehicle *Nereus* reached the Challenger Deep four times. Unfortunately, *Nereus* was lost during its 6-mile dive to



Submersible, Fig. 2 Extraordinary, unexpected life forms discovered by Alvin in the Galápagos Rift in 1977 (www.whoi.edu)



Submersible, Fig. 3 The B28RI nuclear bomb, recovered from 2,850 ft (870 m) of water, on the deck of the USS *Petrel*. (Picture from https://en.wikipedia.org/wiki/1966_Palomares_B-52_crash)



Submersible, Fig. 4 An economic setup for a research vessel. (Picture from Rainbowfish company leaflet)

the Kermadec Trench in May 9, 2014 (<http://web.who.edu/hades/a-sad-day/>). The movie director James Cameron funded a secret project to develop the full ocean depth solo manned submersible *Deepsea Challenge*. In March 26, 2012, he took his privately built one-man submersible 10,898 m down to the Challenger Deep in the Pacific Ocean's Mariana Trench (<http://www.deepseachallenge.com/>). In China, Chinese Academy of Sciences (CAS) set up a specific research project in 2013 to develop a full ocean depth ARV, and the "Hadal" ARV reached the Challenger Deep in 2016, but it was also sadly lost in 2017 which showed the great risk for unmanned submersibles.

New Scientific Discovery

Many scientific discoveries have been made by the use of various submersibles, and they are summarized in NRC (2000). For example, in 1977, scientists diving to the Galápagos Rift in *Alvin* explore seafloor vents gushing shimmering, warm, mineral-rich fluids into the cold, dark depths (www.who.edu). Amazingly, the vents are surrounded by extraordinary, unexpected life forms (Fig. 2).

Salvage of H-Bomb

In 1966, the US Navy called in *Alvin* to help find a hydrogen bomb accidentally dropped into the Mediterranean Sea. *Alvin* located the bomb, and, after the bomb skidded down a slope on the first recovery attempt, it found it again at a depth of 2,900 ft (880 m). Then, an unmanned torpedo recovery vehicle, CURV-I, became entangled in the weapon's parachute while attempting to attach a line to it. A decision was made to raise CURV and the weapon together to a depth of 100 ft (30 m), where divers attached cables to them. The bomb was brought to the surface by USS *Petrel* (Fig. 3).

Economic Setup of a Scientific Research Vessel

As is well-known, the most important cost for the use of submersibles is the cost of ship time. Therefore, a setup as shown in Fig. 4 may produce the best economic performance because researchers can carry out ocean survey with the mothership no matter day or night (Cui et al. 2014). When to a new area, the ARV will carry out a wide area

exploration first to determine the focused research area and the basic marine parameters. Then several baited landers will be deployed to shoot and catch fishes and other fauna. Finally, the manned submersible or ROV will be sent down to complete the refined operations and sampling. During the day, the landers, ARV, and HOV can work collaboratively. At night, the landers and ARV can still be used for the sea floor survey.

Cross-References

- Atmospheric Diving Suit (ADS)
- Autonomous Underwater Vehicle (AUV)
- Deep Submergence Rescue Vehicle (DSRV)
- Deep Tow System
- Glider
- Human Occupied Vehicle (HOV)
- Hybrid Remotely Operated Vehicle (HROV)/Autonomous and Remotely Operated Vehicle (ARV)
- Remotely Operated Vehicle (ROV)
- Rescue Bell
- Submarine
- Underwater Lander
- Wind Tunnel Test

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Submersible Aquaculture Structures

► Modern Aquaculture Structures

Subsea Connector

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Synonyms

Pipeline end termination; Underwater connector

Definition

The subsea connector is the key connection facility in subsea production system. The subsea connector, located at the pipe end of a jumper, is mainly used to interlink subsea production facilities, such as jumpers and subsea trees, jumpers and manifolds, jumpers, and PLETs (pipeline end termination).

Typical Subsea Connectors

According to the connection mechanism, the subsea connector could be classified into two main types: clamp connectors and collet connectors.

Clamp Connectors

Clamp connectors are mainly used in horizontal applications and are particularly well suited for larger bore, lower pressure connection applications. Clamp connectors are generally popular for subsea applications due to their relatively small size, small numbers of bolts to handle, and short installation time (Bai and Bai 2012). Figure 1 illustrates typical clamp connections. The clamp is the main locking device and



Subsea Connector, Fig. 1 Clamp connector. (Adapted from FMC technologies. <http://www.fmctechnologies.com/en/SubseaSystems/Technologies/SubseaProductionSystems/TieInAndFlowlines/Connectors/Clamp.aspx>. Accessed: 2018-10-14)

typically consists of two or three segments that are closed around the mating hubs by a single rotating drive screw. A metal gasket is located between the hubs that creates a seal which should be pressure tested after the connection is made, to check if the connection is tight. One of the primary advantages of clamp systems is that they are easily locked by an ROV (remotely operated vehicle) using a torque tool, eliminating the need for larger tooling.

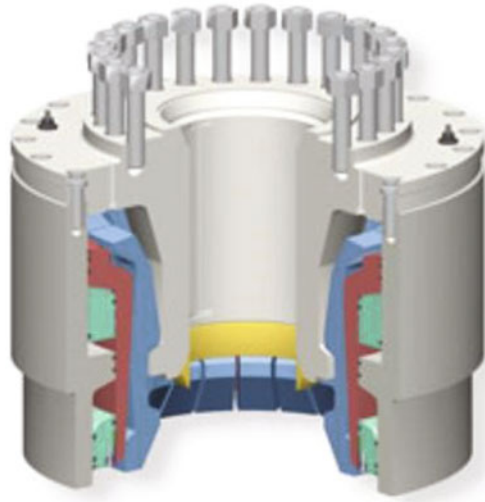
The characteristics of clamp hubs are summarized as follows:

- The clamping device preloads two hubs as the device is tightened.
- A metal gasket is compressed between the two hubs to create a seal.
- It can be made up by divers or ROV-operated tools.
- It is faster to make up than a bolted flange because it has fewer bolts.
- Rotational alignment is not critical.

Collet Connectors

Collet connectors are the most widely used for both vertical and horizontal jumper spool connections and are available in both integral hydraulic and mechanical configurations. The connection principle is similar to that for a hub and clamp except that a series of longitudinally segmented collets replaces the clamp. The collets are activated by an annular locking cam ring. The cam ring slides axially along the collet length to either open or close the connector. The driving angle on the cam ring is typically self-locking to prevent incidental un-locking of the connector. A collet connector consists of a body and hub around in which individual “fingers” or collets are arranged in a circular pattern and attached to the body, engaging the hub to form a fully structural connection as shown in Fig. 2.

The engagement between the hub and collets is either mechanically or hydraulically actuated. Funnels or guide posts provide coarse alignment for the mating hubs. After the collet connector is landed, an ROV installs the hot stab and provides hydraulic power to the stroking cylinders, pulling the hubs together at a controlled rate. The open



Subsea Connector, Fig. 2 Collet connector. (Adapted from Cameron a Schlumberger Company. <https://www.products.slb.com/pressure-control-equipment/collet-connectors>. Accessed: 2018-10-14)

collets provide initial makeup alignment to protect the metal seal. The standard design accommodates up to $\pm 2^\circ$ angular misalignment and ± 1.5 in. axial offset. Once the hubs are mated, hydraulic collet actuating cylinders close the collet fingers to complete the connection. The collets impart a preload that gives the connection the same strength characteristics as the pipe. The seal is pressure tested after makeup via the ROV control panel.

The characteristics of collet connectors are summarized as follows:

- Collet segments are driven around a connector body and hub by an actuator ring.
- A metal gasket is compressed between the body and hub to create a seal.
- It functions by means of integral hydraulics or by an ROV-operated tool.
- It can align misaligned hubs.
- Rotational alignment is not critical.

Connector Comparison

The subsea connector choice should consider factors such as water depth, bore diameter, connection direction, installation method, environment

Subsea Connector, Table 1 Comparison between clamp and collet connectors

	Clamp connectors	Collet connectors
Structure complexity	Simpler	More complex
Duration of installation	Shorter	Longer
Connection direction	Mainly horizontal	Horizontal and vertical
Water depth	Shallower	Deeper
Environment pressure	Lower	Higher
Bore diameter	Larger	Smaller
Reliability	Lower	Higher
Cost	Lower	Higher
System integration	Nonintegral	Integral and nonintegral

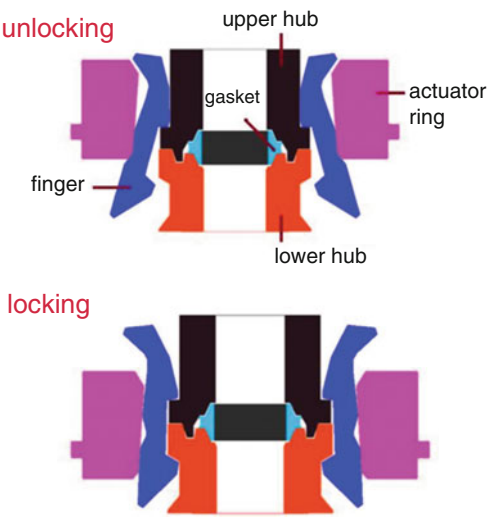
pressure, reliability, cost, and so on. Table 1 shows the relative comparison between the clamp connectors and the collet connectors in applications, where the system integration is defined as:

- Integral: The hydraulic actuation system is a permanent part of the connector assembly and remains subsea.
- Nonintegral: The hydraulic actuation system is part of an installation tool. After installation, the collet connector assembly remains subsea without any hydraulic components.

Sealing Principle

The sealing principle of a subsea connector is that contact loads, derived from extrusion between hubs and gasket, cause hub surface and gasket surface deformation, which will fill the microscopic clearance between the contact surface of hubs and gasket, thereby achieving sealing performance (Zhang et al. 2017). The gasket is then the key component which affects the reliability of connectors. It plays the role of isolating fluid inside from flowing outside. It prevents the internal oil/gas to spill and seawater to enter the pipeline.

Figure 3 shows a schematic of the target subsea connector. A collet connector includes a gasket,



Subsea Connector, Fig. 3 Collet connector. (Adapted from Wang et al. 2012)

an upper hub, and a lower hub, about 12 to 24 fingers. The gasket is located in the interface between the upper and lower hubs. The lower hub is fixed to the base on the manifold, and the upper hub is connected with jumper. When the upper and lower hubs complete docking, the actuator ring is pushed down by the force from the motive source, usually several hydraulic cylinders in the installation tool. Hence, the fingers move inward. Fingers clamp the upper hub and the lower hub. The force is amplified and transmitted from hydraulic cylinders to the gasket by conic surfaces on fingers and hubs. Then, the gasket produces plastic deformation to fill the gap between two contact surfaces and it forms the initial seal. When the connector is in the operating condition, the pressure inside the pipe makes the gasket produce expansion and deformation again but still has to meet the sealing requirements. For this reason, fingers' locking force and the relative displacement between the upper hub and lower hub are both important parameters of the fingers.

Connector Assembly

The connector assembly may consist of the following components:

- Connector receiver (hub and support structure).
- Connector.
- Connector actuator.
- Connector override tool.

Figure 4 illustrates one typical collet connector assembly from OneSubsea CVC system, which includes the connector, connector receiver (hub and support structure), and connector actuator, and the connector assemblies are connected with jumper pipe and are widely used in the connection of subsea structures.

These subsea components are summarized in the following subsections.

Connector Receiver

The connector receiver is mounted on the subsea equipment (such as manifold, PLET, and tree) and includes the connector mating hub, connector alignment system, piping, and support structure. Metal-to-metal seal surfaces for each connector receiver are inlaid with a corrosion-resistant alloy.

The connector receiver should include the following components: (1) an alignment system for guiding the connector onto the mating hub (the alignment system should provide coarse alignment during the initial landing of the jumper connectors onto the connector receivers and align the connector and mating hub during final lowering within the makeup tolerances of the connector), (2) a measurement tool interface, (3) two high-quality inclinometers (90 degrees to one another) to measure the inclination of each hub directly, and (4) pad eyes for attachment of a guideline by an ROV.

The connector receiver should be designed with the following requirements in mind:

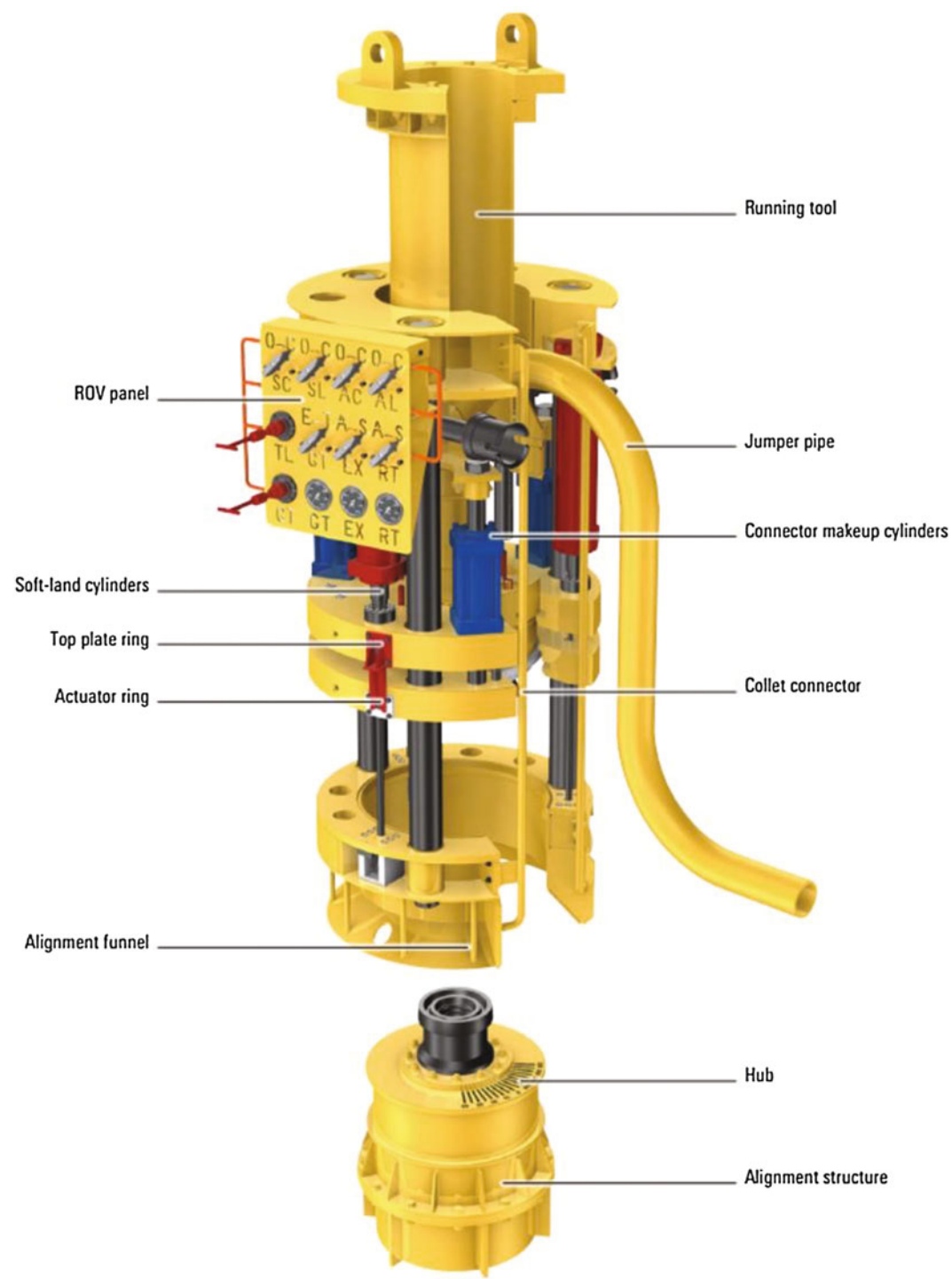
- The connector receiver should be designed to be a self-contained assembly that can be independently tested prior to integration into the subsea equipment (tree, manifold, PLET, etc.).
- The connector receiver should be field weldable to the subsea equipment (e.g., piping weld).

- The connector should be able to transfer jumper loads and connector override loads into the subsea structure (tree frame, manifold structure, PLET, etc.).
- The connector should protect critical surfaces on the mating hub during installation and retrieval.
- The connector should be designed to accept hub end caps.

Connector

The connector should be used at the end of the jumper piping to lock and seal the mating hub on the connector receiver. The connector should be equipped with the following components: (1) a mechanism (e.g., collet fingers, or dogs) designed to resist lateral and longitudinal forces that may be encountered in the process of aligning and final lowering, prior to makeup of the connection (the connector is designed to accommodate the design loads); (2) metal-to-metal seal surfaces inlaid with corrosion-resistant alloy (the seal surfaces shall be relatively insensitive to contaminants or minor surface defects and maintain seal integrity in the presence of the maximum bending moments and/or torsional moments); (3) a metal seal with an elastomeric backup capable of multiple makeups; (4) mechanical position indicators to indicate lock/unlock operations that are clearly readable by an ROV; and (5) a mechanical release override or a hydraulic secondary release system. The connector should be designed to satisfy the following requirements:

- The connector should not permanently deform the mating hub during connection.
- The connector should protect seals and seal surfaces during deployment and retrieval.
- The connector should retain the metal-to-metal seal during jumper installation and retrieval. The metal seal should be capable of being replaced by an ROV without bringing the jumper to the surface.
- The connector should be welded to the jumper pipe. The mechanical unlatch load should not exceed the allowable stresses in the connector receiver structure and piping.



Subsea Connector, Fig. 4 The OneSubsea CVC connector. (Adapted from OneSubsea a Schlumberger Company. [resources/2016/04/02/04/08/oss-cvc-connector-brochure.ashx](https://www.onesubsea.slb.com/-/media/cam/resources/2016/04/02/04/08/oss-cvc-connector-brochure.ashx). Accessed: 2018-10-14)

- The connector should provide for an external pressure test of the metal-to-metal seal after makeup using an ROV hot stab. The external pressure test is at least 1.25 times the ambient hydrostatic head. A volumetric compensator is included in the seal test circuit. This circuit is constructed with welded fittings wherever possible.

Connector Actuator

The connector actuator may be integral with the connector or independently retrievable from the connector. The connector actuator should include the following components: (1) ROV handles, which are positioned to observe critical surfaces while guiding the stabbing operation; (2) an ROV panel with pressure gauges to monitor tool functions, hot stab receptacles, etc., for operation of the soft landing system, connector makeup, and seal test (all control and seal test tubing is to be welded whenever possible; separate ROV hot stabs should be provided for each function); (3) indicators that allow the ROV to monitor tool hydraulic functions; (4) a mechanical release (override); and (5) an optional hydraulic secondary release. Note, however, that even if the hydraulic release is used, the mechanical release is still required.

The connector actuator should be designed considering the following requirements:

- The connector actuator should be designed to preclude accidental unlocking from impact loads, vibration, thermal loads, and any other loads affecting the locking mechanism. A secondary lock should be provided where self-locking mechanisms are used.
- The independently retrievable actuator should be positively locked onto the connector and protect the connector during running, retrieval, and topside handling operations.

Soft Landing System

A soft landing system for retrieval and installation should be provided to minimize the impact loads. This system is an integral part of the connector or connector actuator. It is designed to absorb impact loads while landing the jumper on the connector

receivers and to maintain the separation between the sealing surfaces when the jumper lands on the connector receiver structure. The soft landing system supports the weight of the jumper to isolate the jumper from vessel motion as the connectors are lowered onto the mating hubs. The system pulls the connector and mating hub faces together prior to locking the connector on to the hub by lowering or raising the connector from the mating hub in a controlled manner. It has mechanical position indicators to indicate a lowering operation that are clearly readable by an ROV and allows each end of the jumper to be lowered or raised independently of the other.

Cross-References

- [Pile Capacity](#)

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Subsea Equipment Installation Techniques

- [Subsea Equipment Installation Technology](#)

Subsea Equipment Installation Technology

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Synonyms

[Subsea equipment installation techniques](#); [Underwater equipment fix techniques](#)

Definition

Subsea equipment installation technology is used to fix offshore structures in a certain position according to certain procedures and specifications and also to fix subsea equipment in a certain place according to certain methods and specifications.

Introduction

In recent years, subsea oil and gas development has become the focus of global oil and gas development, and subsea oil and gas production has increased year by year. In view of the depletion of onshore oil resources, subsea will undoubtedly be one of the priorities of the global oil strategy replacement in the future. Gulf of Mexico, West Africa, and Brazil, i.e., the “Golden Triangle,” are the three major subsea core oil and gas development areas. As indispensable equipment for subsea oil and gas development, subsea Christmas tree and subsea manifold have been paid more and more attention by the offshore oil and gas industry.

As the offshore oil industry gradually extends from shallow water to deep water, and ultra-deep water, the installation technology of subsea equipment has attracted more and more attention from China and foreign companies and researchers. Safe and efficient installation technology for

subsea oil and gas equipment is a key technical requirement that must be solved in the implementation of subsea oil and gas equipment from localization to the actual application in the South China Sea. Due to the special geographical location and topographical conditions of the South China Sea, offshore structures are vulnerable to extreme weather conditions. The impact of the marine environment is complex. When subsea oil and gas equipment passes through the splash zone, it is prone to wave breaking, large movements, and overturning. The vibration frequency of the drill pipe or wire rope could resonate with wave actions and lead to fatigue damage. Therefore installation of subsea equipment is a great challenge.

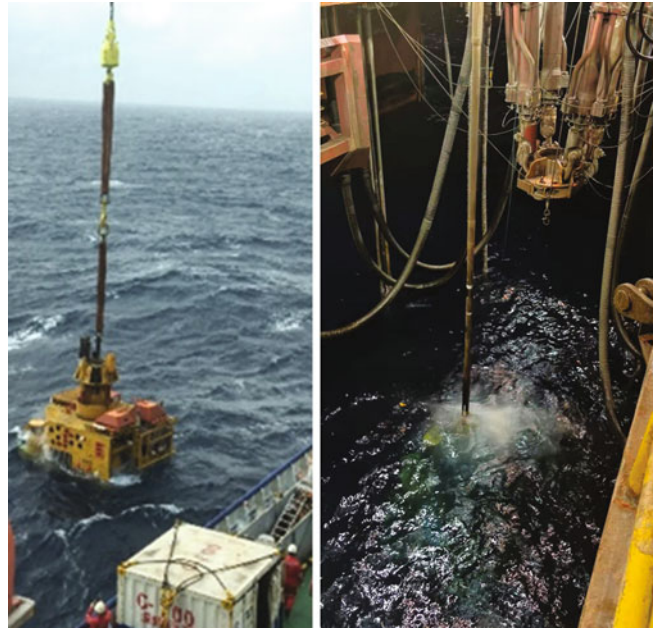
The Main Installation Methods

Subsea equipment can be installed mainly by drill pipes and wire ropes in traditional methods. The newly developed installation methods mainly include pendulous method, sheave lowering method, pencil buoy method, etc. These methods are mainly used for installation of larger subsea equipment such as subsea templates or manifolds. Mainly in Brazil, West Africa, North American Gulf of Mexico, and Norway, these new developed installation methods are relatively well developed and have been successfully used in subsea oil and gas field development projects.

Lifting Method

The lifting method is the main installation method of early subsea manifolds in deep water. There are two main forms of lifting method (Cermelli et al. 2003). One is to transport the manifold to the installation position by a supply vessel, and second is to use a wire rope to lower it to the installation location through a crane on the supply vessel or a crane barge. The other is to lower the subsea manifold to the installation location by a drill pipe. Generally speaking, the installation application depth of this lifting method is basically within 1000 m. For the installation in deeper water, the lifting method still has certain limitation due to its own load capacity and axial resonance

Subsea Equipment Installation Technology,
Fig. 1 Installation method of subsea tree using steel cable and drilling rig, respectively



problems. Figure 1 shows the installation of subsea tree using wire ropes and drilling risers.

Installation with a Drill Pipe

The installation method by means of drill pipe is a process in which the drill pipes are connected and lowered to the subsea location with the subsea equipment. This method mainly uses a supply vessel to transport subsea equipment to a movable offshore drilling device, connects the subsea equipment to the drill pipe, and then lowers it. This process also requires monitoring systems and subsea positioning systems in the installation vessels. Generally, when the water depth is less than 500 m, a guide rope is needed. The subsea equipment is lowered with the guide through four (or two diagonal) tensioned guide ropes. The end of the guide rope is connected to the subsea guide base, and the top is connected to the installation boat. But this method may be limited by the water depth and the length of the guide ropes. When the water depth is greater than 500 m, guideless installation is used. Generally, the installation of the dynamic positioning system and the cone-shaped guide funnel at the bottom of the tree are used. However, the disadvantage of this method is that it takes more time

to connect the drill pipes and has the lower efficiency in deep water.

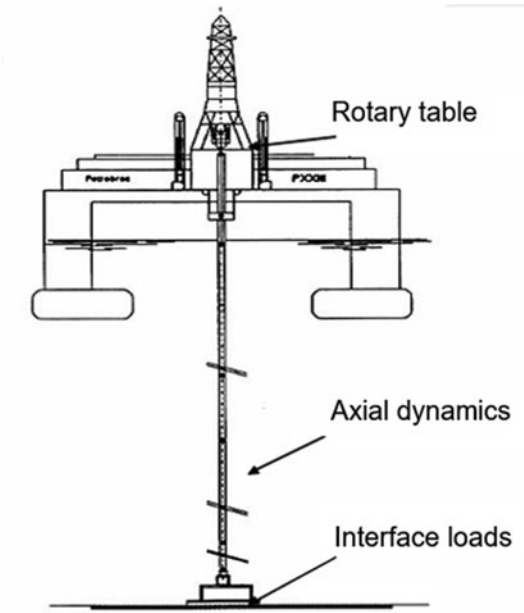
Under this installation method, the subsea tree is firstly transported by a barge to the designated drilling platform and then uses a section of drill pipe to lower the subsea equipment to the sea floor through the moon pool. Drilling vessels capable of deepwater operations usually have a dynamic positioning function, and their main hooks can generally provide heave compensation and have a large lifting capacity, so they have the ability to install a subsea tree. However, compared with multi-purpose engineering vessels, the drilling platform or vessels have fewer resources, higher rental cost, and longer drill pipe connection time.

Installation through drill pipe is limited by the load capacity and the length of the drilling pipes, which limits the applications of this method. When the manifold/base plate is lowered, the mobile drilling device cannot perform drilling and workover operations. It leads to slower installation. Due to the high daily rent of mobile drilling units, the development cost of the entire oil field tends to be very high.

Generally, the installation process of the subsea equipment through drill pipe includes three main steps. The steps are shown in the Figs. 2, 3,

4, 5, 6, and 7. The first step is to transfer the subsea equipment from the dock to the barge and then to the installation position. In the second step, the

drill pipe is connected to the subsea equipment and lifts the equipment off the barge. The third step is to use the semisubmersible platform to locate the subsea equipment at the intended installation. Position vertically upward, and then lower the drill pipe with the subsea equipment to the sea floor.



Subsea Equipment Installation Technology, Fig. 2 Deployment plan of drill pipe conduit installation method



Subsea Equipment Installation Technology, Fig. 4 Use of a semisubmersible platform to move a manifold/basket from a dock to a barge

Subsea Equipment Installation Technology, Fig. 3 Semisubmersible platform with multifunctional tower



Subsea Equipment Installation Technology,

Fig. 5 Positioning a semisubmersible platform with a barge



Subsea Equipment Installation Technology,
Fig. 6 Connection between drill pipe and a manifold/
base plate through a moon groove

Installation by Winch/Crane

The second traditional vertical lowering installation method is by means of a winch/crane installation. When the working water depth of the crane is satisfied, the subsea equipment can be lowered directly by a crane. When the water depth exceeds the working water depth of the crane, a winch can be used to lower the equipment to the sea floor.

The traditional vertical installation through crane/winch includes the following steps. Firstly, the subsea equipment is transported to the sea by a

tugboat or a lifting barge; secondly, connect the subsea equipment to the offshore lifting equipment, and lift it from the deck of the tugboat, and adjust it above the installation position; after completing the above connection work, the equipment is lowered, and the lowering path is monitored by a detection device. After the deployment of the subsea signal positioning system, the equipment is installed. Manifolds with a large installation weight need heavy lifting capacity and unique positioning performance, which are affected by the environment of the installation position.

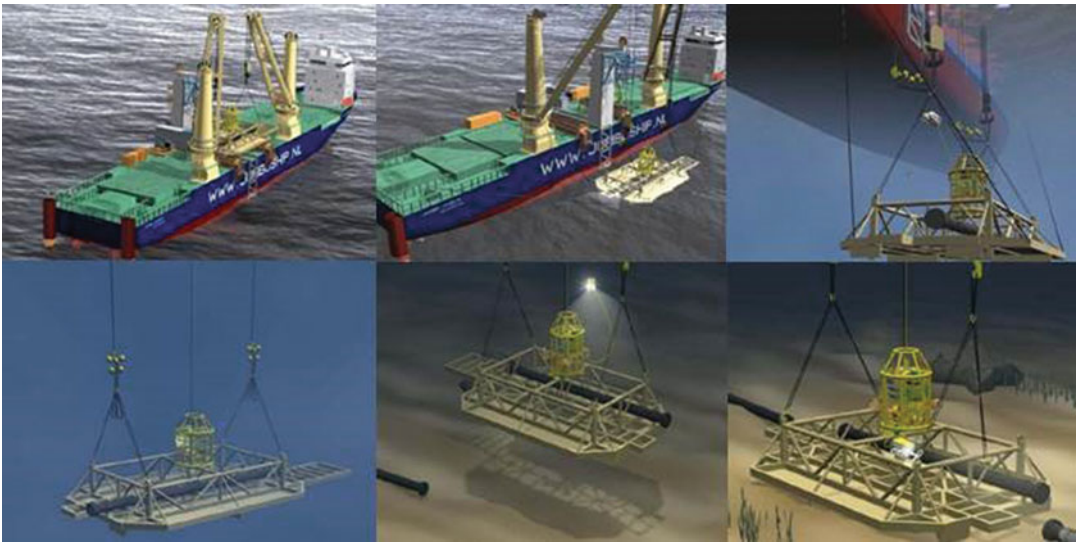
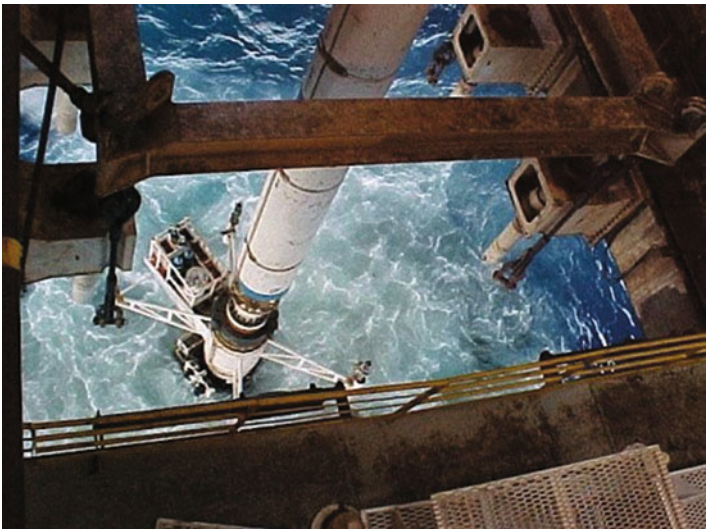
The specific installation process is shown in Fig. 8.

Sheave Method

The sheave method is an effective method to solve the manifold installation problem in deep water (more than 1000 m), and it is an improvement lifting method. This method mainly uses two anchor handling and towing supply (AHTS) vessels and a semisubmersible (SS) drilling platform for the installation of subsea manifolds. The working principle is shown in Fig. 9 (Lima et al. 2008).

In practice, a pulley is installed above the subsea manifold, and two wire ropes are installed on the pulley. One of the wire ropes is connected to the SS platform and AHTS 1 through the sheave, and the other is directly connected to AHTS 2. Besides, the target manifold is connected to

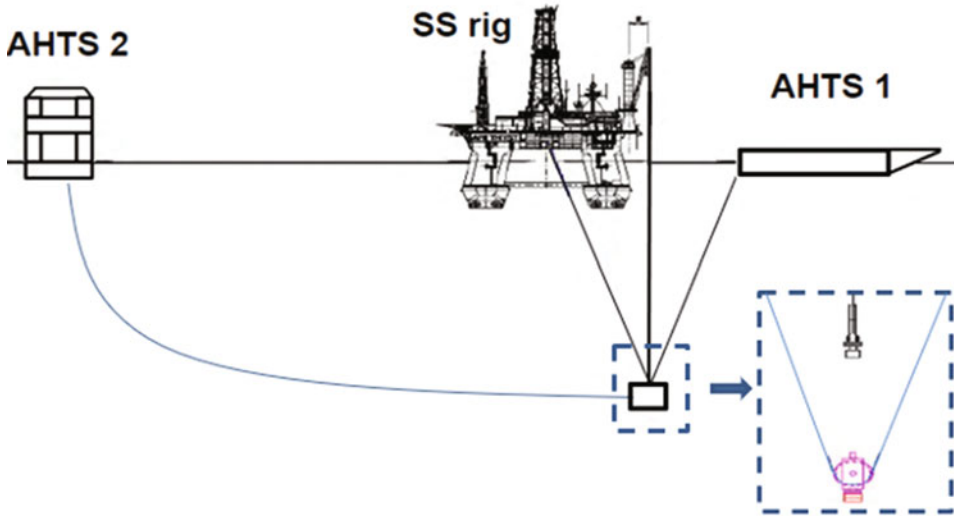
Subsea Equipment Installation Technology, Fig. 7 Lowering to the bottom of the sea through the subsea equipment



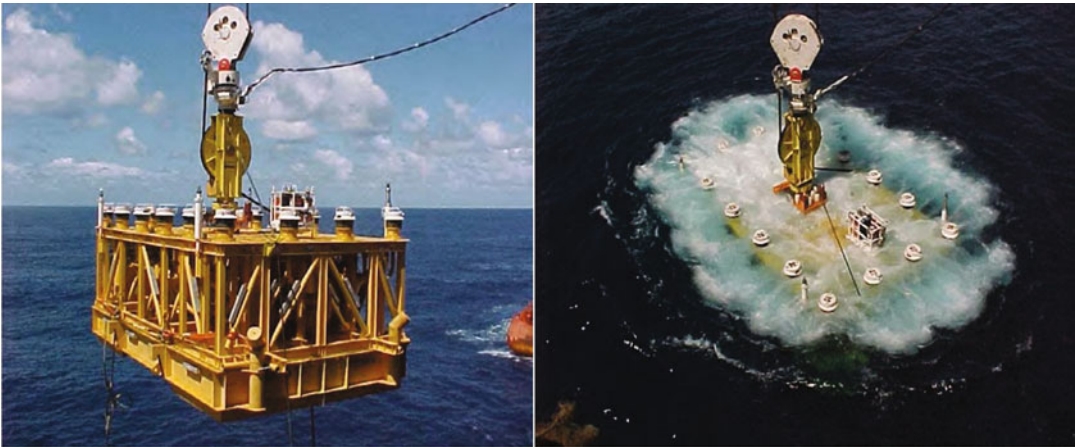
Subsea Equipment Installation Technology, Fig. 8 Vertical installation of subsea equipment

the crane rope on the SS platform. After the connection is completed, the subsea manifold is lowered by the crane to a certain depth, and then detach the crane rope from the subsea manifold, so that the SS platform and AHTS 1 share the gravity of the subsea manifold by the wire rope on the sheave, and begin to lower the manifold slowly. At the same time, the AHTS 2 is used to locate the direction of the subsea manifold to avoid winding or position displacement

due to the load of wind, waves, and current when the manifold is lowered and installed. Because the SS platform has an automatic heave motion compensation function, it can reduce or even avoid the impact of wave motion on the subsea manifold sinking during the process of lowering the manifold. It has high stability and can withstand the harsh marine environment, but it requires two auxiliary vessels. Fig. 10 shows the use of this method to lower the submarine manifold. This



Subsea Equipment Installation Technology, Fig. 9 The installation method of subsea manifold by the sheave method



Subsea Equipment Installation Technology, Fig. 10 The installation of subsea manifold by the sheave method in subsea

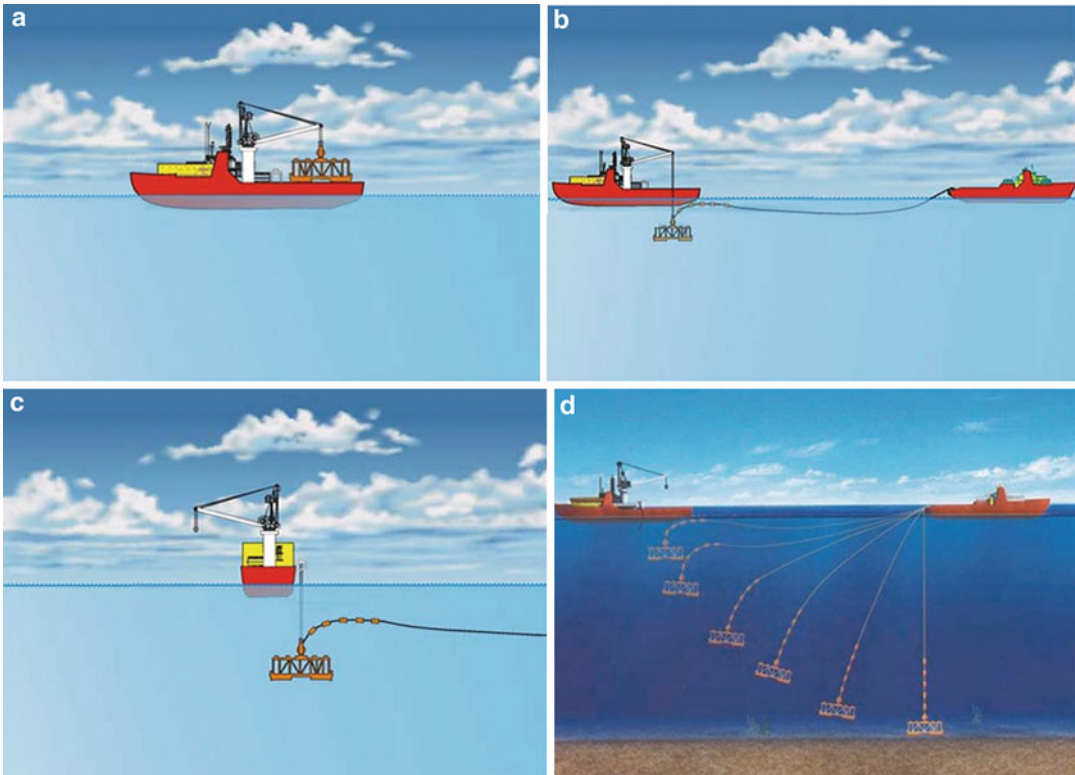
method has been successfully used for the installation of three manifolds in the Roncador oil field in Brazil.

Pendulous Method

The pendulous method is a new installation method boldly proposed by Petrobras in 2003. It is mainly used to solve the installation of large subsea equipment in deep water and ultra-deep water, up to 3000 m water depth. In the same

year, the Brazilian Industrial Property Association granted the Brazilian petroleum company the patent right for the pendulous installation, which is one of the most advanced installation methods for large subsea equipment in ultra-deep water in the world.

The pendulous method was first applied in Brazil's Roncador oil field, and manifolds weighing 200 t and 280 t were installed at water depths of 1900 m and 1845 m, respectively.



Subsea Equipment Installation Technology,
Fig. 11 The installation principle of the pendulous method for manifold in subsea. (a) Transport vessel and submarine manifold/basket. (b) Submarine manifold/

basket is connected into the sea by the installation vessel. (c) Subsea manifold/base plate is lowered from the side of the transport vessel. (d) Manifold/base plate is lowered

Figure 11 (Machado et al. 2005; Ribeiro et al. 2006) shows the installation principle of the subsea pipe manifold pendulous method. During the installation, two work boats (the transport vessel and the installation vessel) are needed. Firstly, the subsea manifold is transported to a transport vessel equipped with a crane. The crane is used to submerge the manifold into a position, which is close to the sea surface, and then the installation cable is connected to the installation vessel (the actual installation position), as shown in Fig. 11a. At this time, the installation cable is required to be slightly shorter than the actual installation water depth, and the distance between the two vessels is about 90% of the installation cable. After the connection is completed, the carrier releases the manifold to make it slowly sag. At the same time, the installation boat controls the movement of the installation cable until it reaches the vertical

position and maintains balance, as shown in Fig. 11b. Then determine the gap with the mud line by testing the elongation of the installation cable. The subsea manifold needs to maintain a proper distance from the mud line in order to further adjust its position by installing the cable. Once the position is determined, it will be implemented.

Pencil Buoy Method

The pencil buoy method (PBM) (Risoey et al. 2007) is a subsea transportation and installation method developed by Aker. The PBM requires a lifting barge to lift the structure from a transport barge with a sufficient depth of water along the coast.

The installation process is as follows. The manifold is transported to the launching position by a barge and lifted from the barge into the seawater. The weight of the manifold is

transferred from the floating crane to the installation vessel. The rigging on the manifold is connected to the winch and pencil-shaped pontoon of the installation boat. The buoy slowly enters the water and the installation boat moves forward slowly. After towing to the installation position, the winch recovers the cables, and the weight of the manifold is transferred from the pontoon to the winch. The pontoon is recovered to the deck of the installation vessel and disconnected. The installation vessel is moved forward slowly to avoid the collision of the manifold with the installation vessel. Once stabilized, lower the manifold slowly to the ocean floor.

A pencil-shaped buoy is a steel structure with internal annular ribs. The buoy can be subdivided into many impervious tanks to meet the requirements for use under certain damage. Generally, buoys need to maintain a wet towing speed of about 3.0–3.5 knots. When the structure arrives at the installation position, the weight of the structure is retransferred from the pencil-shaped pontoon to the tug winch wire rope, and the tow cable is still connected at this stage. After the weight of the structure is completely transferred, the connection between the buoy and the structure is broken. During the load transfer process, the utility tug should move slowly forward to keep no contact between the tug stern and the pontoon. When the structure is lowered to the sea floor, the passive heave compensation system is used. In this method, the structure is transported and lowered through the splash

zone in the coastal sheltered area, not offshore. Subsea wet towing is typically not restricted by summer storms. The lowering of the structure is a standard offshore operation similar to other subsea installation methods.

Figure 12 shows the installation instructions of the pencil buoy method and the position of the pencil buoy before lowering. This installation method requires a three-purpose tugboat with standard equipment, a tailor-made pencil buoy, and a passive heave compensation system. Those are indispensable. The first generation of pencil-shaped buoys has a carrying capacity of 150 t, while the second- and recent third-generation buoys have a carrying capacity of 250 t and 350 t, respectively. Due to the action of waves and ocean currents, both the diving structure and the surface pencil-shaped pontoon suffer from complex hydrodynamic problems.

Moon Pond Wet Drag Method

Also as a method of wet towing, the moon pond wet towing method is developed by the Subsea 7 and has had several successful application examples in the North Sea, where one of the most severe sea conditions happens.

The most representative of these was the installation of a base manifold of 260 t in the Tyrihans oil field of 285 m depth in Norway in the spring of 2007 (Stock et al. 2002); see Fig. 13. Using this method, the subsea structure is firstly transported by a barge, floated and placed on the bottom of the



Subsea Equipment Installation Technology, Fig. 12 Pencil buoy method, by Aker



Subsea Equipment Installation Technology,
Fig. 13 Moon pond wet drag method

sea not far from the construction position through a multifunctional vessel, and then the subsea structure is lifted by the installation vessel the water through the moon pond, wet to the installation position, and after reaching the destination directly down to the ocean floor.

It is required that the installation of the vessel must be equipped with a winch system of considerable capacity and can provide heave compensation function. In this project example, a manifold weighs 260 t, and the winch installed with the Botnica multi-purpose vessel has only a 300 t winch system. The main reason for this efficient application to be used in this project is that this method avoids the occurrence of passing through the splash zone. The large dynamic load and the shallower water depth of the rigging itself are relatively small, so it is very suitable for the implementation of the sea area where the sea conditions are more severe. It can expand the climate windows for installation and increase operating efficiency.

Adaptability Comparison of Five Installation Methods

There are five main installation methods for subsea equipment, namely, the lifting method, the sheave method, the pendulous method, the pencil buoy method, and the moon pond wet drag method. The comparison of different installation methods is shown in Table 1. Among those methods, the lifting method has the most mature technology, and it is most widely used within a depth of 1000 m. However, due to its own load capacity, axial resonance, and high rents, there are still some limitations in the installation of subsea and ultra-deepwater manifolds. Although fiber ropes are considered instead of wire ropes, further research is needed. The sheave method can be applied to the installation of subsea manifolds with harsh sea conditions and greater than 1000 m, but it is necessary to consider the lifting capacity of vessel resources and whether related auxiliary equipment meets the conditions. The pendulous method does not require special vessels, and it can overcome the disadvantages of axial resonance and economically and efficiently install large equipment. It is the current advanced technology for ultra-deepwater and heavy equipment subsea installation, but it has higher requirements for the installation of climate windows in the sea. The pencil buoy method and the moon pond wet drag method are not applied widely in China.

Conclusions

This chapter mainly describes the installation technology and method of subsea equipment. As for the installation method of drill pipe, it has the characteristics of convenient installation, but the installation efficiency is low, and the size of the object to be installed will be limited due to the impact of the size of the moon pool. Moreover, it is not commonly used in deep water due to the limited bearing capacity of the drill string. For lifting method, the installation operation is also simple, but deep water is prone to resonance. For the pendulous method, the mass and volume of the object to be installed are not limited, and the

Subsea Equipment Installation Technology, Table 1 Performance comparison of lifting method, sheave method, pendulous method, pencil buoy method, and moon pond wet drag method

	Lifting method	Sheave method	Pendulous method	Pencil buoy method	Moon pond wet drag method
Required resources	Supply vessel or crane barge with crane or mobile drilling unit	Anchor towing vessel (2 vessels), SS drilling platform with crane	2 ordinary vessels, one of which has crane capacity	Multi-purpose boat	Multi-purpose boat
Suitable water depth (m)	Within 1000 m	More than 1000 m	Up to 1000 m	Within 1000 m	Within 1000 m
Improved capacity	Requires higher lifting capacity	Required to be medium	The lifting capacity of the installation vessel is medium	Required to be medium	Wet mop method is adopted, required to be medium
Environmental requirements	Drilling vessels are susceptible to wind and waves and have poor stability	Reduces the requirements for vessel resources and the impact of wind and waves and better requirements for installing climate windows	Low requirements for climate windows	The impact of wind and waves is small, not subject to the limitation of summer storms; suitable for sea areas with harsh sea conditions	Adopt wet towing method; the impact of wind and waves is less
Economical	Dynamically positioned drilling vessels are expensive and economically inferior	Reduced requirements for the capacity of hoisting ropes, cranes, or winches, reducing the cost of expensive vessel installation resources	Relatively scarce resources for installing vessels, PIM is more economical	Multifunctional vessel, which integrates transportation and installation, has better economy	Multifunctional vessel, which integrates transportation and installation, has better economy

requirements for working ships are also low. It is suitable for ultra-deepwater installation, but the installation is relatively complex, the carrying capacity of the wire rope is limited, and there is a risk of collision of working ships. For the sheave method, it can install equipment in deep water, with relatively loose size and weight, but there is also the risk of ship collision in operation. For pencil buoy method, it is easy to operate, and the installation equipment is not limited, and it can complete the lifting and transfer of subsea facilities in the offshore area. However, the cost of wet towing is high, and the ability of winch in deep-water installation depends on laziness and is easily affected by the environment. For the moon pond wet drag method, it requires less to install the climate window, and the smaller installation ship can install larger subsea equipment, but it

also has a high cost of wet towing, which requires very high winch capacity.

For the development of offshore oil and gas fields, it is particularly important to choose the appropriate and economical installation method when installing subsea equipment. The detailed description of installation method in this chapter is expected to provide some reference for readers.

Cross-References

- [Christmas Tree](#)
- [Dynamic Analysis Method](#)
- [Installation of Offshore Pipelines](#)
- [Lifting System](#)
- [Marine Operations](#)
- [Mooring Lines](#)

- [Net Structures: Hydrodynamics](#)
- [Semi-submersible Platform](#)
- [Ship Operational Environment](#)
- [Static Analysis Method](#)
- [Winch](#)

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Subsea Hazards

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Definition

Subsea hazards are referred to natural processes that happen above seabed and have hazardous effects on structures in subsea engineering.

Subsea engineering structures are normally facing various subsea natural hazards, including subsea landslides, storms, migration of sand

waves, volcano eruptions, subsea earthquakes, etc. Within the above mentioned hazards, subsea landslide is arguably the hardest to predict and has the most catastrophic impact on subsea structures. Therefore, this entry is focused on introducing the current understanding and existing challenges about subsea landslide and its impact on subsea structures.

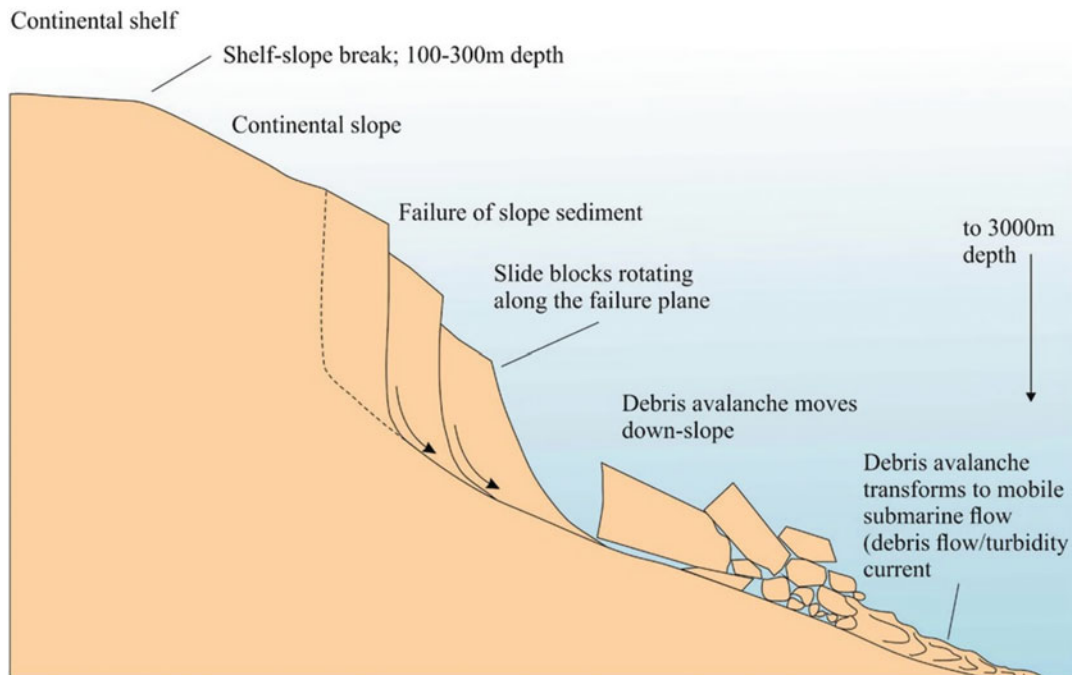
Formation Mechanisms of Submarine Landslide

Sediment formed on land normally is carried by river to the ocean and gets deposited on seabed due to reduced flow velocity. Continuous sediment deposition from river supply will form temporary storage on upper continental slope. However, these deposits only serve as temporary sediment storage, as instability in the slope deposits leads to periodic slope failure, landsliding, and onward downslope transport. Besides the above-mentioned slope failure, earthquake, volcanic activity, waves, tidal and gas hydrate, etc., could also trigger submarine landslide (Masson et al. 2006).

Process of Submarine Landslide

Figure 1 is a commonly accepted sketch for the processes of submarine landslide. In this diagram, large volumes of cohesive sediment fails from continental shelf break due to a triggering reason, then the sediment moves downslope and gradually breaks into smaller pieces and forms debris to avalanche in the form of falling, bouncing, and rolling; with further traveling downslope and water entrainment, the debris flow evolves into high density turbidity current. Eventually the sediment in the turbidity current resettles on seabed at locations with much deeper water depth. In the overall processes, sediment could travel hundreds kilometers distance.

There were different opinions about how far the landslide material travel before transforming into turbidity flow. Some research argued that landslide travels hundreds of kilometers without transformation into turbidity current (Bryn et al.



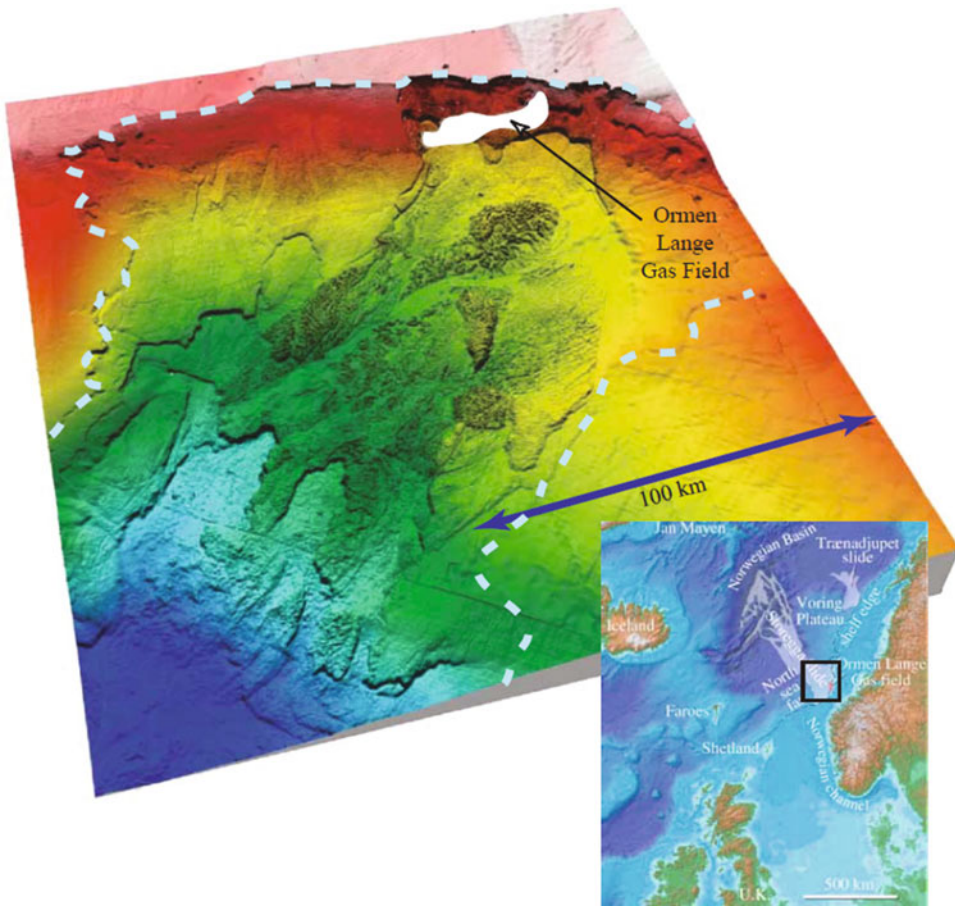
Subsea Hazards, Fig. 1 A sketch of subsea landslide. (The picture is from <https://theconversation.com/scars-left-by-australias-undersea-landslides-reveal-future-tsunami-potential-85982>)

2005), but some others believe this could happen near the source of the landslide (Talling et al. 2002). Through some laboratory experiments, Elverhoi et al. (2010) demonstrated that the clay-to-sand ratio is a critical parameter that governs the behavior of submarine landslide. The tests by Elverhoi et al. (2010) were conducted in a tank with 10 m in length, 3 m in depth, and 0.6 m in width and a 6° bottom slope. It was found that the slide sediment with 25% clay (by weight) tends to remain compact and attain long run-out distance and the frontal of the slide remains in laminar state and experiences hydroplaning. No obvious deposition was formed during the sliding. When the clay content is reduced to 15%, the frontal of the slide becomes turbulent and a layer of deposition is formed behind the front. When the clay content is further reduced to 5%, water intrusion happens very quickly to the front of the slide and most of the sediment settles on the seabed within a relatively short distance. Therefore the clay content is an important parameter that controls the submarine landslide process. It should be noted that all the laboratory tests were conducted with small-

scale models and there is not a clear understanding about the scaling effect yet. Therefore more research work is still need to understand the processes of submarine land slide.

Where Does Submarine Landslide Happen?

Submarine landslides are widely spread in oceans. For example, it has been identified in the region of Hawaiian Ridge (Moore et al. 1989), around Australia (Mollison et al. 2020; Hengesh et al. 2012), and in Norway Basin (Haflidason et al. 2005) and many other locations. During a landslide, significant volumes of material can be moved. Survey data shows that the moved volume can be in the scale of $10^6 \sim 10^{14} \text{ m}^3$ according to Masson et al. (2006). Some details about the Storegga Slide are given as an example. *The Storegga Slide has affected an area of approximately 95,000 km² and a sediment volume in the range of approximately 2400 ~ 3200 km³* (Haflidason et al. 2005). The slide was dated back to 8200 years ago. The Ormen



Subsea Hazards, Fig. 2 Seabed profile of Storegga slide. The dash line labels the boundary line of the slide and the Ormen Lange Gas Field is located within the slide (Masson et al. 2006)

Lange field has been identified and developed right in the scar of the Storegga slide as shown in Fig. 2 (Masson et al. 2006). The estimated recoverable gas reserve in this field is about 300 billion cubic meters and it is one of the main energy suppliers for England. The total value of the subsea structure built for the Ormen Lange gas field is about US\$12 billion. The slide risk has received particular focus for developing the field due to the historical slide (Solheim et al. 2005).

The Impact of Submarine Landslide on Subsea Pipelines

If a subsea structure, such as a pipeline is (unfortunately) located in the route of a subsea

landslide, the structure will be strongly impacted by the landslide when it passes over or around the structure. It depends on the relative location of the subsea structures to the source location of the landslide. For structures near the source location of a landslide, the sliding material is mainly in solid form. With the sliding material moving away from the source point, the sliding material will mix with water and form non-Newtonian fluid, which does not follow Newton's law about fluid viscosity. Further away from the source point, the solid material gradually settles on the seabed and only a low concentration turbidity current remains to travel further into deeper ocean. As described above, the interaction between landslide and subsea structures can be treated in three different ways: (1) structure–soil

interaction, (2) structure–Non-Newtonian fluid interaction, and (3) structure–Newtonian fluid interaction. Normally these three interactions are investigated separately using different theories.

Due to the significant volume of seabed material moved by landslide, hardly any pipelines will be able survive sliding material in solid form. However with proper research work, it is possible to design pipelines to withstand slurry flow induced by a landslide. Most of the work about pipeline–slurry interaction was carried out under the framework of fluid–structure interaction. Here it is important to introduce the concept of Newtonian fluid and non-Newtonian fluid. A Newtonian fluid is *a fluid in which the viscous stresses arising from its flow, at every point, are linearly correlated to the local strain rate—the rate of change of its deformation over time* (Panton 2006). For example, sea water and freshwater are Newtonian fluids and the viscosity are only affected by temperature. In contrast, a non-Newtonian fluid does not follow Newton’s law of viscosity. In non-Newtonian fluids, viscosity can change with stress level to be either more liquid or more solid. Landslide-induced slurry is a typical non-Newtonian fluid.

The impact force F from submarine slurry flow on pipelines are normally calculated in the form of

$$F = \frac{1}{2} \rho D U^2 C_D$$

where ρ is the density of the slurry (or turbidity current) generated from a landslide, U is the current velocity, and C_D is the drag coefficient. Slurry flow with relatively high sediment concentration is a kind of non-Newtonian fluid. C_D value under slurry flow was obtained by Zakeri (2009) and Liu et al. (2015) through numerical simulation and laboratory experiments. It was found that C_D is correlated to non-Newtonian Reynolds number ($Re_{\text{non-Newtonian}}$), which is defined as

$$Re_{\text{non-Newtonian}} = \frac{\rho U^2}{\tau}$$

where τ is the fluid shear stress and it can be calculated according to the power law model or

the Herschel-Bulkley model. Through physical model tests, Zakeri (2009) gave the normal drag coefficient (C_{D-90}) on a free-span of a subsea pipeline as below:

$$C_{D-90} = 1.4 + \frac{17.5}{Re_{\text{non-Newtonian}}^{1.25}}$$

When the pipeline is sitting on a seabed (without a gap), a reduction to drag coefficient happens as below.

$$C_{D-90} = 1.25 + \frac{11.0}{Re_{\text{non-Newtonian}}^{1.15}}$$

It should be noted that the model tests reported by Zakeri (2009) were conducted with a fixed water content (35%) and varied clay and sand contents to achieve $Re_{\text{non-Newtonian}}$ in the range of 1 to 200. With further increase of $Re_{\text{non-Newtonian}}$, C_{D-90} reaches a constant value of 1.4 asymptotically. Through numerical simulations, Zakeri (2009) also presented the drag coefficient for slurry flow in parallel to a free span in the form of

$$C_{D-0} = 0.08 + \frac{9.2}{Re_{\text{non-Newtonian}}^{1.10}}$$

The above-mentioned approach was under fluid mechanics framework. A geotechnical approach was proposed by Randolph and White (2012), in which the impact force is expressed as

$$F = \frac{1}{2} \rho D U^2 C_D + N_p s_u D$$

where $N_p s_u D$ is the bearing force term, N_p is the bearing factor, and s_u is the undrained soil strength. The prediction using this geotechnical approach was consistent with the prediction under fluid mechanics framework.

With the sediment in the slurry settling gradually, the slurry flow will experience a transition from non-Newtonian fluid to Newtonian fluid. The transition of slurry flow was investigated by Coussot (1995) using clay-water suspensions. Coussot defined two parameters, $T = \tau/\tau_c$ and $\Gamma = \mu\dot{\gamma}/\tau_c$, where τ_c is the shear strength of the

particle interaction network at rest, μ is the viscosity of the corresponding force-free particle suspension, and $\dot{\gamma}$ is the shear rate magnitude. For non-Newtonian fluid, T and Γ follow a relationship of $T = 1 + K\Gamma^n$, where K and n are material parameters (see details in Coussot 1995). For Newtonian fluid, T and Γ are equal to each other. Coussot (1995) found that clay slurry with $\Gamma > 50$ is in the Newtonian fluid regime and slurry with $\Gamma < 0.3$ is in the non-Newtonian fluid regime. In the range of $0.3 < \Gamma < 50$, a gradual transition from non-Newtonian to Newtonian flow happens. Therefore, for slurry flow with $\Gamma > 50$, the drag coefficient measured in Newtonian fluid should be used. A comprehensive summary about the C_D on a pipeline in Newtonian fluid flow has been given by Sumer and Fredsøe (2006). It was found that the Reynolds number based on pipe diameter ($Re_D = UD/\nu$) has a strong effect on the value of $C_D - 90$ in Newtonian fluid due to the boundary layer on the pipe surface transition from laminar to turbulent state. However, limited work has been done for this effect under non-Newtonian fluid condition.

Remaining Challenges

Although a significant amount of work has been carried out in the past few decades on submarine landslides and its impact on subsea engineering, it still remains a great challenge in many aspects. The challenges related to offshore engineering are summarized as follows.

1. A good understanding about the whole landslide process is still to be developed. This understanding is extremely important for design of subsea structures with impact from subsea landslide. It will also help to predict landslides.
2. The interaction between submarine landslide and pipelines has been studied reasonably well, but its impact to other subsea structures (such as piles, foundations, subsea well, etc.) is still to be examined.
3. The fundamental understanding about the interaction between non-Newtonian slurry

flow and structures is still to be explored. Rich fluid dynamic phenomena have been identified for Newtonian fluid flow around subsea pipelines, such as flow separation, vortex shedding, drag crisis, etc. It is expected that the non-Newtonian flow could have some fundamentally different flow behavior to be investigated.

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Subsea Pipeline

► Iceberg Scouring

Subsea Production System

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Synonyms

Subsea system; Underwater production system

Definition

The industries of oil and gas areas are facing growing competition, and the onshore resources in some areas almost reached an exhausted state. With the increasing energy demands around the globe, the oil and gas development activities in deep water become more and more urgent. As the operation environment of this equipment corresponds to the deep sea or ultrahigh deep sea, traditional equipment that has been previously used in offshore and onshore environments is not appropriate for this field development. Therefore, to collect oil and gas in the deep sea, equipment that is specialized for the deep sea needs to be developed. As an important support of floating production facilities (e.g., tension leg platform (TLP), semisubmersible platform, spar platform,

floating production storage and offloading (FPSO)), the subsea production system plays a dominated role in the exploration of deepwater oil and gas fields.

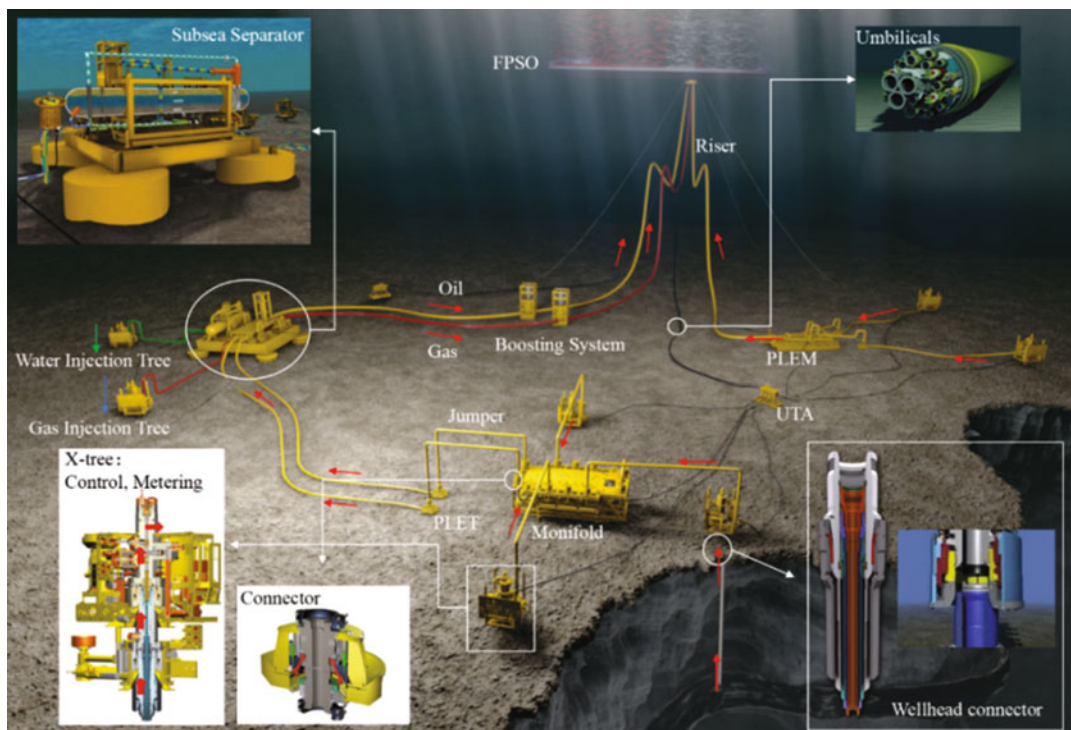
As the oil and gas fields move further offshore into deeper water and deeper geological formations in the quest for reserves, the technology of drilling and production has advanced dramatically. The latest subsea technologies have been proven and formed into an engineering system, namely, the subsea production system, which is associated with the overall process and all the equipment involved in drilling, field development, and field operation. The subsea production system consists of the following components (Fig. 1):

- Christmas trees
- Processing systems
- Tie-in and pipeline systems
- Umbilical and riser systems
- Power and control systems
- Subsea structures and manifold system

The most advantages of the SPS implement are economic superiority and time efficiency. Additionally, it is insensitive to water depth and is not easily affected by the environments of wind and waves in the sea surface. These merits make the SPS a system with good safety and high reliability, as well as wide application range.

Christmas Trees

Christmas trees are used on both surface and subsea wells. It is common to identify the type of tree as either “subsea tree” or “surface tree.” Each of these classifications has a number of variations. Examples of subsea include conventional, dual bore, mono bore, TFL (through flowline), horizontal, mudline, mudline horizontal, side valve, and TBT (through-bore tree) trees. The deepest installed subsea tree is in the Gulf of Mexico at approximately 9000 ft (2700 m). (Current technical limits are up to around 3000 m and working temperatures of –50 °F to 350 °F with a pressure of up to 15,000 psi.)



Subsea Production System, Fig. 1 Composition of subsea production system (Bai and Bai 2019)

The primary function of a tree is to control the flow, usually oil or gas, out of the well. (A tree may also be used to control the injection of gas or water into a nonproducing well in order to enhance production rates of oil from other wells.) When the well and facilities are ready to produce and receive oil or gas, tree valves are opened, and the formation fluids are allowed to go through a flowline. This leads to a processing facility, storage depot, and/or other pipeline eventually leading to a refinery or distribution center (for gas). Flowlines on subsea wells usually lead to a fixed or floating production platform or to a storage ship or barge, known as a floating storage offloading (FSO) vessel, or floating processing unit (FPU), or floating production storage and offloading (FPSO) vessel.

A tree often provides numerous additional functions including chemical injection points, well intervention means, pressure relief means, monitoring points (such as pressure, temperature, corrosion, erosion, sand detection, flow rate, flow composition, valve and choke position

feedback), and connection points for devices such as downhole pressure and temperature transducers (DHPT). On producing wells, chemicals or alcohols or oil distillates may be injected to preclude production problems (such as blockages).

Functionality may be extended further by using the control system on a subsea tree to monitor, measure, and react to sensor outputs on the tree or even down the well bore. The control system attached to the tree controls the downhole safety valve (SCSSV, DHSV, SSSV) while the tree acts as an attachment and conduit means of the control system to the downhole safety valve. Tree complexity has increased over the last few decades. They are frequently manufactured from blocks of steel containing multiple valves rather than being assembled from individual flanged components. This is especially true in subsea applications where the resemblance to Christmas trees no longer exists given the frame and support systems into which the main valve block is integrated.

Note that a tree and wellhead are separate pieces of equipment not to be mistaken as the same piece. The Christmas tree is installed on top of the wellhead. A wellhead is used without a Christmas tree during drilling operations, and also for riser tieback situations that later would have a tree installed at riser top. Wells being produced with rod pumps (pump jacks, nodding donkeys, grasshopper pumps, and so on) frequently do not utilize any tree owing the absence of a pressure-containment requirement.

Processing Systems

Subsea processing is used to increase the hydrocarbon production in a cost-optimized way for new field developments, tiebacks, and existing facilities. It can also enable production in marginal fields. The main types of subsea processing include combinations of oil-water separation, gas/liquid separation, gas-liquid-multiphase boosting, and water injection in the oil or gas field's wellstream on the seafloor. The systems can:

- Increase hydrocarbon recovery
- Optimize field development solutions
- Debottleneck existing production facilities (risers, topside process capacity)
- Allow subsea produced water separation and reinjection
- Allow subsea gas/liquid separation and boosting
- Enable long-distance subsea tiebacks
- Allow production in harsh environments and remote locations
- Allow seawater treatment for injection

While subsea developments have been made possible by technologies such as subsea trees, risers, and umbilicals, subsea processing has been an elusive solution for many years. Whether describing subsea separation, reinjection, or boosting, subsea processing has helped to revolutionize offshore developments worldwide.

With production equipment located on the seafloor rather than on a fixed or floating platform,

subsea processing provides a less expensive solution for myriad offshore environments. Originally conceived as a way to overcome the challenges of extremely deepwater situations, subsea processing has become a viable solution for fields located in harsh conditions where processing equipment on the water's surface might be at risk. Additionally, subsea processing is an emergent application to increase production from mature or marginal fields.

Subsea processing can encompass a number of different processes to help reduce the cost and complexity of developing an offshore field. The main types of subsea processing include subsea water removal and reinjection or disposal, single-phase and multiphase boosting of well fluids, sand and solid separation, gas/liquid separation and boosting, and gas treatment and compression.

Tie-In and Pipeline Systems

Subsea flowlines are used for the transportation of crude oil and gas from subsea wells, manifolds, offshore process facilities, loading buoys, S2B (subsea to beach), as well as reinjection of water and gas into the reservoir. Achieving successful tie-in and connection of subsea flowlines is a vital part of a subsea field development.

Vertical Tie-In Systems

Vertical connections are installed directly onto the receiving hub in one operation during tie-in. Since the Vertical Connection System does not require a pull-in capability, it simplifies the tool functions, provides a time-efficient tie-in operation, and reduces the length of rigid spools.

Horizontal Tie-In Systems

Horizontal tie-in may be used for both first-end and second-end tie-in of both flowlines, umbilicals and jumper spools. The termination head is hauled in to the tie-in point by use of a subsea winch. Horizontal tie-in may be made up by clamp connectors operated from a tie-in tool, by integrated hydraulic connectors operated through the ROV, or by non-hydraulic collet connectors

with assistance from a connector actuation tool (CAT) and ROV. Horizontal connections leave the flowline/umbilical in a straight line, and it is easy to protect if over trawling from fishermen should occur.

Pipeline Systems

Subsea flowlines are the subsea pipelines used to connect a subsea wellhead with a manifold or the surface facility. The flowlines may be made of flexible pipe or rigid pipe, and they may transport petrochemicals, lift gas, injection water, and chemicals. For situations where pigging is required, flowlines are connected by crossover spools and valves configured to allow pigs to be circulated. Flowlines may be single pipe, or multiple lines bundled inside a carrier pipe. Both single and bundled lines may need to be insulated to avoid problems associated with the cooling of the produced fluid as it travels along the seabed. Subsea flowlines are increasingly being required to operate at high pressures and temperatures. The higher pressure condition results in the technical challenge of providing a higher material grade of pipe for high-pressure, high-temperature (HP/HT) flowline projects, which will cause sour service if the product includes H₂S and saltwater. In addition, the higher-temperature operating condition will cause the challenges of corrosion, down-rated yield strength, and insulation coating. Flowlines subjected to HP/HT will create a high effective axial compressive force due to the high fluid temperature and internal pressure that rises when the flowline is restrained (Davalath et al. 2002).

Umbilical and Riser Systems

Umbilical Systems

Subsea umbilicals are deployed on the seabed (ocean floor) to supply necessary control, energy (electric, hydraulic), and chemicals to subsea oil and gas wells, subsea manifolds, and any subsea system requiring remote control, such as a remotely operated vehicle. Subsea intervention umbilicals are also used for offshore drilling or workover activities.

Riser Systems

The production riser is the portion of the flowline that resides between the host facility and the seabed adjacent to a host facility. Riser dimensions range from 3 to 24 (76.2–609.6 mm) in diameter. Riser length is defined by the water depth and riser configuration, which can be vertical or a variety of waveforms. Risers can be flexible or rigid. They can be contained within the area of a fixed platform or floating facility, run in the water column.

Subsea Control Module (SCM)

The SCM is an independently retrievable unit. SCMs are commonly used to provide well control functions during the production phase of subsea oil and gas production. Typical well control functions and monitoring provided by the SCM are as follows:

- Actuation of fail-safe return production tree actuators and downhole safety valves
- Actuation of flow control choke valves, shutoff valves, etc.
- Actuation of manifold diverter valves, shutoff valves, etc.
- Actuation of chemical injection valves
- Actuation and monitoring of surface-controlled reservoir analysis and monitoring systems, sliding sleeve, choke valves
- Monitoring of downhole pressure, temperature, and flow rates
- Monitoring of sand probes and production tree and manifold pressures, temperatures, and choke positions

SCM Components

The typical SCM receives electrical power, communication signals, and hydraulic power supplies from surface control equipment. The subsea control module and production tree are generally located in a remote location relative to the surface control equipment. Redundant supplies of communication signals and electrical and hydraulic power are transmitted through umbilical hoses and cables ranging from 1000 ft to several miles in length, linking surface equipment to subsea

equipment. Electronics equipment located inside the SCM conditions the electrical power, processes communication signals, transmits status, and distributes power to solenoid piloting valves, pressure transducers, and temperature transducers.

The subsea electrical module (SEM) is fed with AC power directly from the EPU. It normally incorporates two AC-DC converters and output to DC bus bars (typically 24 and 5 V). Actuation of the electrohydraulic valves is tapped into a 24-bus bar, while the sensors are tapped into a 5-V bus bar.

Low-flow-rate solenoid piloting valves are typically used to pilot high-flow-rate control valves. These control valves transmit hydraulic power to end devices such as subsea production tree valve actuators, choke valves, and downhole safety valves. The status condition of control valves and their end devices is read by pressure transducers located on the output circuit of the control valves.

Auxiliary equipment inside the typical SCM consists of hydraulic accumulators for hydraulic power storage, hydraulic filters for the reduction of fluid particulates, electronics vessels, and a pressure/temperature compensation system. Previous devices have used an oil-filled chamber to compensate for hydrostatic pressure increases outside of the device during use to keep seawater away from cable assemblies.

The SCM is typically provided with a latching mechanism that extends through the body of the SCM and that has retractable and extendable dogs or cams thereon to engage a mating receptacle in a baseplate.

SCM Control Mode Description

The SCM utilizes single LP and HP hydraulics, supplied from the umbilical system. Each supply has a “last chance” filter before supply is manifolded to the main supply and pilot supply ports of the solenoid-operated pilot valves.

The LP supply has an accumulator that is mounted internally to the SCM. The HP supply may also have an accumulator. A pressure transducer measures the HP and LP supply pressures for display at the MCS. There are four pilot valves: three on the LP for tree valves and choke

and one on the HP for the downhole safety valve (DHSV), usually referred to the surface-controlled subsurface safety valve (SCSSV). The hydraulic return line has a nonreturn valve to vent used fluid to the sea and prevent seawater ingress.

Valve Actuation

Most of the pilot valves have two solenoids to operate them, one to open and one to close. The solenoids are driven by the solenoid drivers in the SEM.

To open a tree valve, the appropriate solenoid is commanded from the MCS, and the microprocessor in the SEM activates the solenoid driver, which energizes the open solenoid for 2 s. (typically; this may be adjusted at the MCS). This allows hydraulic fluid to flow into the function line to the tree valve actuator. The pressure in this line will rise very quickly to a value that allows the valve to latch open hydraulically. Thereafter, the valve will remain open as long as the hydraulic supply pressure remains above about 70 bar.

To close a tree valve, the close solenoid is energized in a similar manner to the open solenoid. This causes the spool in the pilot valve to move, venting the hydraulic fluid from the tree valve actuator. The used fluid is vented to sea via a nonreturn valve.

Choke Operation

The choke has two hydraulic actuators, one to open and one to close, that move the choke via a pawl and ratchet mechanism.

The choke is moved by applying a series of hydraulic pressure pulses to the appropriate actuator. On each pressurize/vent cycle, the choke will move by one step. To provide these pressure pulses, the SCM has two pilot valves, one for opening the choke and one for closing the choke. These valves are not hydraulically latched and will only pass hydraulic fluid when the solenoid is energized.

To operate the choke, the MCS sends a series of commands to the SCM. For example, if the operator wishes to move the choke by 20 steps, the MCS will send 20 appropriately timed valve commands to the SCM to energize the appropriate solenoid.

Subsea Structures and Manifold System

Subsea manifold is a flow-routing subsea hardware (subsea flow router) that connects between subsea trees and flowlines. It is used to optimize the subsea layout arrangement and reduce the quantity of risers connected to the platform. If connected to dual flowlines, the manifold can typically accommodate pigging and have the capability of routing production from a particular tree to a particular flowline (Laurent et al. 2006).

Subsea Manifold Types

There are several manifold types and it is the field layout that determines the type of manifold required.

- Template manifold

Template manifold is a drill-through structure designed to house multiple subsea Christmas trees on top of it and gather/route flow at the same time. It is required when subsea Christmas trees are grouped side by side. Trees are connected directly into the manifold via mandrels when installed.

Template manifold must be fabricated and installed before the start of drilling, which gives less schedule flexibility. It does not allow the reuse of exploration wells because they are typically not grouped together. Statoil's Norne in North Sea has template manifold.

- Cluster manifold

Cluster manifold is a stand-alone structure designed to direct fluids for multiple subsea Christmas trees placed around it. It typically can accommodate 4–8 wells per manifold. Both the TOTAL and Shell's manifolds shown in the picture are cluster manifolds. When subsea Christmas trees are grouped closely in a central location (but not side by side), either a cluster manifold or a PLEM is required depending on the number of trees anticipated. A Daisy Chain field layout has the same manifold requirements.

- Pipeline end manifold (PLEM)

For large-diameter trunk lines, PLEMs may be necessary to support the tie-in spool connection

systems/SSIV and hence aren't always associated with wellheads. For example, some of the large LNG projects use a PLEM to host multiple flexible gas risers and the diverless pig launching facilities which then connect to the gas export pipeline via a PLET and diverless-installed spool. A PLEM generally connects directly to a subsea flowline without the use of a pipeline end termination (PLET).

Manifold Components

A manifold is typically composed of the following major components:

- Pipework and valves – contain and control the production and injection fluids.
- Structure framework – protects and supports the pipework and valves.
- Subsea connection equipment – allows subsea tie-in of multiple pieces of equipment. Types include vertical, horizontal, and stab-and-hinge-over connections.
- Foundation – interface between the manifold structure and seabed.
- Controls equipment – allows the remote control of any hydraulically actuated subsea manifold valves and the monitoring of production and injection fluids. Control pods may be either internal or external to the manifold.

Framework Structure

The framework is a welded structure to provide support for the pipework and valves and contain the foundation interface structure. The pipework is allowed to float inside the framework within limits and it is not rigidly attached to the frame. The frame can also be used for lifting and landing of the jumper tie-in tools. In addition, a major function of the framework structure is often to provide mechanical protection to the internal pipework and equipment from third-party interactions, as well as often aiming to facilitate over-trawling by fishing gear.

Cross-References

- [Design of Pipelines and Risers](#)
- [Installation and Decommissioning](#)
- [Subsea Production System](#)

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Subsea Production System (SPS)

► Suction Piles

Subsea Production Systems

► Umbilical Cable

Subsea System

► Subsea Production System

Suction Anchor

► Mooring Anchor

Suction Piles

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Synonyms

Arbitrary Lagrangian-Eulerian (ALE); Coupled Eulerian-Lagrangian (CEL); Interpolation

technique by small strain (RITSS); Large deformation analysis (LDA); Subsea production system (SPS)

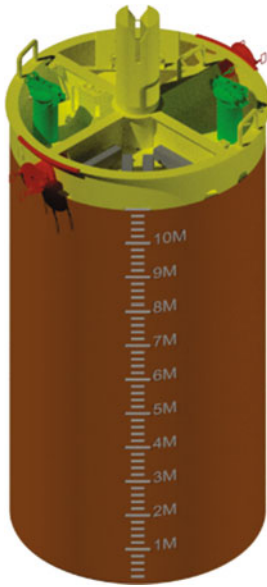
Definition

In general, a typical suction pile structure is a closed-top, open-bottom steel structure or a steel-concrete composite structure with a closed drain/air outlet at one end and a pump-connected suction outlet (Figs. 1 and 2). The suction pile is similar to the buckled bucket, so it is also called bucket foundation. It has many advantages such as (a) precise positioning in deep water, (b) ease of removal and relocation, (c) less dependency on heavy installation equipment, and (d) large diameter that can develop considerable resistance against significant loads generated by top structures, so it is widely used in the foundation structure of subsea production system (SPS), which is suitable for deep and ultra-deep subsea environment.

When suction pile works, it sinks to a certain depth under the action of dead weight firstly. With the help of pump system, it sucks out the water inside the pile barrel to form negative pressure in the closed barrel body, thus generating downward



Suction Piles, Fig. 1 Suction pile. (<http://www.cranemaster.com/wp-content/uploads/2016/11/References-offshore-installation-suction-pile-bourbon-passive-heave-compensation-02.jpg>)



Suction Piles, Fig. 2 Suction pile structural representation

thrust and pressing the pile into the predetermined depth in the soil.

When the suction pile is recovered, the pump system works in reverse and pours water into the pile barrel to make the internal pressure greater than the external pressure. The positive pressure difference between inside and outside is used to push the pile out of the mud surface.

The design of suction pile should consider the following key factors: design of overall lifting facilities for suction pile jacket; the maximum yaw angle of jacket transport vehicle, vessel, and transport process; college core for suction pile lifting; suction pile jacket installation; and anti-erosion measures for suction pile foundation.

Scientific Fundamentals

History of Suction Pile

The innovative idea of using negative pressure sinking was firstly introduced by Statoil in the Gullfaks field in the North Sea in 1985 (Tjelta et al. 1986). It conducted a large-scale offshore penetration test so as to gained valuable experience in the use of skirt foundations on

gravity-based platforms using negative pressure. To further understand the performance of such foundations, a series of model tests combining static and cyclic loads to simulate the loading of subsea platforms are carried out. In 1990, Norwegian Aker Engineering Company proposed the use of under pressure caisson piles instead of the traditional pile foundation, and the entire design was completed in 1994 after a series of on-site insertion tests were carried out on the actual seabed to increase the understanding of the structural form. The successful installation of the Europipe 16/11E jacket in the North Sea in July 1994 marked the entry of this technology into the practical phase of industrialization (Bye et al. 1995), and the first suction pile moorings for permanent use were installed 1995 in 100–200 m water depth and for catenary anchor lines (Sparrevik 1996), and the first step into deep water using suction piles moorings was taken in Brazil in 1997 for Petro-bras P-19 and P-26 floaters at the Marlin field (Mello et al. 1998). Today, the suction pile is one of the most common foundation forms of SPS, and there are hundreds of suction piles operative in permanent mooring systems around the world.

Working Principle of Suction Pile

Suction pile is installed with the aid of suction (under pressure) created within the cylinder. The creation of the suction can be with the aid of a suction source, such as a pump, that is mounted on, close to, or at a distance from the suction pile. The applied level of the suction can be substantially constant, smoothly increase or decrease, or comprise a number of pulses. After use, the suction pile can be removed by creating an overpressure within the cylinder by pumping in water.

Suction pile is somewhat misleading since it is also used for short piles with large diameter. The aspect ratio L/D (length over diameter) for some suction piles is close to 1 and is clearly not a pile in a classical sense. Suction pile has a number of advantages over conventional offshore foundations, mainly being quicker to install than pile and being easier to remove during decommissioning. This methodology is now used extensively worldwide for anchoring large offshore installations to the seafloor at great depths,

or where the seabed is soft clays or low strength sediments.

Installation of Suction Pile

Flow chart of suction pile installation process (Fig. 3):

The installation of suction pile usually has three steps as shown in Fig. 4: deployment, self-penetration, and embedment.

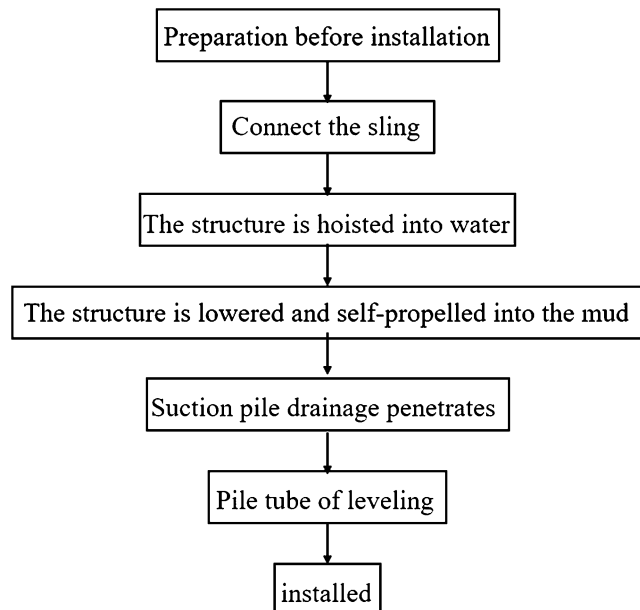
In the first deployment step, the suction pile was handled, transported, and installed by an off-shore construction vessel, with or without an A-frame. Since gravity is required to insert into the seabed at the initial stage of installation, a thick steel plate or concrete layer is usually used as the counterweight at the closed end. The side wall of the barrel is rolled with steel plate. In order to prevent buckling of the barrel wall, it is usually necessary to weld various kinds of strong ribs in the barrel body. The installation vessel first approached the suction pile, lifted it up, moved it out to the open water, and then lowered the suction pile from the water surface to the water bottom, removably securing the pumping unit to the top of suction pile either before or after the lowering of the pile. Loads are resisted through the structure with mooring padeyes or pile top footings to the soil via direct bearing and skin friction.

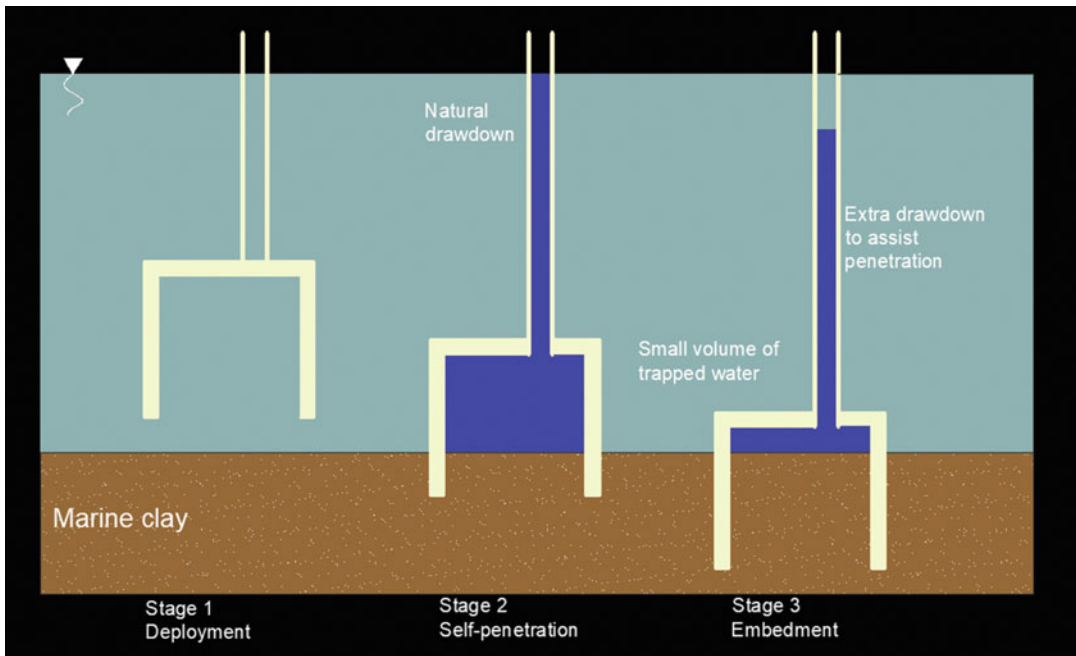
Next step is self-penetration; the suction pile is placed on the bottom of the sea, and the pile's lower edge is embedded in the soil by the weight of the pile. When the suction pile was correctly positioned and oriented, the crane further lowered the pile slowly until the pile penetration into the seafloor soil stopped due to its own weight. The suction pile can penetrate up to 60% of its length under its own weight, depending on soil conditions and the pile properties.

Finally is the embedment step, the pump is pumped outward with the help of the submersible pump installed on the pile cap and a remote-operated vehicle (ROV). At the same time, the amount of water pumped out exceeds the amount of water infiltrated from the bottom, resulting in the reduction of the internal pressure of the pile tube. When the vertical downward pressure caused by the pressure difference between the inside and outside of the pile cylinder exceeds the resistance of the soil to the pile cylinder, the pile cylinder can be continuously pressed into the soil until the top surface of the pile cylinder contacts with the seabed, and the pile sinking stops. At this point, the submersible pump can be removed, and the pressure difference between the inside and outside of the pile cylinder gradually disappears. The pressure in the cylinder reverts to the same pressure as the environment.

Suction Piles,

Fig. 3 Installation technology of suction pile





Suction Piles, Fig. 4 Suction pile installation

Key Technology

Calculation Methods for Bearing Capacity

The axial and lateral bearing capacities of suction piles need to be evaluated in the design process, and there are three methods which normally used in the evaluation: limit equilibrium method, semi-empirical beam column analysis method, and finite element method (Wang and Pan 2011).

Limit equilibrium method: The determination of reasonable failure mode of suction pile is mainly based on the combination of experimental observation, numerical analysis, and engineering experience. The disadvantage of this method is that the assumption is introduced and the analysis is approximate. The theoretical solution has some limitations. However, this method is the theoretical basis for the calculation of the bearing capacity of suction pile. The sliding surface is assumed first, and the ultimate bearing capacity is calculated according to static equilibrium and failure criteria.

Semiempirical beam column analysis method: It is called semiempirical method because the beam column method relies on a series of

empirical rules to express the soil resistance without defining the failure mechanism. Instead, the soil resistance is expressed by a series of uncoupled, nonlinear soil springs (P-y curves) along the pile length. Beam column method is based on the integration of experimental and analytical results, and it is difficult to obtain general results.

Finite element method: The finite element method program can automatically find the critical failure mechanism with high calculation accuracy and can consider nonlinear factors, which can be used to simulate complex soil properties. Of course, there are also corresponding requirements, for example, advanced numerical analysis requires specific knowledge, and the establishment of appropriate models takes a long time.

Large Deformation Analysis (LDA) of Penetration Process

The penetration process of suction pile installation has one of the most challenging topics in computational geomechanics which is LDA, particularly in problems involving complicated structure-soil interaction. The total Lagrangian and the update

Lagrangian finite element approaches cannot solve this problem due to the mesh being seriously distorted. To capture large deformation phenomena that occur frequently in suction pile penetration process, the traditional numerical approaches established within Lagrangian framework are replaced by, for example, those based on the framework of arbitrary Lagrangian-Eulerian (ALE).

Among a variety of ALE approaches, three FE methods are widely used in research and industry for analysis of geotechnical engineering problems: the remeshing and interpolation technique by small strain (RITSS), an efficient ALE (EALE) approach, and the coupled Eulerian-Lagrangian (CEL) approach (Wang et al. 2015).

The RITSS approach was originally presented by Hu and Randolph (1998), in which the deformed soil is remeshed periodically and Lagrangian calculation is implemented through an implicit time integration scheme. The advantage of RITSS is that the remeshing and interpolation strategy can be coupled with any standard FE program, through user-written interface codes.

The EALE approach is based on the operator split technique proposed by Benson (1989). This method is a well-known variant of r-adaptive FE methods, which have been designed to eliminate possible mesh distortion by changing and optimizing the location of nodal points without modifying the topology of the mesh.

In the CEL method, the element nodes move temporarily with the material during a Lagrangian calculation phase, which is followed by mapping to a spatially “fixed” Eulerian mesh. The calculation in the Lagrangian phase is conducted with an explicit integration scheme. In contrast to RITSS and EALE, an element in CEL may be occupied by multiple materials fully or partially, with the material interface and boundaries approximated by volume fractions of each material in the element. The comparatively rigid structural part is usually modeled as a Lagrangian body and the soils as Eulerian materials. A “general contact” algorithm by means of an enhanced immersed boundary method describes frictional contact between Lagrangian and Eulerian materials.

Typical Application

Dalia SPS

The SPS of Dalia (Lafitte et al. 2007) was the largest subsea order in offshore history at that time in 2002, with 71 wells, 71 drill through horizontal trees, 9 6-slot production manifolds, 2 work-over systems, and a complex control and chemical injection system. Deepwater environment with an external pressure of up to 145 bars with soft soil conditions requires installation of subsea equipment using suction piles. Each manifold assembly consists of a foundation bottom structure and manifold. The foundation bottom structure comprises a suction pile and a manifold support structure, which is adjustable in level to ± 5 degrees. The size of its suction pile is diameter $6.2 \text{ m} \times 11.8 \text{ m}$ and 31 tonnes (Fig. 5).

BP King Subsea Pump

BP King is an oil field located in the Gulf of Mexico. It consists of three wells, and the wells are located in water depths ranging from 1520 to



Suction Piles, Fig. 5 SPS manifold module with suction pile

1655 m. At the time of 2005, the deepest installed subsea multiphase pump was situated in 900 m of water, and the longest step-out was 9 km. The pump equipment suitable for a depth of approximately 1700 m and a step-out of 27 km were required in the King pump project. In November 2007, the BP King subsea pumps (Davis et al. 2009) were commissioned in record-breaking water depth of 1700 m offshore Louisiana. Two twin-screw multiphase pumps were installed with a 27 km tieback to the Marlin Tension Leg Platform. The King pump systems comprise of a pump manifold and a separately retrievable multiphase pump module; they were installed as fully assembled units onto the suction pile foundations.

Perdido Development

The Perdido (Denney 2010) development includes the Great White, Silvertip, and Tobago fields and is in the Perdido basin and fold belt in the Alaminos Canyon protraction area. This area is 200 miles south of Freeport, Texas, USA, and 8 miles north of the Mexico maritime border. All three fields are developed with subsea wells tied back to a spar with full offshore processing capabilities and pipelines for export. For the Perdido development, two production manifolds were

located under the host spar structure. Each manifold has two main headers, and each header is connected to one of the five artificial lift subsea boosting systems. In addition, 22 wells were batch set before the manifolds, so suction piles were used to provide the foundation support, instead of the traditional mudmat in order to minimize the need of an extensive flat level area with minimum elevation tolerance.

Cascade and Chinook Subsea Development

Cascade and Chinook (Masson et al. 2011) are two separate fields which located in the Walker Ridge Outer Continental Shelf leasing area of the central Gulf of Mexico. The Cascade and Chinook Subsea Development risers are the deepest production risers in the world by 2011(2500 m depth,). And each field is independently developed but shared a FPSO (floating production, storage, and offloading) unit, which located between the two fields, and export infrastructure including two shuttle tankers and a gas export pipeline. The suction piles which diameters of 5.49 m (Fig. 6) were installed by Technip's Deep Blue pipe laying ship in August 2009. The riser suction piles are horizontally offset 371.86 m from the center of the FPSO turret buoy. Similarly,

Suction Piles, Fig. 6 The riser suction pile on supply boat



Suction Piles, Fig. 7 The hoisting of YC 13-4 gas field PLEM with three suction piles



the manifold and the pump basement in the Cascade and Chinook used the same suctions as the lower riser components, which was also performed with the Deep Blue and managed by the same project.

YC 13-4 Gas Field

YC 13-4 gas field (Huang et al. 2014) is located in the South China Sea and belongs to the marginal oil and gas field at the water depth about 89 m. According to the requirement of YC 13-4 gas field development, a new type underwater manifold which has the suction pile foundation was designed. The soil of seabed in YC 13-4 gas field is relatively soft, and the traditional method of plate-pile fixed is not reliable. Therefore, both PLET and PLEM have three suction piles. The whole structure is connected by three independent suction piles through the frame. The advantages of suction pile fixation are fully verified in the installation process. A guide column and a lock pin structure are arranged on the suction pile for fixing the connection with the upper pipe manifold, which can realize leveling well and prevent the fishing net from towing. It can be well adjusted to the level and prevent from the fishing nets dragging (Fig. 7).

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Super Short Baseline (SSBL)

- Long Baseline Underwater Acoustic Location Technology
- Ultra-short Baseline Underwater Acoustic Location Technology

Super-Short-Baseline System (SSBL)

- Obstacle Avoidance Technology for Underwater Vehicle

Surface Buoy

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Synonyms

SMB: Surface Marker Buoy

Definition

A surface buoy is a device floating on the water surface and can have many purposes. It can be anchored (stationary) or allowed to drift with ocean currents. There are many types of buoys for different purposes.

Scientific Fundamentals

To keep the surface buoy floating on the water surface, there must be some forces acting on it, among which the most important force is buoyancy.

Archimedes' Principle

Archimedes' principle is named after Archimedes of Syracuse, who first discovered this law in 212 B.C (Acott 1999). For objects, floating and sunken, and in gases as well as liquids (i.e., a fluid), Archimedes' principle may be stated thus in terms of forces: Any object, wholly or partially immersed in a fluid, is buoyed up by a force equal to the weight of the fluid displaced by the object with the clarifications that for a sunken object the volume of displaced fluid is the volume of the object, and for a floating object on a liquid, the weight of the displaced liquid is the weight of the object.

More tersely : Buoyancy = weight of displaced fluid.

Archimedes' principle does not consider the surface tension (capillarity) acting on the body (Floater clustering 2005), but this additional force modifies only the amount of fluid displaced and the spatial distribution of the displacement, so the principle that Buoyancy = weight of displaced fluid remains valid.

The weight of the displaced fluid is directly proportional to the volume of the displaced fluid (if the surrounding fluid is of uniform density). In simple terms, the principle states that the buoyancy force on an object is equal to the weight of the fluid displaced by the object, or the density of the fluid multiplied by the submerged volume times the gravitational acceleration, g . Thus, among completely submerged objects with equal masses, objects with greater volume have greater buoyancy. This is also known as upthrust.

Assuming Archimedes' principle to be reformulated as follows,

$$\begin{aligned} & \text{apparent immersed weight} \\ &= \text{weight of object} \\ & \quad - \text{weight of displaced fluid} \end{aligned} \quad (1)$$

then inserted into the quotient of weights, which has been expanded by the mutual volume

$$\frac{\text{density of object}}{\text{density of fluid}} = \frac{\text{weight}}{\text{weight of displaced fluid}} \quad (2)$$

yields the formula below. The density of the immersed object relative to the density of the fluid can easily be calculated without measuring any volumes:

$$\frac{\text{density of object}}{\text{density of fluid}} = \frac{\text{weight}}{\text{weight} - \text{apparent immersed weight}} \quad (3)$$

(This formula is used for example in describing the measuring principle of a dasymeter and of hydrostatic weighing.)

Forces and Equilibrium

The equation to calculate the pressure inside a fluid in equilibrium is:

$$f + \text{div } \sigma = 0 \quad (4)$$

where f is the force density exerted by some outer field on the fluid and σ is the Cauchy stress tensor. In this case the stress tensor is proportional to the identity tensor:

$$\sigma_{ij} = -p\delta_{ij} \quad (5)$$

Here δ_{ij} is the Kronecker delta. Using this the above equation becomes:

$$f = \nabla p \quad (6)$$

Assuming the outer force field is conservative, that is it can be written as the negative gradient of some scalar valued function:

$$f = -\nabla\Phi \quad (7)$$

Then:

$$\nabla(p + \Phi) = 0 \Rightarrow p + \Phi = \text{const} \quad (8)$$

Therefore, the shape of the open surface of a fluid equals the equipotential plane of the applied outer conservative force field. Let the z -axis point downward. In this case the field is gravity, so $\Phi = -\rho_f g z$ where g is the gravitational acceleration, ρ_f is the mass density of the fluid. Taking the pressure as zero at the surface, where z is zero, the constant will be zero, so the pressure inside the fluid, when it is subject to gravity, is

$$p = \rho_f g z \quad (9)$$

So pressure increases with depth below the surface of a liquid, as z denotes the distance from the surface of the liquid into it. Any object with a nonzero vertical depth will have different pressures on its top and bottom, with the pressure on the bottom being greater. This difference in pressure causes the upward buoyancy force.

The buoyancy force exerted on a body can now be calculated easily, since the internal pressure of the fluid is known. The force exerted on the body can be calculated by integrating the stress tensor over the surface of the body which is in contact with the fluid:

$$\mathbf{B} = \oint \sigma d\mathbf{A} \quad (10)$$

The surface integral can be transformed into a volume integral with the help of the Gauss theorem:

$$\begin{aligned} \mathbf{B} &= \oint \text{div } dV = - \int \mathbf{f} dV = -\rho_f \mathbf{g} \int dV \\ &= -\rho_f \mathbf{g} V \end{aligned} \quad (11)$$

where V is the measure of the volume in contact with the fluid, that is the volume of the submerged part of the body, since the fluid does not exert force on the part of the body which is outside of it.

The magnitude of buoyancy force may be appreciated a bit more from the following argument. Consider any object of arbitrary shape and volume V surrounded by a liquid. The force the liquid exerts on an object within the liquid is equal

to the weight of the liquid with a volume equal to that of the object. This force is applied in a direction opposite to gravitational force, that is, of magnitude:

$$B = \rho_f V_{disp} g \quad (12)$$

where ρ_f is the density of the fluid, V_{disp} is the volume of the displaced body of liquid, and g is the gravitational acceleration at the location in question.

If this volume of liquid is replaced by a solid body of exactly the same shape, the force the liquid exerts on it must be exactly the same as above. In other words, the “buoyancy force” on a submerged body is directed in the opposite direction to gravity and is equal in magnitude to

$$B = \rho_f V g \quad (13)$$

The net force on the object must be zero if it is to be a situation of fluid statics such that Archimedes principle is applicable and is thus the sum of the buoyancy force and the object’s weight

$$F_{net} = 0 = mg - \rho_f V_{disp} g \quad (14)$$

If the buoyancy of an (unrestrained and unpowered) object exceeds its weight, it tends to rise. An object whose weight exceeds its buoyancy tends to sink. Calculation of the upwards force on a submerged object during its accelerating period cannot be done by the Archimedes principle alone; it is necessary to consider dynamics of an object involving buoyancy. Once it fully sinks to the floor of the fluid or rises to the surface and settles, Archimedes principle can be applied alone. For a floating object, only the submerged volume displaces water. For a sunken object, the entire volume displaces water, and there will be an additional force of reaction from the solid floor.

In order for Archimedes’ principle to be used alone, the object in question must be in equilibrium (the sum of the forces on the object must be zero), therefore,

$$mg = \rho_f V_{disp} g \quad (15)$$

and therefore

$$m = \rho_f V_{disp} \quad (16)$$

showing that the depth to which a floating object will sink, and the volume of fluid it will displace, is independent of the gravitational field regardless of geographic location.

(Note: If the fluid in question is seawater, it will not have the same density (ρ) at every location. For this reason, a ship may display a Plimsoll line.)

It can be the case that forces other than just buoyancy and gravity come into play. This is the case if the object is restrained or if the object sinks to the solid floor. An object which tends to float requires a tension restraint force T in order to remain fully submerged. An object which tends to sink will eventually have a normal force of constraint N exerted upon it by the solid floor. The constraint force can be tension in a spring scale measuring its weight in the fluid and is how apparent weight is defined.

If the object would otherwise float, the tension to restrain it fully submerged is:

$$T = \rho_f V g - mg \quad (17)$$

When a sinking object settles on the solid floor, it experiences a normal force of:

$$N = mg - \rho_f V g \quad (18)$$

Another possible formula for calculating buoyancy of an object is by finding the apparent weight of that particular object in the air (calculated in Newtons), and apparent weight of that object in the water (in Newtons). To find the force of buoyancy acting on the object when in air, using this particular information, this formula applies:

$$\begin{aligned} \text{Buoyancy force} = & \text{weight of object in empty space} - \\ & \text{weight of object immersed in fluid} \end{aligned} \quad (19)$$

The final result would be measured in Newtons.

Air's density is very small compared to most solids and liquids. For this reason, the weight of an object in air is approximately the same as its true weight in a vacuum. The buoyancy of air is neglected for most objects during a measurement in air because the error is usually insignificant (typically less than 0.1% except for objects of very low average density such as a balloon or light foam) (Website [2018a](#)).

Key Applications

Sea Mark Buoy

A sea mark buoy is a form of aid to navigation and pilotage that identifies the approximate position of a maritime channel, hazard, or administrative area to allow boats, ships, and seaplanes to navigate safely. Some navigational buoys are fitted with a bell or gong, which sounds when waves move the buoy. The sea mark buoy consists of a floating object that is usually anchored to a specific location on the bottom of the sea or to a submerged object.

Sea mark buoys are used to indicate channels, dangerous rocks or shoals, mooring positions, areas of speed limits, traffic separation schemes, submerged shipwrecks, and for a variety of other navigational purposes. Some are only intended to be visible in daylight (daymarks), and others have some combination of lights, reflectors, bells, horns, whistles, and radar reflectors to make them usable at night and in conditions of reduced visibility. Figure 1 shows a red sea mark buoy in San Diego Harbor, with a light, number, and radar corner reflectors (Website [2018b](#)).

Lifebuoy

A lifebuoy is a lifesaving buoy designed to be thrown to a person in the water, to provide buoyancy and prevent drowning. Some modern lifebuoys are fitted with one or more seawater-activated lights, to aid rescue at night.

The lifebuoy is usually ring- or horseshoe-shaped and has a connecting line allowing the casualty to be pulled to the rescuer in a boat. They are carried by ships and are also located beside bodies of water that have the depth or



Surface Buoy, Fig. 1 Red sea mark buoy in San Diego Harbor, with a light, number, and radar corner reflectors. (Also with a lounging sea lion.) Located at 32.69561°N 117.23099°W (Website [2018b](#))

potential to drown someone. They are often subjected to vandalism which, since the unavailability of lifebuoys could lead to death, may be punished by fines (up to £5,000 in the United Kingdom) or imprisonment. Figure 2 shows a ring-shaped lifebuoy with a light on a cruise ship (Website [2018c](#)).

The “kisby ring,” or sometimes “Kisbie ring,” is thought to be named after Thomas Kisbee (1792–1877) who was a British naval officer.

The UK Royal Life Saving Society considers lifebuoys unsuitable for use in swimming pools because throwing one into a busy pool could injure the casualty or other pool users. In these locations, lifebuoys have been superseded by devices such as the torpedo buoy.

In the United States, Coast Guard approved lifebuoys are considered Type IV personal



Surface Buoy, Fig. 2 A ring-shaped lifebuoy with a light on a cruise ship (Website 2018c)

flotation devices. At least one Type IV PFD is required on all vessels 26 feet or more in length.

Large Navigational Buoy (LNB)

A Large Navigational Buoy (LNB) is a contraction of Large Automatic Navigation Buoy. LNBs were first made in the USA by General Dynamics and adapted by Hawker Siddeley Dynamics for use in British waters in the early 1970s. The buoys were intended to replace lightships and were constructed as a circular hull with a central light to provide all-round visibility and a foghorn. They may also contain radio and radio beacons. The navigation buoy was to be monitored remotely from onshore and was designed to run for extended periods without repair. The running costs were estimated to be as little as 10% of those of a lightship.

However, experience showed that it was difficult to attain the required reliability in British waters due to the high acceleration forces experienced in rough seas with 14 m waves and 7 knot currents. Alternative experiments were made with

platforms such as the Royal Sovereign Lighthouse. The automatic technology was later used successfully in more conventional lightships. Figure 3 shows a LNB (on left) that replaced Lightship Columbia at the Columbia River Maritime Museum in Oregon (Website 2018d).

Sonobuoy

A sonobuoy is used by antisubmarine warfare aircraft to detect submarines by SONAR. It is a relatively small buoy (typically 13 cm or 5 in, in diameter and 91 cm or 3 ft. long) expendable sonar system. Sonobuoys are ejected from aircraft in canisters and deploy upon water impact. An inflatable surface float with a radio transmitter remains on the surface for communication with the aircraft, while one or more hydrophone sensors and stabilizing equipment descend below the surface to a selected depth that is variable. The specific depth depends on environmental conditions and the search pattern. The buoy relays acoustic information from its hydrophone(s) via UHF/VHF radio to operators on board the aircraft. Figure 4 shows the sonobuoy deployment procedures after impacting water (Website 2018e).

Sonobuoys are classified into three categories: active, passive, and special purpose.

Surface Marker Buoy

A surface marker buoy, SMB, or simply a blob is a buoy used by scuba divers, with a line, to indicate the diver's position to their surface safety boat while the diver is underwater. Two kinds are used: one (SMB) is towed for the whole dive and indicates the position of the dive group, and the other (DSMB) is deployed towards the end of the dive as a signal to the surface that the divers have started to ascend. Both types can also function as a depth reference for controlling speed of ascent and accurately maintaining depth at decompression stops. Figure 5 shows the picture of a typical surface marker buoy (SMB) (Website 2018f).

SMB

SMBs are floated on the surface during a dive to mark the diver's position during drift dives, night dives, mist, or disturbed sea conditions such as



Surface Buoy, Fig. 3 LNB (on left) that replaced Lightship Columbia at the Columbia River Maritime Museum in Oregon (Website [2018d](#))

Beaufort force 2 or greater. The buoy lets the dive boat follow the divers and highlights their position to other boat traffic.

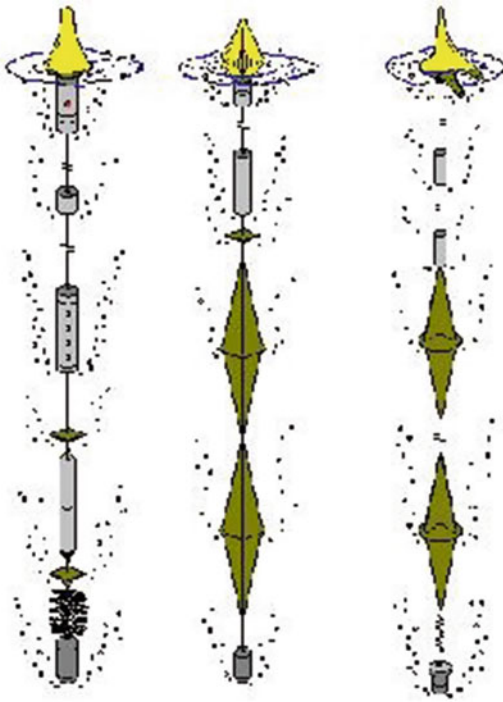
Buoys for this use are usually either inflated and sealed by a valve or cap, or made from buoyant material, so they do not deflate or flood during the dive, rendering them ineffective. High-visibility colors such as red, orange, and yellow are popular. Sometimes the float includes a diving flag. If the buoy is to be towed by the diver at any speed, a low drag float and small diameter line can reduce the drag significantly.

To avoid losing the reel, a lanyard may be used to attach the diving reel to the diver. This lanyard can clip to the buoyancy compensator or go around the wrist. Alternatively, the lanyard can be long enough to float above the diver and stay out of the way. If the lanyard clips to the buoyancy compensator, the user should take care to release if there is surface boating activity, as boats may drag divers up by their SMB reels.

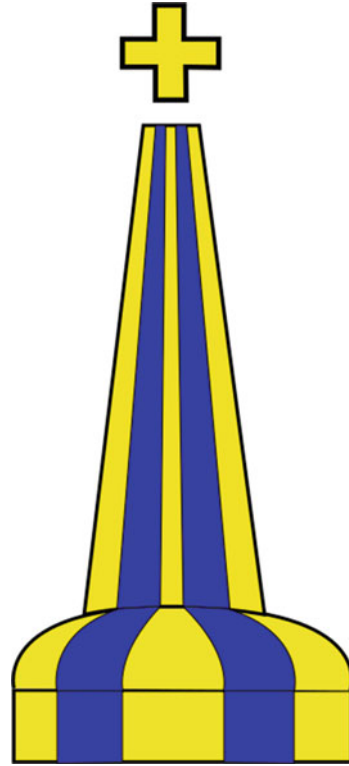
The DIR diving philosophy considers unsafe any attachment of the diver to equipment or objects which end above the water surface, due to high risk associated with snagging the buoy on a boat and dragging the diver upwards in spite of their decompression obligation or maximum ascent speed limit.

DSMB

A delayed surface marker buoy (DSMB), decompression buoy, or deco buoy is similar to a surface marker buoy but is deployed while the diver is submerged and generally only towards the end of the dive. The buoy marks the diver's position underwater so the boat safety cover can locate the diver even though the diver may have drifted some distance from the dive site while doing decompression stops. A reel and line connect the buoy on the surface to the diver beneath the surface. Alternative means of marking one's position while doing decompression stops are diving shots and decompression trapezes.



Surface Buoy, Fig. 4 Sonobuoy deployment procedures after impacting water (Website 2018e)



Surface Buoy, Fig. 6 Emergency wreck buoy (Website 2018g)



Surface Buoy, Fig. 5 Surface marker buoy (Website 2018f)

A closed SMB, inflated through a valve, is likely to be more reliable, by remaining inflated, than an open ended buoy which seals by holding the opening under water. A decompression buoy is not intended to be used to lift heavy weights: for this purpose divers use a lifting bag.

Emergency Wreck Buoy

An emergency wreck buoy is used to warn of a new wreck which has not yet been listed in maritime documents. The buoy is expected to be deployed for the first 24–72 h after the wreck occurs. After that time, more permanent buoyage (such as isolated danger marks or cardinal marks) should be deployed and charts updated.

The buoy is designed to provide to “provide a clear and unambiguous” mark of a new and uncharted danger. The buoy is painted with 4, 6, or 8 vertical stripes of alternate yellow and blue. In addition it may have the word “WRECK” painted on it. Optionally it may carry a vertical (St. George’s) cross painted yellow. The light flashes alternate yellow and blue for 1 s each with a half second gap between. No other navigation mark uses blue. Figure 6 shows the schematic diagram of an emergency wreck buoy (Website 2018g).

The IALA (International association of maritime aids to navigation and lighthouse authorities) defined the buoy in response to the sinking of the MV Tricolor and the subsequent collisions with the wreck by the Dutch vessel Nicola and Turkish-registered fuel carrier Vicky.

Weather Buoys

Weather buoys are instruments which collect weather and ocean data within the world's oceans, as well as aid during emergency response to chemical spills, legal proceedings, and engineering design. Moored buoys have been in use since 1951, while drifting buoys have been used since 1979. Moored buoys are connected with the ocean bottom using either chains, nylon, or buoyant polypropylene. With the decline of the weather ship, they have taken a more primary role in measuring conditions over the open seas since the 1970s. During the 1980s and 1990s, a network of buoys in the central and eastern tropical Pacific Ocean helped study the El Niño-Southern Oscillation. Moored weather buoys range from 1.5 to 12 m (5–40 ft) in diameter, while drifting buoys are smaller, with diameters of 30–40 cm (12–16 in). Drifting buoys are the dominant form of weather buoy in sheer number, with 1250 located worldwide. Wind data from buoys has smaller error than that from ships. There are differences in the values of sea surface temperature measurements between the two platforms as well, relating to the depth of the measurement and whether or not the water is heated by the ship which measures the quantity.

Weather buoys, like other types of weather stations, measure parameters such as air temperature above the ocean surface, wind speed (steady and gusting), barometric pressure, and wind direction. Since they lie in oceans and lakes, they also measure water temperature, wave height, and dominant wave period. Raw data is processed and can be logged on board the buoy and then transmitted via radio, cellular, or satellite communications to meteorological centers for use in weather forecasting and climate study. Both moored buoys and drifting buoys (drifting in the open ocean currents) are used. Fixed buoys measure the water temperature at a depth of 3 m

(9.8 ft). Many different drifting buoys exist around the world that vary in design and the location of reliable temperature sensors varies. These measurements are beamed to satellites for automated and immediate data distribution. Other than their use as a source of meteorological data, their data is used within research programs, emergency response to chemical spills, legal proceedings, and engineering design. Moored weather buoys can also act as a navigational aid, like other types of buoys.

Weather buoys range in diameter from 1.5 to 12 m (5–40 ft). Those that are placed in shallow waters are smaller in size and moored using only chains, while those in deeper waters use a combination of chains, nylon, and buoyant polypropylene. Since they do not have direct navigational significance, moored weather buoys are classed as special marks under the IALA scheme, are colored yellow, and display a yellow flashing light at night.

Discus buoys are round and moored in deep ocean locations, with a diameter of 10–12 m (33–39 ft). The aluminum 3-m (10 ft) buoy is a very rugged meteorological ocean platform that has long term survivability. The expected service life of the 3-m (10 ft) platform is in excess of 20 years and properly maintained, and these buoys have not been retired due to corrosion. The NOMAD is a unique moored aluminum environmental monitoring buoy designed for deployments in extreme conditions near the coast and across the Great Lakes. NOMADs moored off the Atlantic Canadian coast commonly experience winter storms with maximum wave heights approaching 20 m (66 ft) into the Gulf of Maine.

Drifting buoys are smaller than their moored counterparts, measuring 30–40 cm (12–16 in) in diameter. They are made of plastic or fiberglass and tend to be either bi-colored, with white on one half and another color on the other half of the float, or solidly black or blue. It measures a smaller subset of meteorological variables when compared to its moored counterpart, with a barometer measuring pressure in a tube on its top. They have a thermistor (metallic thermometer) on its base, and an underwater drogue, or sea anchor, located 15 m (49 ft) below the ocean surface connected



Surface Buoy, Fig. 7 Weather buoy operated by the National Data Buoy Center (Website 2018h)

with the buoy by a long, thin tether. A weather buoy operated by the National Data Buoy Center is shown in Fig. 7 (Website 2018h).

Spar Buoy

A spar buoy is a tall, thin buoy that floats upright in the water and is characterized by a small water plane area and a large mass. Because they tend to be stable ocean platforms, spar buoys are popular for making oceanographic measurements. Adjustment of the water plane area and the mass allows spar buoys to be tuned so they tend to not respond to wave forcing. This characteristic differentiates them from large water plane area buoys such as discus buoys that tend to be wave followers. Spar buoys are often used as stable platforms for wave measurement devices and air–sea interaction measurements. Spar buoys range in length from a few feet to the 354-foot (108 meter) RV FLIP. To avoid the difficulties inherent with shipboard launch and recovery, helicopter deployment of large spar buoys has been studied. Figure 8 is a diagram of an anchored spar buoy (Website 2018i).

Surface Buoy, Fig. 8 A diagram of an anchored spar buoy (Website 2018i)



Self-Locating Datum Marker Buoy (SLDMB)

A self-locating datum marker buoy (SLDMB) is a drifting surface buoy designed to measure surface ocean currents. The design is based on those of the Coastal Ocean Dynamics Experiment (CODE) and Davis-style oceanographic surface drifters – National Science Foundation (NSF) funded experiments exploring ocean surface currents. The SLDMB was designed for deployment by United States Coast Guard (USCG) vessels in search and rescue (SAR) missions and is equipped with a Global Positioning Satellite (GPS) sensor that, upon deployment in fresh- or saltwater, transmits its location periodically to the USCG to aid in SAR missions. Additionally, SLDMB are deployed in oceanographic research in order to

study surface currents of the ocean. This design has also been utilized by Nomis Connectivity for secure ocean-based communications.

The SLDMB is based on the Davis-style drifter design, which attempts to minimize the effects of wind and surface waves. This is accomplished by reducing the area above the ocean surface to small floats and an antenna. Below the surface is a series of drogue vanes to 70 cm (less than the 1 m Davis-style buoy) in depth, which catch the ocean current, along with electronic equipment that deploys the vanes and antenna, receives the GPS signal, and transmits the location to the USCG.

SLDMB construction varies by manufacturer, but those used by the USCG consists of four orthogonal drag vanes 0.5 m wide and 0.7 m high of nylon fabric. These are supported by PVC arms at top and bottom, which extend from a cylindrical hull that contains the electronic equipment. Small floats are attached to the end of each upper arm in order to maintain buoyancy and a small antenna projects above the SLDMB.

Deployment of the SLDMB may be accomplished by aircraft (both fixed-wing and rotary) or by ship. SLDMB deployed by aircraft are encased in a tube and attached to a parachute which decreases the impact produced upon hitting the water but without so much drag that the buoy can drift off-course according to USCG SAR guidelines. Upon reaching the ocean surface, the outer casing and parachute break away from the SLDMB, and the spring-loaded antenna deploys. Electronics consist of a GPS receiver, electronic transmitter, and sufficient batteries to provide continuous data collection for a period of 2 weeks to 1 month.

Because it has a small above-water surface and high underwater surface area, the effect of surface winds and waves has a negligible effect, instead moving with the flow of the upper 1 m of the water column. The USCG has found that this instrument behaves as a “zero-leeway” object, moving with the top meter of the water column, with no additional motion due to the direct effect of the wind on the SLDMB’s exposed areas. The USCG maintains several hundred SLDMBs for deployment and responds to more than 5,000 SAR cases each year. In the year 2006, more than 400 SLDMBs

were deployed in SAR applications, with an average lifetime of 22 days. The USCG may release SLDMBs at their discretion to aid in search efforts. In remote areas, SLDMBs are deployed via C-130 aircraft or helicopters. The GPS unit on each SLDMB calculates its position every 30 min and transmits the data via the ARGOS data collection system to USCG Operational Support Center (OSC). During high traffic periods, the USCG may predeploy units in order to have existing data in areas where SAR operations are more likely, reducing the time required to collect ocean current data during the SAR process. SLDMB may be released as single units or as a group, depending on the situation required. In cases where the last known position is known and the time lag to SLDMB deployment are minimal, only a single unit may be necessary. However, if a sufficient time lag exists, or the last known position is not available, multiple SLDMBs should be used. An example of this second case would be a downed fishing vessel, in which only the approximate area of the vessel is known (Website 2018j).

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Surface Mineral Storage and Dump

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Synonyms

MSS – mineral storage system; MSTS – mineral storage and transport system; MTS – mineral transport system

Introduction

With the depletion of land mineral resources, the seabed mineral will become the most important resources. In order to exploit seabed resources, the world is vigorously developing deep-diving technology and deep-sea mining system (Bath 1989; Lodge and Verlaan 2018). The MSTS is the most important sea surface support system in deep-sea mining system and is an important guarantee for the seabed mineral storage and transporting to the barge (Lin et al. 2019). The MSTS has the characteristics of multiple equipment, large space, large weight and complex structure layout. The storage and transport efficiency of the system is an important indicator that directly determines whether the deep-sea mining system can continuously exploit seabed mineral (Yang et al. 2019).

The MSTS consists of two subsystems: MSS and MTS. The function of this system is to put the ores collected from the seabed into the cabin and then transfer it to the barge.

Mineral Storage and Transport System

Function

With flexible storage and transport ability, the system can properly stack and efficiently transport the pretreated ores to minimize the impact of the transport process on the stability of the mining vessel.

The storage and transport efficiency of the system should match the cutting efficiency of the seafloor mining system to guarantee the continuous operation and transport of seafloor, sea surface, and land.

The system can adapt the impact of the movement of the mining vessel itself.

Component

The MSTS consists of two subsystems: MSS and MTS. The MSS includes the cabin subdivision system and stacking system. And the MTS includes the reclaiming system, the hoisting system, and the transport system (Fig. 1).

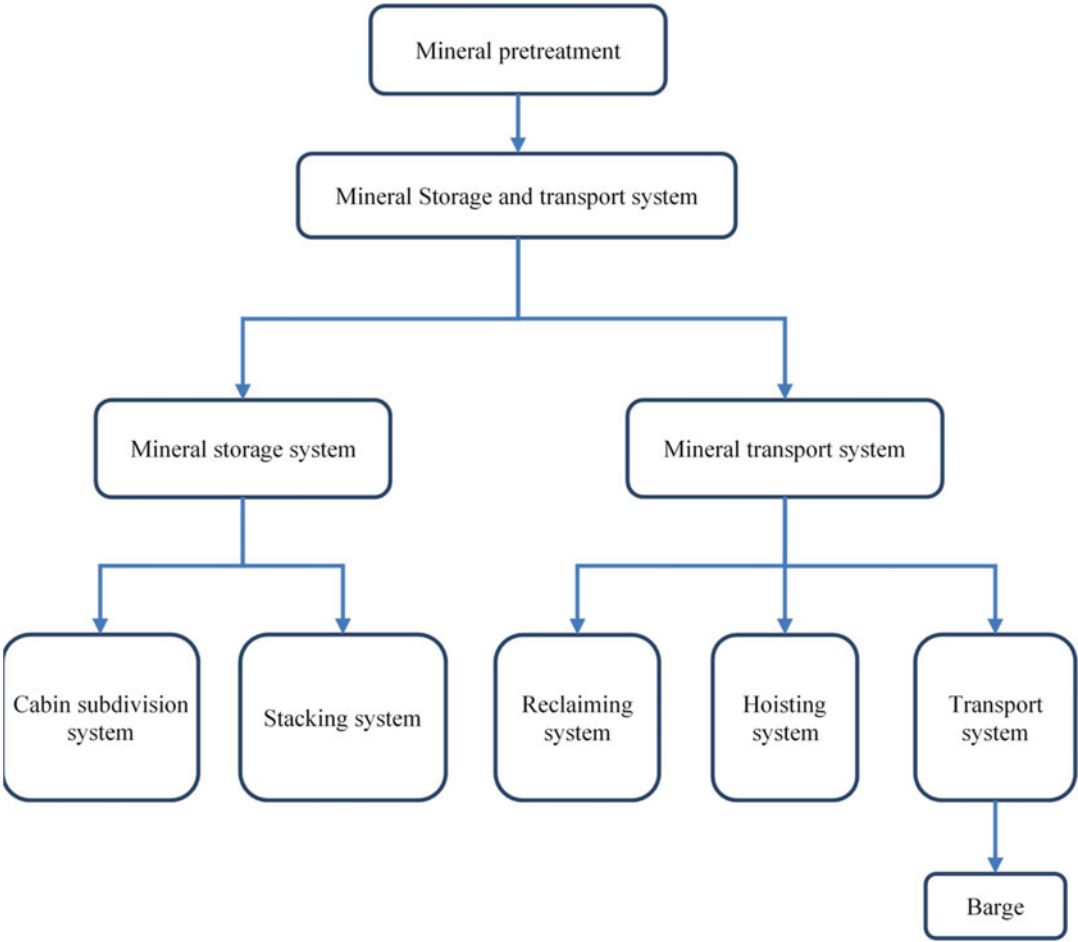
Mineral Storage System

The cabin subdivision system transports the pretreated ores into designed cabins. Reversible belt conveyor, belt conveyor, and en masse conveyor can be used (Wang 2001).

After the ores are loaded in the cabin, the stacking plane needs to be relatively flat. Otherwise, it will cause the waste of the storage capacity around the four corners of the cabin and the imbalance of the vessel's center of gravity, which will affect the operation of the mining vessel and the safety of navigation. The stacking system mainly guarantees uniform mineral arrangement and reduces empty cabin rate. The system mainly includes throwing machine, real-time stacking monitoring system, and corresponding electronic control system.

Mineral Transport System

The reclaiming system collects ores scattered throughout the cabin and provides them to the hoisting system, which can use the double-row bucket conveyors, scraper conveyor, bucket elevator, and so on (China Heavy Machinery Industry Association 2015).



Surface Mineral Storage and Dump, Fig. 1 Components of the MSTs

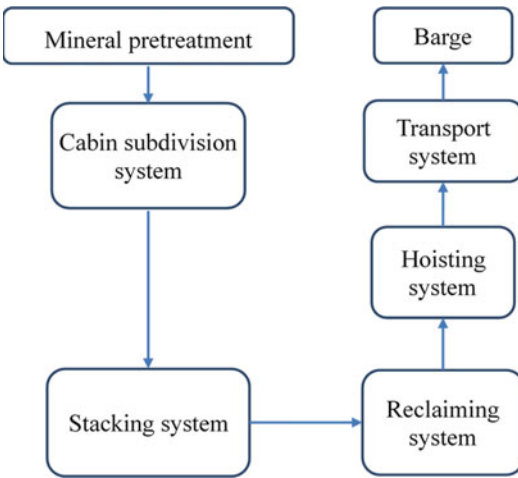
The hoisting system lifts the ores from the cabin to the height that can be transported to the side of the vessel. The commonly used lifting methods include belt conveyor and mechanical type. Belt conveyor mainly includes inclined belt conveyor, C belt conveyor, waving belt conveyor, and so on. Mechanical hoisting type mainly includes bucket elevator, screw conveyor, bucket wheel machine, grab crane, and so on.

The transport system transports mineral that is hoisted to a certain height to the barge. The common equipment includes the telescopic loading machine, shuttle conveyor, and so on. The telescopic loading machine is composed of the boom belt conveyor, transition belt conveyor, telescopic barrel, pitching device, rotary device, and etc. Shuttle belt conveyor

developed on the basis of DTII(A) belt conveyor is a kind of mobile belt conveyor with running wheels.

Basic Procedure

The ores collected on the seabed is transported to the sea surface by the seafloor mining system and then processed by the mineral pretreatment system. The cabin subdivision systems, according to ship loading requirements, deliver ores to different cabins. The ores are distributed reasonably and evenly in the cabins with the stacking system. When the cargo hold is almost filled with ores, the reclaiming system collect the ores and lift them to the deck level through the hoisting system. Finally, with the help of the transport system, these ores are transported to the barge. The MSTs



Surface Mineral Storage and Dump, Fig. 2 Basic procedure of the MST

provides efficient storage and transports to ensure continuous operation of the deep-sea mining system as well as continuous transport of seabed mineral to the land (Fig. 2).

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Surface Plasmon Resonance (SPR)

► [Fiber Optic Hydrophone](#)

Surface Plasmon Wave (SPW)

► [Fiber Optic Hydrophone](#)

Surface Vessels

► [Surface Warships](#)

Surface Warships

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Synonyms

[Military ships](#); [Surface vessels](#); [Warships](#)

Definition

The surface warships are a kind of naval vessels, which are usually equipped with a certain number of soldiers, weapons, and special equipment and mainly sail on the surface of water with tactical and technical capabilities required by warfare or service support missions.

Scientific Fundamentals

Historic Evolution of Surface Warships

From ancient times, ships have been extremely important transportation vehicles and military weapons. During the Stone Age, ancient humans waded through canoes and rafts to carry out fishing and hunting. With the development of technology, the ancient humans created sails and oars to control navigation of ships, and wooden sailing warships began to dominate the sea from then on. Since the modern industrial revolution,

especially after the application of welding technology and steel materials, the performance of ships has been greatly enhanced (Marshall 1995; Gibbons 2001).

In the long period of feudalism, wooden sailing warships had been dominant control by naval forces. Since the mid-fourteenth century, sailing warships had gradually been equipped with artillery on both sides; by the seventeenth century, the tactics of ship-to-ship gunnery had developed into the main tactics of naval warfare; during the eighteenth century, the naval fleets gunnery tactic had further developed into the stable and well-known “line-of-battle” tactic (Konstam and Bryan 2001). At that time, the warships with large tonnage and heavy artillery were directly responsible for the combat mission of the “line-of-battle” tactics, which was later known as battleships.

At the beginning of the nineteenth century, steam engine technology and metal smelting technology made great progress, which resulted in a greater breakthrough of modern shipbuilding industry, thus contributing to the profound significance of the warships.

In the Battle of Sinop during the Ceimean in 1853, General Nakhimov of Russia Navy commanded the Black Sea Fleet performing ingenious tactic and newly assembled blasting cannons to annihilate the sailing warships of Turkish Navy. In the later period of the naval warfare, the advanced steam-powered warships of both Britain and France perfectly demonstrated the advantages of maneuverability, and finally sailing warships lost the dominant position of naval warfare (Badem 2010). The Crimean War ended in 1856, and formally established the important dominance of steam-powered warships in naval fleets and maritime warfare, and prompted the development of warships toward the direction of armor.

At the early twentieth century, the British “HMS Dreadnought” was launched into service. The “HMS Dreadnought,” completely equipped with large-caliber artillery, was the first warship to be driven by a steam turbine, and the world’s most advanced and largest battleship at that time (Fairbanks 1991). The armored “Dreadnought” had dominated the world navies for 40 years, marking the arrival era of brilliant “Big-ship,

Big-gun” (Robert 1992). As a result, arms races spread around the world. In 1916, the Battle of Jutland made the glory of “Dreadnought” reach its peak; the world navies began to manufacture giant battleships equipped with large-caliber artillery (Black 2016).

The main types of surface warships at different times are shown in Fig. 1. After World War II, with the rapid development of modern science and technology, the industrial automation of shipbuilding has been increasingly improved, as well as the quality and manufacturing efficiency. Moreover, with the carrier-based weapons, power plants, shipbuilding materials, electronic equipment, and shipbuilding processes booming, the development of surface warships steps into modernization.

Main Types and Characterizations of Modern Surface Warships

Surface warships are the oldest type of naval vessels, so it can be said that without surface warships, there would be no naval fleets. At present, according to different missions undertaken by surface warships, they can be generally classified into combat warships and auxiliary warships.

The Battleship

The battleship is a kind of large-scale surface combat ship with large-caliber artillery as main weapon and heavy armored protection, and also capable of carrying out ocean-going combat missions. Battleships used to be the capital ships of navy fleets, and are generally divided into single column as “line-of-battle” tactic for artillery battles (Lavery 1983). The battleship was also known as Ironclad or Panzership at early times.

From the late nineteenth century to the early twentieth century, the battleship was considered to be the most powerful warships, and have always been one of the capital ships in major maritime powers. The most prominent feature of battleship is a great number of large caliber guns, and the maximum diameter of a single gun can reach 460 mm. What’s more, the maximum thickness of armors is at least the same as main gun caliber, and the displacement can reach 50,000~60,000 tons, with once endurance ranging from 10,000 to 15,000 miles. After World War II,



▲ Santisima Trinidad. The most spectacular battleship of the entire sail era—the only four-story artillery deck battleship during the 18th century.

▼ KM Bismarck Battleship. Served in Nazi Germany in 1940, one of the largest and most technologically advanced battleships during World War II.



▼ USS Long Beach. The world's first nuclear-powered guided missile cruiser, served in the United States Navy in 1961.



► HMS Dreadnought. Served in the British Royal Navy in 1906, and was the first main warship powered by steam turbines during the history of modern navy.



▼ USS Enterprise. The world's first nuclear-powered aircraft carrier, served in the United States Navy in 1961.



◀ DING YUAN Ironclad. Served in the BeiYang Fleet of Qing Dynasty in 1885, and was the largest warship in the Far East during the time.

▼ Horizon-class Destroyer. A class of air-defence destroyers in service with the French Navy and the Italian Navy in 2008.



◀ Type 23 Frigate. Served in 1990, as a large ocean-going multipurpose frigate under the British Royal Navy.

Surface Warships, Fig. 1 Main types of surface warships at different times (<https://en.wikipedia.org/>)

the strategic dominance of battleship was gradually replaced by aircraft carrier and ballistic missile submarine.

As shown in Fig. 2, the US “Iowa-class” battleship was latest decommissioned, and also the largest tonnage built by the US navy during World War II. There are a total of four “Iowa-class” battleships built since 1940, namely “Iowa”(BB-64), “New Jersey”(BB-62), “Missouri”(BB-63), and “Wisconsin”(BB-64). The hull length of “Iowa-class” is 270.4 m with 33 m in width, 11.6 m in draught, and a standard displacement of 45,000 tons. The whole ship was equipped with armor protection, as the thickness

of common parts is 150 mm and important parts can reach 400 mm, which was the thickest armored surface warship in the world after the war (Burr and Bull 2010). During World War II, the “Iowa-class” battleships successively participated in the Battle of Marshall Islands, the Battle of Marianas, the Battle of Okinawa, etc.

The Cruiser

The cruiser is a kind of large-scale surface combat ship with a displacement of more than 7,000 tons, equipped with guided missiles, naval guns, and carrier-based helicopters. Cruisers are a kind of relatively old navy warships, which



Surface Warships, Fig. 2 Iowa fires during a target exercise near Vieques Island (<http://ww2db.com/>)



Kirov-class Battlecruiser "Pyotr Velikiy"



Ticonderoga-class Cruiser "Normandy"

Surface Warships, Fig. 3 Typical cruiser in active service (<https://en.wikipedia.org/>)

almost were born and developed at the same time as battleships. Cruiser has the characteristics of high speed, great endurance, and better sea-keeping, which can make a contribution to carrying out ocean-going combat missions a long time in atrocious weather. The main weapons of the early cruisers were the artillery, once known as artillery cruiser, but then replaced by missile cruiser. In the twenty-first century, cruisers are basically no longer built, which may gradually be replaced by destroyers.

At present (as of April 2018), there still are 15 "Ticonderoga-class" cruisers in active

service of the US Navy, all of which are conventional-power missile cruisers; Russia's active cruisers include two nuclear-powered "Kirov-class" battlecruiser, three conventional-powered "Slava-class" cruisers. Fig. 3 shows the active "Peter the Great" nuclear cruiser and the "Normandy"-guided missile cruiser.

The "Kirov-class" battlecruiser was developed by the former Soviet Union in the early 1980s. A total of four "Kirov-class" cruisers were built, and there are still two cruisers serving in the Russian Northern Fleet presently. The "Kirov" (original name) is the first battlecruiser of this

class with a plump shape, and adopting with an outboard bow, a large square stern, and flight deck. The “Kirov” is 250.1 m in length, 28.5 m in width, with a maximum speed of 31 knots, full displacement of 28,000 tons, and a crew of 900. The “Kirov” began to take service in July 1980 that was the largest cruiser built after World War II (Pair 2011).

The Destroyer

The destroyer is a kind of medium-scale surface combat ship mainly equipped with guide missiles, naval guns, and anti-submarine weapons with multiple battle capabilities, which has been a kind of important naval warships since the nineteenth century. Modern destroyers can be mainly classified into the anti-aircraft destroyers and the anti-submarine destroyers.

The displacement of anti-aircraft destroyers is generally between 6,000 and 10,000 tons, with the main missions of fleet air defense (FAD); likewise there is a trend of further large-scale development, such as the US Navy latest “Zumwalt-class” with full-load displacement reaching 15,000 tons. The anti-aircraft destroyers are mainly equipped with a set of phased array radars, long-range ship-to-air guided missiles, and high-performance automated command system, as well as naval guns, short-range defense systems, anti-submarine deep-water bombs, and torpedoes. There are typical anti-aircraft destroyers such as the US “Arleigh Burke-class” destroyer, the “Type 055” destroyer of China, and the European “Horizon-class” destroyer.

The displacement of anti-submarine destroyers is generally between 3,000 to 6,000 tons, with the main combat missions of anti-submarine and anti-ship warfare. The anti-submarine destroyers are mainly equipped with sophisticated sonar anti-submarine system and anti-ship weapons, at the same time, by means of short-range missiles to assist anti-aircraft operations. There are typical anti-submarine destroyers such as the Russian “Udaloy-class” destroyer, the US “Spruance-class” destroyer, and Japanese “Hatakaze-class” destroyer.

The “Type 055” destroyer is a class of guided missile destroyer being constructed for Chinese

People’s Liberation Army Navy, and also is a multi-mission design aimed to undertake expeditionary missions and form the primary escort for the aircraft carriers of Chinese Navy. The “Type 055” adopts a conventional flared hull with distinctive stealthy features including an enclosed bulbous bow which can hide mooring points, anchor chains and other equipment, in addition, the bow and main deckhouse are configured similarly to previous “Type 052D”. The full-load displacement of “Type 055” is about 12~13,000 tons, with a length of 180 m and maximum service speed reaching 30 knots. Fig. 4 demonstrates the sea trial of the first “Type 055” destroyer. The first ship of “Type 055” launched on 28 June 2017, which indicated that “Type 055” was the largest post-Second World War warships in East Asia.

The Frigate

The frigate is a kind of light-scale surface combat ship that is mainly equipped with guided missiles, naval guns, deepwater bombs, and anti-submarine torpedoes. The frigates originated from a three-mast armed sailing ship in sixteenth century, and are a kind of traditional naval warships, which are also a type of surface warships with the largest number, the most extensive distribution, and the most opportunities for warfare in major coastal countries. Frigates are mainly applied for anti-submarine, sea escort, reconnaissance, mine-laying, and support landing operations.



Surface Warships, Fig. 4 Sea trial of the first “Type 055” destroyer (<http://www.guancha.cn/>).

There are more than 50 countries and regions owning frigates all over the world. According to the difference of displacement and operational environment, the frigate can be classified into offshore frigate and ocean-going frigate. The offshore frigates typically include the South Korean “Pohang-class” corvette and the Chinese “type-056” frigate. The ocean-going frigates typically include the US “Oliver Hazard Perry-class” frigate, the British “Duke-class” frigate, and the Russian “Neustrashimyy-class” frigate.

The “Oliver Hazard Perry-class” frigate is a kind of ocean-going frigates with moderate performance of the US Navy. The “Perry-class” frigates have a variety of tactical roles and can undertake various missions which are one of the most advanced missile frigates. The “Perry-class” frigate is 135.6 m in length, 13.7 m in width, 6.7 m deep in draft, and a full-load displacement of 3,600 tons, with a speed of 30 knots. In addition, the “Perry-class” frigates also are exported to nearly 10 countries and regions in the world (Zhang 2015). Fig. 5 shows that the “Ford” of “Perry-class” was performing a cruise mission. In September 2015, the last frigate of the “Perry-class” “Kauffman” (FFG-59) was decommissioned, marking the complete decommission of the “Perry-class”.

The Aircraft Carrier

The aircraft carrier is a kind of large-scale surface warship with carrier-based aircrafts as main weapons, which is usually equipped with a huge full-length flight deck and island located on the starboard side. The aircraft carrier is the capital ship of a navy fleet, mainly responsible for air supports and long-range strikes, and other warships of the fleet are responsible for providing protection and supply. So far, the aircraft carrier has been one of the largest, most complex and powerful weapons in the world. Moreover, the aircraft carrier has become indispensable for the modern navy, and also been a symbol of comprehensive national strength.

Modern aircraft carriers are usually classified into large-scale aircraft carriers (over 60,000 tons), medium-scale aircraft carriers (about 30,000 to 60,000 tons), and light-scale aircraft carriers (below 30,000 tons), according to the bulk of full-load displacement. In addition, according to the different type of power-plants, aircraft carriers can also be classified into nuclear powered aircraft carriers and conventional power aircraft carriers. At present, only eight countries possess aircraft carriers, but only the US “Nimitz-class,” “Ford-class,” and French “De Gaulle-class” are currently equipped with nuclear-power, and the rest are conventional-power aircraft carriers.

Surface Warships,
Fig. 5 Oliver Hazard
Perry-class frigate
“Ford” (<http://www.navsource.org/>)



Surface Warships,

Fig. 6 Type 001 Aircraft Carrier “Liaoning” (<http://www.haijun360.com/>)



The Chinese “type 001” aircraft carrier “Liaoning” is the first aircraft carrier of the PLA navy, whose predecessor was the Soviet navy “Kuznetsov-class” aircraft carrier “Varyag.” In 2005, the uncompleted aircraft carrier was modified and reconstructed for scientific research, experiment, and training. In September 2012, the “Varyag” was renamed as “Liaoning,” and was delivered to the PLA navy. Fig. 6 demonstrates the sea trials of “Liaoning.” At present, the “Type 001A” aircraft carrier designed and constructed autonomously by China was launched on April 26, 2017.

The Landing Craft

The surface landing craft is a kind of warship that mainly transports landing soldiers, replenishments, military weapons, and performs landing frontages. Landing frontages are always the most difficult in naval warfare, and the landing forces would receive fire supports from the air and sea, in order to ensure that landing crafts can safely transport soldiers and equipment to the coast. Landing crafts can be classified into amphibious warfare ship (such as LPH/LHA/LHD), dock landing ship (LSD), amphibious cargo ship (such as LKA), and light landing craft (such as LCU/LCM/LCAC), amphibious command ship (such as LCC/AGF), depending

on different missions undertaken at warfare. So far, most countries have strengthened the development of landing crafts, and many landing crafts have both plane landing and vertical landing capabilities.

The amphibious assault ship is a kind of landing craft that can provide air and surface supports behind the battlefield when conducting amphibious operations in hostile coastal areas. This type of landing craft was developed from Landing Helicopter Dock, but also combines with the well deck design, which can be used to load landing boats, transport tanks, and provides take-off and landing environment for carrier-based aircraft (Friedman 2002). The US active “Wasp-class” amphibious assault ship is currently the largest multifunctional amphibious assault ship in the world, with a full-load displacement of 41,150 tons. Fig. 7 shows the “Wasp-class” amphibious assault ship “Makin Island.” The “Wasp-class” can carry almost a whole US Marine Corps unit and campaign at the battlefield of hostile territory through helicopters and other landing crafts.

The Hospital Ship

The hospital ship is a kind of special auxiliary warship that combines rescue and wounded

Surface Warships,

Fig. 7 Wasp-class Amphibious Assault Ship “Makin Island” (<http://www.navsource.org/>)

**Surface Warships,**

Fig. 8 Peace Ark Hospital Ship “Daishan Dao” (<http://www.haijun360.com/>)



evacuation on the ocean, and is equipped with medical treatment staff, departments and medical equipment, a large number of relatively advanced rescue and evacuation facilities, known as the “marine hospital.” Large-scale hospital ships are one of the important symbols of the modern navy, and there are a few countries in the world, such as the United States, Britain, Canada, and China, which possess hospital ships with capacity to provide medical assistance away from the land.

“Peace Ark” is a professional large-scale hospital ship designed for maritime medical treatment in China. The original name is “Daishan Dao,” with a full-load displacement of 14,300 tons, which is equipped with nearly 2400

sets of advanced medical equipment and more than 500 beds. Fig. 8 shows the hospital ship “Peace Ark.” The “Peace Ark” is the world first large-scale professional hospital ship of 10,000 tons, which formally commissioned in the East Sea Fleet at the end of 2008, and has carried out humanitarian medical treatment many times and “Rim Pacific” military exercises.

The Replenishment Ship

Replenishment ships are a kind of auxiliary warships that are used to provide military supplies such as fuel, ammunition, food, and attachments for aircraft carrier fleets, or warships. The replenishment ships were widely used at the Pacific



Surface Warships, Fig. 9 Sacramento-class fast combat support ship “Camden” (<http://www.globalsecurity.org/>)

battlefield during World War II. The modern replenishment ships are equipped with advanced remote-maintenance system, which are main logistics supports for military activities in the ocean.

The US “Sacramento-class” fast combat support ship was the first class of integrated replenishment ships in the world, and the first ship “Sacramento” (AOE-1) was built in the 1960s. So far, she has been still the largest and fast integrated replenishment ship of the world. Fig. 9 shows the “Sacramento-class” fast combat support ship “Camden.” The main missions of “-Sacramento-class” are to accompany the aircraft carrier fleets to provide supplies. The “Sacramento-class” can carry 177,000 barrels of fuel, 2,150 tons of ammunition, and 7,500 tons of dry-cold cargo. What’s more, she also has 15 refueling stations, and is usually equipped with three “UH-46” helicopters for vertical replenishment, enabling the fleet sail away from the base for a long time to maintain battle activities. Up to now, the “Sacramento-class” fast combat support ships have all decommissioned.

Key Applications

Advanced Large-Scale Aircraft Carriers of Maritime Power

As the most powerful maritime military power in the world, the US Navy has as many as 11 aircraft carriers in active service (10 for the “Nimitz-

class” and 1 for the “Ford-class”), and all belong to large-scale aircraft carriers with full-load displacement over 100,000 tons. The “Nimitz-class” aircraft carriers are current main forces of the ocean-going navy fleets, and are also the second-generation of US nuclear-powered aircraft carriers succeeded from the “Enterprise” aircraft carrier. The “Nimitz-class” aircraft carrier CVN-68 is shown in Fig. 10.

The “Ford-class” aircraft carrier started to be built in August 2005, and took active service in July 2017. The first sea trial of “Ford-class” CVN78 in 2017 is shown in Fig. 10. The “Ford-class” is the current largest nuclear-powered aircraft carrier of the US Navy, with a maximum speed of over 30 knots and also the first carrier to adopt electromagnetic launching system.

Lost Battleship During World War II

The Japanese “Yamato-class” battleship used to be the largest one in the world, with a standard displacement of 64,000 tons, and a full-load displacement of nearly 73,000 tons, more than 20 large-caliber main and auxiliary guns, and a speed of 27 knots (Taylor 2017). The first ship “Yamato” began serving in the Imperial Japanese Navy in 1941 and was once known as “the world first battleship.” Fig. 11 shows the trial of the Yamato. Unfortunately, during the battle of Okinawa on April 7, 1945, Yamato was attacked by waves and waves of American carrier planes. She received serious damage from falling bombs



Nimitz-class "CVN-68"



Ford-class "CVN78"

Surface Warships, Fig. 10 Nimitz-class and Ford-class Aircraft Carrier (<http://www.maritimequest.com/>)



Surface Warships, Fig. 11 Yamato on trials in Sukumo Bay, October 30, 1941 (<http://battleshiplist.com/>)

within the first 15 min of the battle, then was struck by torpedoes on the port side. The largest battleship in the world finally sank, after an agonizing for 2 h.

The Active Destroyers of US Navy

The US Navy destroyers in active service include the "Arleigh Burke-class" and the "Zumwalt-class." The characteristics and configuration parameters of the two types of warships are shown in Fig. 12.

The "Arleigh Burke-class" is a kind of primary air defense guided missile destroyer, and also the world's first destroyer equipped with the "Aegis" system and completely adopting a stealth shaping design (Zhou et al. 2003).

The "Arleigh Burke-class" destroyer adopts a wide-short shaped type of single hull, with a V-shaped profile at the bow, a clear outboard above the waterline, and an inward slope on the side, which can greatly improve the seakeeping. In order to improve the vitality of the warships, the tactical operations centers are all in the main hull, and important cabins are fully equipped with "KEVLAR" armors, with full considerations given to the measurements to reduce damages and maintain combat effectiveness.

The "Zumwalt-class" destroyer is the new generation for multi-mission Aegis warships. This class of destroyers from the ship hull design, pipeline communication, network communication and motor power, weapon



Arleigh Burke-class "DDG-51"



Zumwalt-class "DDG- 1000"

Surface Warships, Fig. 12 US navy active destroyers (<http://www.navsourc.org/>)



Surface Warships, Fig. 13 Zubr air-cushioned landing craft (<http://mil.ru/>)

systems are beyond the cutting edge of modern technology (Xiong et al. 2016). The “Zumwalt” is the first destroyer of this class, and began to formally active service in October 2016. The hull of “Zumalt” is 182 m long which is 1.5 times longer than the “Arleigh Burke-class” destroyers, and a full-load displacement of 15,000 tons, ranked as the largest and most advanced destroyer of the US Navy.

Amphibious Warfare Pioneers of offshore Landing Crafts

The light landing craft is a kind of small-scale amphibious ship that is specifically developed for the transport of landing soldiers, weapons and landing supplies. The landing craft first appeared before the World War II and played an important role during the Normandy Landing

campaign. The European “Zubr” is the largest air-cushioned landing craft in the world, and is mainly used to perform offshore amphibious campaign in a confined sea area. Fig. 13 shows the “Zubr” air-cushioned landing craft, of which the full-load displacement is 555 tons, with a maximum speed of 63 knots, and an endurance of 300 miles at average speed of 55 knots.

Cross-References

- [Ship Design Process](#)
- [Ship-Fitting Design](#)
- [Ship Machinery Design](#)
- [Ship Overall Design](#)
- [Ship Structural Design](#)
- [Special Marine Vehicle](#)
- [Submarine](#)

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Surf-Riding and Broaching

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Synonyms

[Broaching](#); [Broaching stability](#)

Definition

Broaching is a violent uncontrollable turn that occurs despite maximum steering efforts to maintain the course. As with any other sharp turn event, broaching is accompanied with a large heel angle, which has the potential effect of a partial or total stability failure. Broaching is usually preceded by surf-riding which occurs when a wave, approaching from the stern, captures a ship

and accelerates the ship to the speed of the wave. Surf-riding is a single wave event in which the wave profile does not vary relative to the ship. Because most ships are directionally unstable in the surf-riding condition, this maneuvering yaw instability leads to an uncontrollable turn – termed “broaching” (IMO 2016). Broaching is considered as one of the most dangerous phenomena in following and stern-quartering waves for high-speed ships, such as destroyers and fishing vessels.

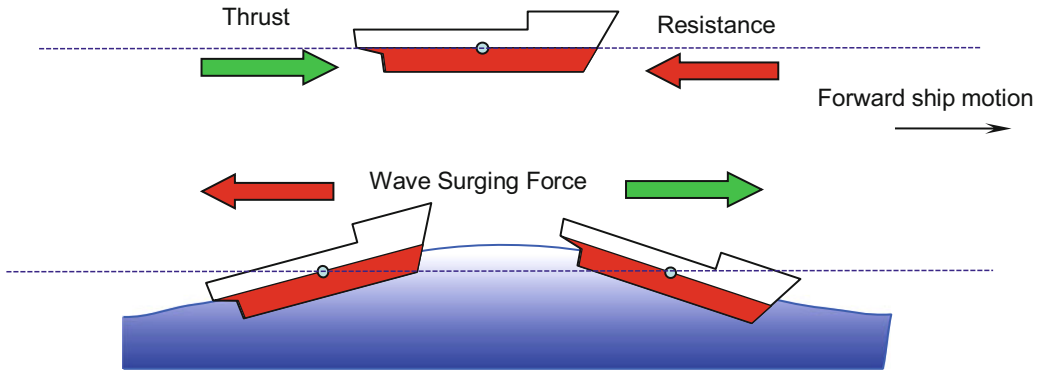
When a ship proceeds in following waves, three main forces act in the axial direction. Thrust is the force produced by the ship's propulsor to propel the ship forward. Resistance is the force that opposes the forward ship motion. The surging wave force is the force imparted by a wave to either push the ship forward or back depending on whether the ship is on the face or back of a wave, respectively. These forces are represented in Fig. 1 (IMO 2016).

When a surging wave force is present, three conditions are possible: surging motion, surf-riding under certain initial condition, and surf-riding under any initial condition.

Surging motion. This condition occurs when the wave surge force is insufficient to overcome the difference between the thrust of the ship's propulsor and the resistance of the ship at the wave celerity. In this case, the ship oscillates from increasing speed when on the front side of the wave to decreasing speed when on the back side of the wave (IMO 2016).

Surf-riding under certain initial condition (first threshold of surf-riding). This condition occurs is the ship speed for which the forward surge force of the wave at a particular point on the wave can exceed the difference between the thrust of the ship's propulsor and the resistance of the ship at the wave celerity. In this case, surf-riding could occur if the ship is accelerated with some external force under the self-propelled condition (IMO 2016).

Surf-riding under any initial condition (second threshold of surf-riding). This is the situation for which the ship kinetic energy is too large for the ship to be overtaken by a wave. Thus, the oscillatory surge motion cannot stably exist that surf-



Surf-Riding and Broaching, Fig. 1 Forces acting on a ship in following waves (Belenky et al. 2011; IMO 2016)

riding occurs regardless the initial ship position and forward speed (IMO 2016).

The resistance curve is approximated using the following polynomial expression with respect to ship speed u :

Scientific Fundamentals

The Generalized Expression for an Approximated Surge

The equation representing nonlinear surge motion is described as follows:

$$(m + m_x)\ddot{\xi}_G + [R(u) - T_e(u, n)] - X_w = 0 \quad (1)$$

where the overdot denotes differentiation with respect to time t . Here, R the ship resistance, T_e the propeller thrust, m the ship mass, m_x the added mass in x direction, u the ship forward velocity, and n the propeller revolution rate (RPM).

Assuming that the hull form is almost longitudinally symmetric, the Froude-Krylov force is represented as a first-order approximation and takes a form as follows:

$$X_w \approx -f \sin(k\xi_G) \quad (2)$$

$$f = \rho g k \frac{H}{2} \mu_x \sqrt{F_C^2 + F_S^2} \quad (3)$$

where the wavenumber k is defined as $2\pi/\lambda$, and λ is the wavelength. f is the exited wave force in the surge direction. H is the wave height. F_C and F_S is the real part and the image part of the Froude-Krylov. μ_x is the coefficient considering the effect of diffraction force.

$$R(u) = \sum_{i=0}^N r_i u^i = r_1 u + r_2 u^2 + \dots \quad (4)$$

where, $r_i (i = 0, 1, 2, \dots)$ is a polynomial fit of the resistance coefficient curves, and $r_0 = 0$.

The ship thrust is represented by the following quadratic expression:

$$T_e(u, n) = (1 - t_p) \rho n^2 D^4 K_T(J) \quad (5)$$

$$K_T(J) \approx \sum_{i=0}^N \kappa_i J^i = \kappa_0 + \kappa_1 J + \kappa_2 J^2 + \dots \quad (6)$$

$$J = \frac{u(1 - w_p)}{nD} \quad (7)$$

where κ_i is a polynomial fit of the thrust coefficient curves. In these equations, t_p and w_p are the thrust deduction and wake fraction, respectively. These are taken at their still water values. Here D and ρ are the propeller diameter and water density.

The effective propeller thrust in still water can then be expressed as:

$$T_e(u, n) = \sum_{i=0}^N \frac{(1 - t_p)(1 - w_p)^i \rho \kappa_i u^i}{n^{i-2} D^{i-4}} \quad (8)$$

Substituting Eqs. (2), (3), (4), (5), (6), (7), and (8) into Eq. (1), here the relative velocity $\dot{\xi}_G = u - c_w$, replacing u with $\dot{\xi}_G$ and c_w , the following equation is obtained:

$$\begin{aligned} (m + m_x)\ddot{\xi}_G + \sum_{i=1}^N \sum_{j=1}^i c_i \binom{i}{j} \dot{\xi}_G^j c_w^{i-j} \\ + f \sin(k\xi_G) \\ = T_e(c_w; n) - R(c_w) \end{aligned} \quad (9)$$

where the coefficients c_i are defined as

$$c_i \equiv -\frac{(1-t_p)(1-w_p)^i \rho k_i}{n^{i-2} D^{i-4}} + r_i; \quad (10)$$

Here c_w is the wave celerity.

Melnikov's Method for Obtaining the Generalized Expression for an Approximated Surge

Melnikov's method is one of the usual analytical approaches, and this technique is applied in this context referring (Maki et al. 2010; Maki et al. 2016).

Putting $y = k\xi_G$, and $\tau = t\sqrt{fk/(m+m_x)}$ in Eq. (9) yields the following equation:

$$\begin{aligned} \ddot{y} + \sum_{i=1}^N \sum_{j=1}^i C_{ij} \dot{y}^j + \sin y \\ = \frac{T_e(c_w; n) - R(c_w)}{f} \end{aligned} \quad (11)$$

$$C_{ij} \equiv \frac{c_i}{fk^j} \binom{i}{j} \frac{(fk)^{j/2}}{(m+m_x)^{j/2}} c_w^{i-j} \quad (12)$$

Putting ε which is $0 < \varepsilon \ll 1$ in Eq. (11) yields the following equation:

$$\ddot{y} + \sin y = \varepsilon \left(\frac{T_e(c_w; n) - R(c_w)}{f} - \sum_{i=1}^N \sum_{j=1}^i C_{ij} \dot{y}^j \right) \quad (13)$$

When $\varepsilon = 0$, the Hamiltonian part of Eq. (13) is:

$$\ddot{y} + \sin y = 0 \quad (14)$$

Putting $y = y$ and $z = \dot{y}$ in Eq. (14) yields the following equation:

$$\begin{cases} \dot{y} = z \\ \dot{z} = -\sin y \end{cases} \quad (15)$$

Equation (15) is also a Hamiltonian system, and the flowing equation can be used:

$$H(y, z) = \frac{z^2}{2} - \cos y \quad (16)$$

Putting $\dot{y} = 0$, $\dot{z} = 0$ in Eq. (15) yields the following equation:

$$\begin{cases} \dot{y} = z = 0 \\ \dot{z} = -\sin y = 0 \end{cases} \quad (17)$$

Three stable points (0,0), $(\pm\pi, 0)$ can be obtained by Eq. (17). $(\pm\pi, 0)$ are saddle points and (0,0) is the center.

When $H(\pm\pi, 0) = 1$, heteroclinic trajectory connecting the saddle points $(\pm\pi, 0)$ is as following:

$$H(\pm\pi, 0) = \frac{z^2}{2} - \cos y = 1 \quad (18)$$

$$z = \pm\sqrt{2(1 + \cos y)} = \pm 2 \cos(y/2) \quad (19)$$

where “+” is the upper part and “−” is the lower side of the heteroclinic trajectory.

The heteroclinic trajectory is shown in Fig. 2. The point is close to the saddle points A and B when $\tau \rightarrow +\infty$ and $\tau \rightarrow -\infty$.

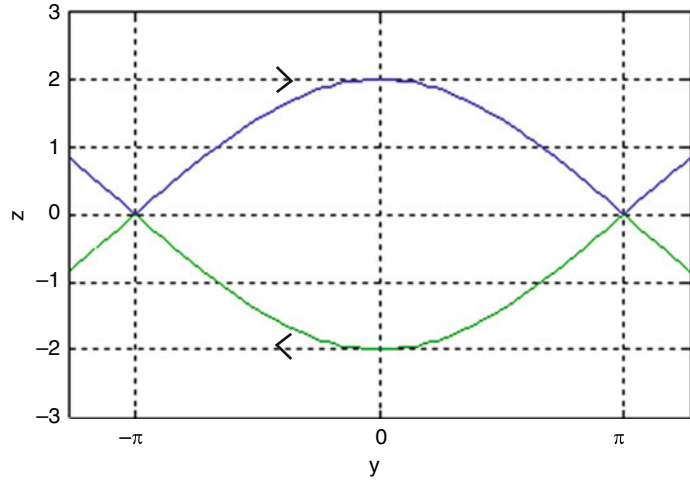
The heteroclinic trajectory is the boundary of surf-riding and stable motion, and the two saddle points is the balance points of surf-riding. There is heteroclinic trajectory for the balance point A to balance point B after a long time and is called heteroclinic bifurcation.

So the trajectory connecting the two saddle points $(\pm\pi, 0)$ on the lower side of vector field can be determined as follows:

$$z = \dot{y} = -2 \cos(y/2) \quad (20)$$

Here, the Melnikov function is defined as follows:

Surf-Riding and Broaching, Fig. 2 The heteroclinic trajectory



$$M = \int_{-\infty}^{+\infty} z \left(\frac{T_e(c_w; n) - R(c_w)}{f} - \sum_{i=1}^N \sum_{j=1}^i C_{ij} \dot{y}^j \right) d\tau \quad (21)$$

Substituting Eq. (20) into Eq. (21) and changing variables yield:

$$\begin{aligned} M &= \int_{-\infty}^{+\infty} \frac{T_e(c_w; n) - R(c_w)}{f} \frac{dy}{d\tau} d\tau \\ &- \sum_{i=1}^N \sum_{j=1}^i C_{ij} \int_{-\infty}^{+\infty} \dot{y}^j \frac{dy}{d\tau} d\tau \\ &= - \int_{-\pi}^{\pi} \frac{T_e(c_w; n) - R(c_w)}{f} dy + \sum_{i=1}^N \\ &\times \sum_{j=1}^i C_{ij} \int_{-\pi}^{\pi} \dot{y}^j dy \end{aligned} \quad (22)$$

Assuming that $M = 0$, the following relationship can be obtained:

$$\begin{aligned} 2\pi \frac{T_e(c_w; n) - R(c_w)}{f} \\ = \sum_{i=1}^N \sum_{j=1}^i C_{ij} (-2)^j I_j \end{aligned} \quad (23)$$

$$I_j = \int_{-\pi}^{\pi} \cos^j(y/2) dy \quad (24)$$

Equation (23) is the approximated condition for the surf-riding threshold. Critical revolution

Surf-Riding and Broaching, Table 1 Principal particulars of the ITTC ship A-2

Items	Ship
Length: L	34.5 m
Draft: T	2.65 m
Breadth: B	7.60 m
C_B	0.597

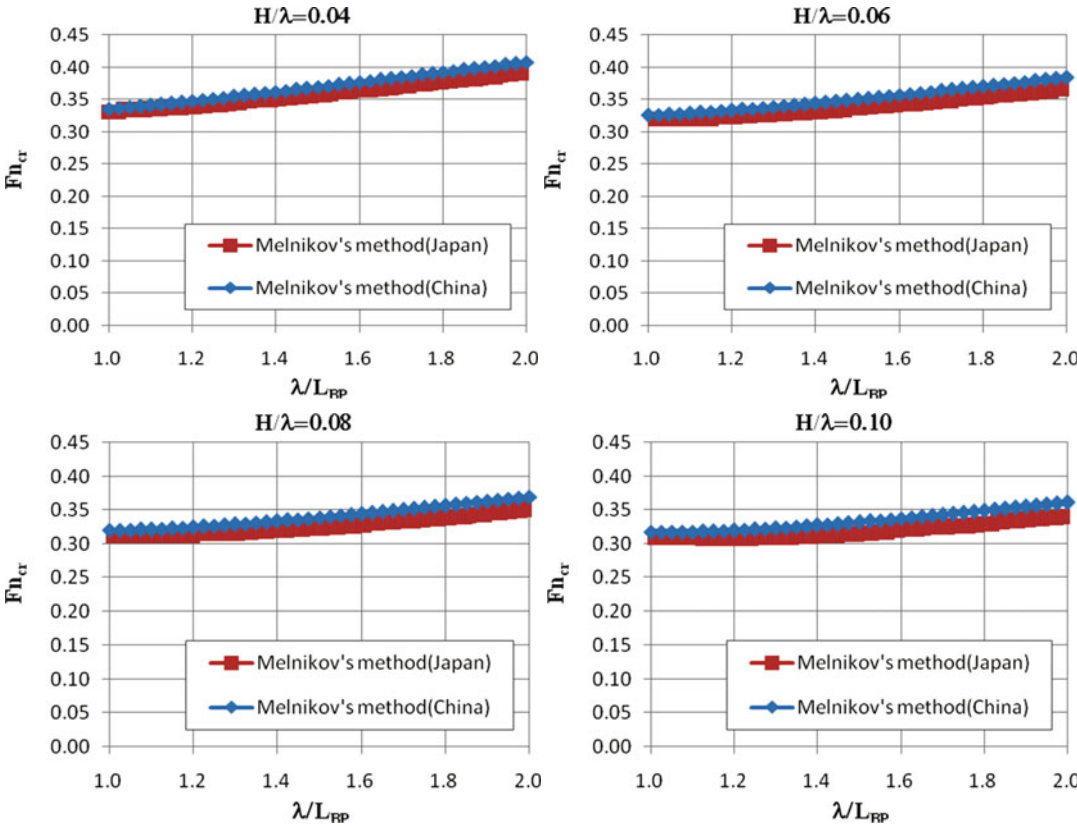
n_{cr} can be obtained by Eq. (22), and then critical ship forward speed u_{cr} and Fn_{cr} can be obtained by the following equation.

$$T_e(u_{cr}; n_{cr}) - R(u_{cr}) = 0 \quad (25)$$

Key Applications

The first example is the ITTC ship A-2 whose principal particulars are given in Table 1. The calculated results of the surf-riding threshold using Melnikov method are shown in Fig. 3.

The second example is about the ONR tumblehome vessel. The free running experiment of the ONR tumblehome vessel was conducted to recur the surf-riding and broaching phenomena in regular following and stern-quartering waves at the maneuvering and seakeeping basin of China Ship Scientific Research Center (CSSRC). The basin is 69 m length, 46 m breadth, and 4 m depth, which is equipped with flap wave makers at the two adjacent sides of the basin. The ship



Surf-Riding and Broaching, Fig. 3 The surf-riding threshold using Melnikov method

Surf-Riding and Broaching, Table 2 Principal particulars of the ONR tumblehome

Items	Ship	Model
Length: L	154.0 m	3.800 m
Breadth: B	18.8 m	0.463 m
Depth: D	14.5 m	0.358 m
Draft: d	5.494 m	0.136 m
Displ.: W	8507 ton	127.8 kg
C _B	0.535	0.535
GM	2.068 m	0.051 m
δ _{max}	35°	35°



Surf-Riding and Broaching, Fig. 4 A snapshot of surf-riding in the experiment

model was equipped with double propellers and double rudders. Ship motions were measured by the MEMS (Micro Electro-Mechanical System)-based gyroscope placed on the ship model.

The principal particulars of the ONR tumblehome vessel are shown in Table 2. The ship model in free running experiment is shown Figs. 4 and 5.

As shown in Figs. 4 and 5, the ship experiences surf-riding and broaching in two wave conditions, respectively. Surf-riding and broaching often occur on the down slope of a wave, and broaching always accompanied with a large heel angle,

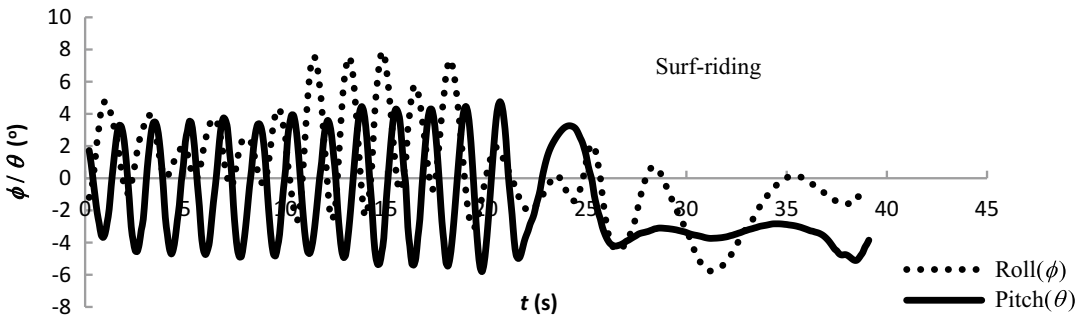


Surf-Riding and Broaching, Fig. 5 A snapshot of broaching in the experiment

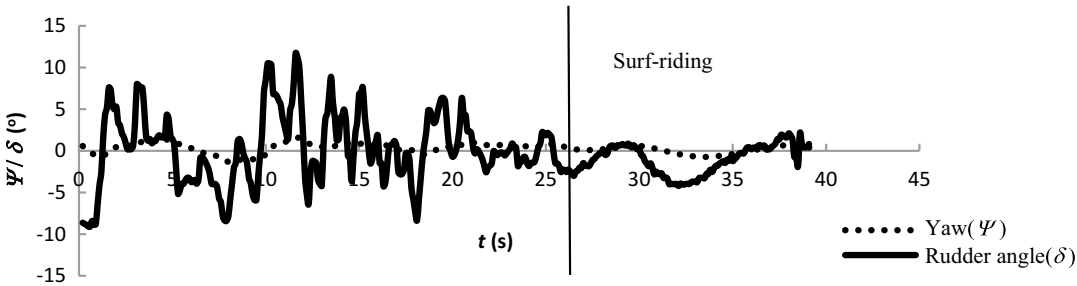
which may lead to stability failure, or even capsizing.

The experiment results in following waves are shown in Figs. 6 and 7. The pitch motion of the ship appears periodic at the beginning, and then the amplitude of pitch motion is almost unchanged in later time. While yaw motion is generally small all the time. It reveals that the ship experiences stable surf-riding.

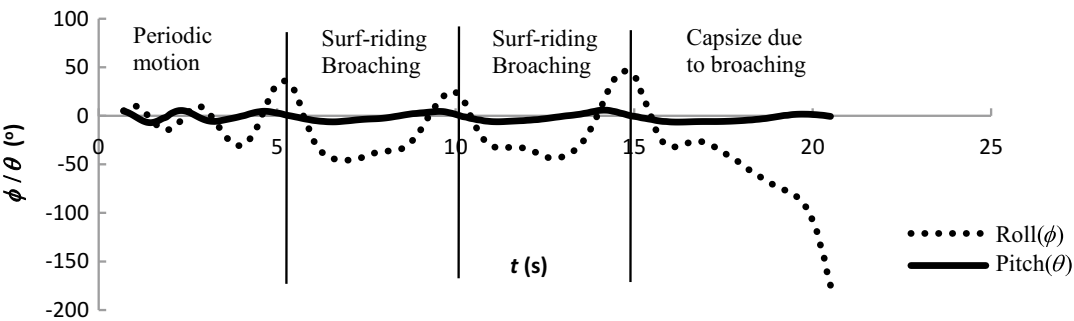
With the wave steepness increasing and the heading changing to stern-quartering waves as shown in Figs. 8 and 9, the ship surf-riding occurs quickly. Then the ship cannot keep its course even with maximum steering effort, and broaching occurs. At the same time, roll angle is increasing fast. But with the action of rudders, the ship is stable at a new heading temporarily. And then the ship is captured again by a new wave and surf-riding and broaching occur once more. At the



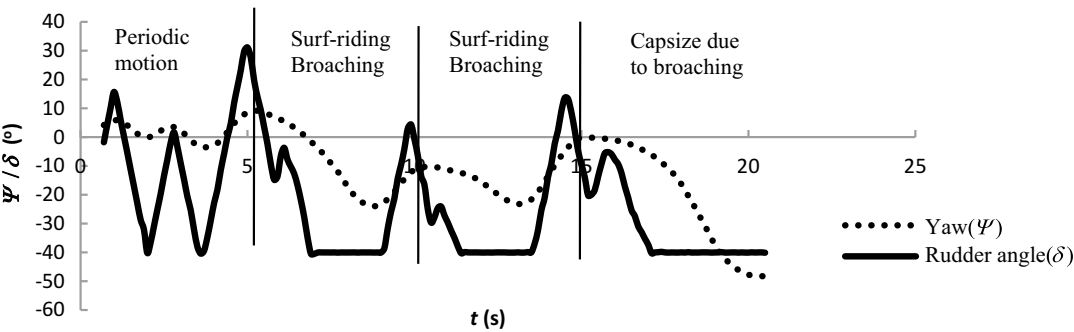
Surf-Riding and Broaching, Fig. 6 Time histories of roll and pitch ($Fn = 0.4$, $\lambda/L = 1.25$, $H/\lambda = 0.05$, following waves $\chi = 0^\circ$)



Surf-Riding and Broaching, Fig. 7 Time histories of yaw and rudder angle ($Fn = 0.4$, $\lambda/L = 1.25$, $H/\lambda = 0.05$, following waves $\chi = 0^\circ$)



Surf-Riding and Broaching, Fig. 8 Time histories of roll and pitch ($Fn = 0.4$, $\lambda/L = 1.25$, $H/\lambda = 0.06$, stern-quartering waves $\chi = 30^\circ$)



Surf-Riding and Broaching, Fig. 9 Time histories of yaw and rudder angle ($Fn = 0.4$, $\lambda/L = 1.25$, $H/\lambda = 0.06$, stern-quartering waves $\chi = 30^\circ$)

third broaching event, the roll angle is so large that the ship occurs capsize at last.

Survey Vessel

► [Research Ship](#)

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SWAN - Simulating Waves Nearshore

► [Resource Assessment](#)

Synthetic Aperture Sonar (SAS)

► [Environmental Perception for Underwater Vehicles](#)

Synthetic Fiber Rope

- ▶ [Taut Mooring](#)

System Decomposition

- ▶ [Multidisciplinary Design Optimization \(MDO\)](#)

Synthetic Lines

- ▶ [Dynamic Analysis Method](#)

System Modeling

- ▶ [Multidisciplinary Design Optimization \(MDO\)](#)

T

Tandem Propulsion System (TPS)

► [AUV/ROV/HOV Propulsion System](#)

TAO: Tropical Atmosphere Ocean Buoy

► [Tropical Atmosphere Ocean \(TAO\) Buoy](#)

Taut Mooring

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Synonyms

[Deepwater](#); [Synthetic fiber rope](#); [Vertical load anchor](#)

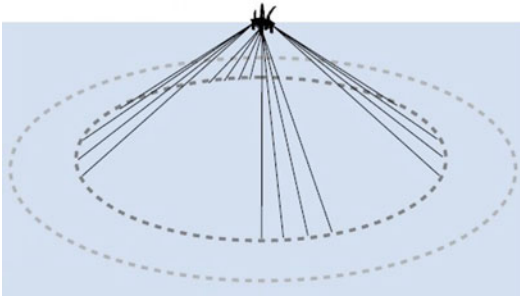
Definition

Taut mooring, as a new type of mooring systems for deepwater operation platforms, maintains taut condition by applying pretension on both ends of the cable. The composition materials tend to choose the combination form of the chain-

polyester-chain to meet the needs of deep-sea work. It has been widely used due to its small mooring radius, low total mass, and good corrosion resistance.

Introduction

With the increasing scarcity of oil and gas resources on land and in the shallow sea, development toward the deep sea has become an inevitable trend. But the harsh environment on the ocean and the positioning requirements of the deep-sea drilling platform have made the research, design, and manufacture of the deep-sea marine structure mooring system face new challenges and once became the technical bottleneck of deep-sea oil and gas exploration. In recent years, research and development of new mooring methods and mooring materials have made great progress, from the traditional catenary mooring system to the deep-sea drilling platform, such as the taut mooring system (Fig. 1) adopted by Spar and TLP platforms. Traditional large-mass anchor chains and steel cables are increasingly unable to meet the needs of deep-sea drilling and development. With the increased depth of the floating structure operation, the chain will be a rapid increase in weight, not only increases the tension in the chain, and increased by the vertical load of the floating structure, such that the payload further floating structure can have the reduction. In addition, with the increase



Taut Mooring, Fig. 1 Taut mooring system of a semi-submersible platform. (From anchor manual 2005)

of water depth, positioning capabilities of chain mooring system will decline rapidly. Thus, deep-water mooring system for floating marine structures often uses taut mooring.

Composition Materials

The taut mooring system was developed with the application of fiber materials to the deep sea and was first applied to the tension leg platform and the Spar platform in the 1980s. It fixes the platform directly to the sea floor through mooring cables. The middle part of the mooring cable is mostly made of light-weight high-strength nylon rope, polyester rope or other synthetic materials, and wear-resistant steel anchor chains at both ends of the cable or wire rope (Fig. 2). Synthetic fiber ropes are usually made of PET, HMPE, Aramid, etc., of which polyester fiber is the most widely used. By comparison, these materials used in taut mooring systems have a larger strength-mass ratio to mitigate the loss of vessel effective payload, low axial stiffness to facilitate the rope elongation to adapt the vessel offset, obviously shorter mooring lines to reduce the mooring system footprint, and larger restoring forces to decrease the vessel offset (Loevstad et al. 1995). The synthetic fiber ropes have become one of the keys to the study of mooring systems due to the wide application of taut mooring systems on deep-sea drilling platforms (Johnsen and Ritsland 1999).

In 1997, Petrobras first used a polyester rope + anchor chain taut mooring system of the FPSO.

This mooring system can effectively reduce the vertical load and displacement under the harsh sea conditions compared with the anchor chain and the steel cable, saving a lot of economic costs. The rope comes in at a 30–45° angle on the seabed where it meets the anchor (suction piles or vertically loaded anchors), which is loaded vertically. When the platform drifts horizontally with wind or current, the lines stretch, and this sets up an opposing force.

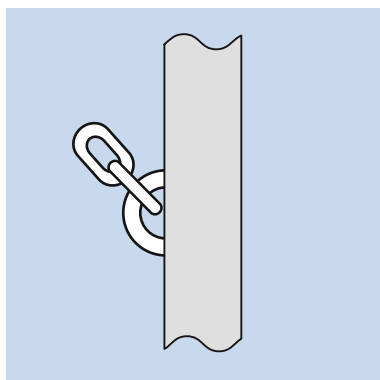
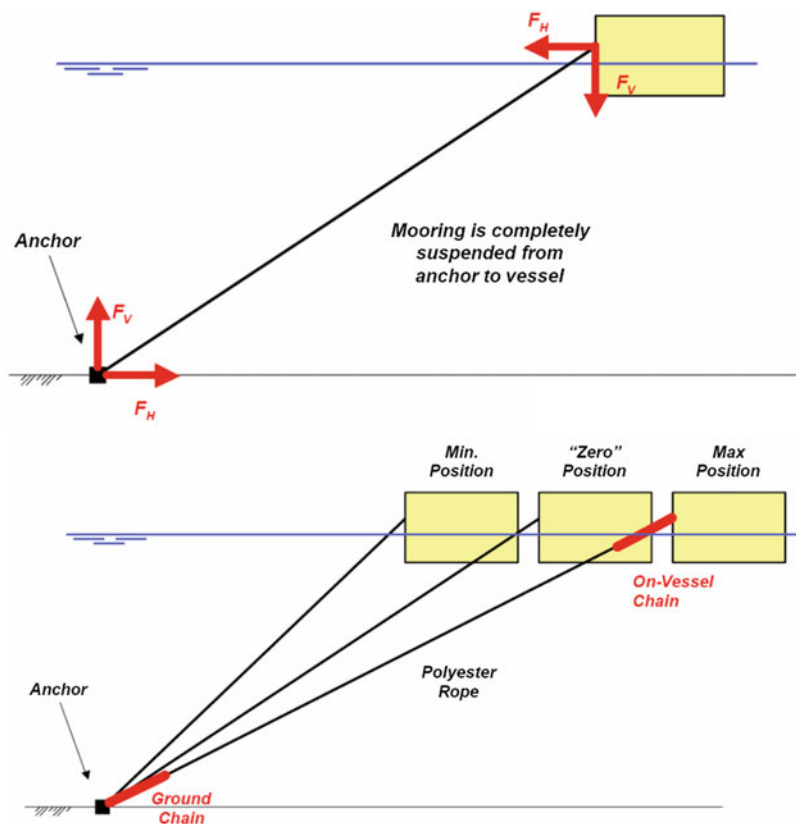
Selection of Anchors

The major difference between a catenary mooring and a taut leg mooring is that when the catenary mooring arrives at the seabed horizontally, the taut leg mooring arrives at the seabed at an angle. This means that in a taut leg mooring, the anchor point has to be capable of resisting both horizontal and vertical forces (Fig. 2), while in a catenary mooring, the anchor point is only subjected to horizontal forces. In a catenary mooring, most of the restoring forces are generated by the weight of the mooring line. In a taut leg mooring, the restoring forces are generated by the elasticity of the mooring line (Loevstad et al. 1995).

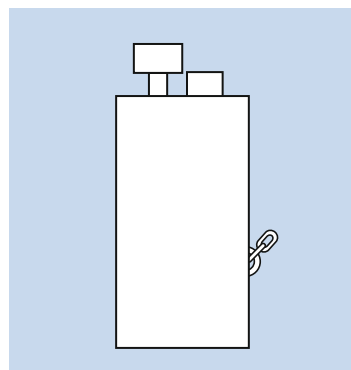
Taut leg mooring system, which due to the taut profile of the lines, has no grounded line lying on the seabed in the nominal pretensioned condition. Therefore, the seabed anchor both are subjected to horizontal forces and vertical forces, and the mooring line anchor in a taut mooring line experiences an uplift angle (at the nominal burial point of the line) under pretension. Furthermore, the mooring line's uplift angle at the burial point can significantly exceed the 10° limit for high-holding capacity drag-embedment anchors. In this case, pile anchors, suction anchors, and vertical load anchors are more suitable for taut mooring systems.

The pile anchor (Fig. 3) is a hollow steel pipe that is installed into the seabed by means of a piling hammer or vibrator. The holding capacity of the pile is generated by the friction of the soil along the pile and lateral soil resistance. Generally, the pile has to be installed at great

Taut Mooring,
Fig. 2 Taut mooring line



Taut Mooring, Fig. 3 Pipe anchor. (From anchor manual 2005)

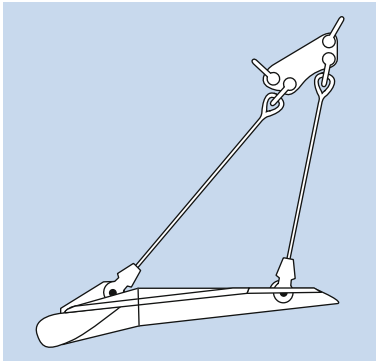


Taut Mooring, Fig. 4 Suction anchor. (From anchor manual 2005)

depth below seabed to obtain the required holding capacity.

The suction anchor (Fig. 4), like the pile anchor, is a hollow steel pipe, although the diameter of the pipe is much larger than that of the pile. The suction anchor is forced into the

seabed by means of a pump connected to the top of the pipe, creating a pressure difference. When pressure inside the pipe is lower than outside, the pipe is sucked into the seabed. After installation the pump is removed. The holding capacity of the suction anchor is generated by the friction



Taut Mooring, Fig. 5 Vertical load anchor. (From anchor manual 2005)

of the soil along the suction anchor and lateral soil resistance (Wang et al. 2012).

A new development is the vertical load anchor (VLA, Fig. 5). The vertical load anchor is installed like a conventional drag embedment anchor but penetrates much deeper. When the anchor mode is changed from the installation mode to the vertical (normal) loading mode, a VLA is installed just like a conventional drag embedment anchor. During installation (pull-in mode), the load arrives at an angle of approximately $45\text{--}50^\circ$ to the fluke. After triggering the anchor to the normal load position, the load always arrives perpendicular to the fluke. This change in load direction generates 2.5–3 times more holding capacity in relation to the installation load.

All three anchors have a common feature that can withstand horizontal and vertical loads. As VLA is deeply embedded and always loaded in a direction normal to the fluke, the load can be applied in any direction. Consequently, the anchor is ideal for taut leg mooring systems, where generally the angle between mooring line and seabed varies from 30° to 45° .

Advantages and Disadvantages

The structural analysis involving the taut mooring line is the same as the catenary, except that the taut mooring line can be regarded as a straight line, and it is easy to obtain its static characteristics. As for the dynamic analysis of mooring lines,

the lumped mass method and the slender rod theory are commonly used.

In general, the taut mooring system has four advantages compared to catenary mooring system:

1. The synthetic fiber has large recovery stiffness and provides greater restoring force, and the platform horizontal deviation decreases more.
2. Under the same water depth, shorter mooring cables are needed, which save costs, especially in deepwater and ultra-deepwater conditions.
3. The mooring line used in the mooring system is reduced in length while avoiding the application of large-mass steel chains, thereby reducing the mooring gravity, i.e., reducing the load on the platform and increasing the variable load of the platform.
4. With a small mooring radius, the mooring foundation occupies a small area of the seabed, reducing the risk of collision with other underwater facilities nearby. In the catenary mooring system, the catenary line has a larger vertical angle with the platform, so that the anchor chain has a large downward pulling force on the platform, which reduces the effective load of the platform. The application of the pontoon increases the mooring material and installation costs and increases the mooring drag force. The taut mooring system can effectively solve this problem.

The disadvantage of the taut mooring system is that the mooring needs to bear huge tension and axial stress under working conditions, so higher requirements are imposed on the characteristics of mooring modulus and life, and the floating structure is required to provide a large reserve buoyancy (Fan et al. 2010).

Cross-References

- Catenary Anchor Leg Mooring
- Deepwater
- Floating Structure
- Single Anchor Leg Mooring
- Station Keeping System
- Synthetic Fiber Rope
- Vertical Load Anchor

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Taut Mooring Lines

- [Static Analysis Method](#)

TCell (Truss Cell Spar)

- [SPAR Platform](#)

TDB: Tsunami Detection Buoy

- [Tsunami Warning Buoy](#)

Technical and Economical Barriers on Green Energy Utilization in Shipping

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Synonyms

[Design and installation](#); [Economic barriers](#); [EEDI \(Energy Efficiency Design Index\)](#); [Environmental impact](#); [Fuel prices](#); [Green energy](#); [Operational challenges](#); [Shipping energy efficiency](#)

Definition

More than 90% of the world trade is transported by ships. Due to the growth of maritime transport, emissions from shipping are expected to increase 50–250% by 2050. Some countries have set up climate goal with 20% improvement of energy efficiency and a minimum of 10% renewable energy in transport sector by 2020. According to the inventory study carried out by Nelissen et al. (2016), the utilization of green energy as a ship's auxiliary propulsion is one of the most promising technologies to decarbonize maritime transport. In addition to reduce fuel cost and environmental impact, the requirements to comply with regulations on utilizing renewable energy will create large opportunities for the development of green energy technologies used in the shipping industry. Among all green energy technologies, the utilization of wind power as auxiliary propulsion seems to provide the most mature solutions in the market. The majority of the green propulsion technologies/concepts have been proposed only for bulkers, tankers, and general cargo vessels. However, from the shipping company's perspective, i.e., the demand side of wind propulsion technologies, only two commercial ships are installed a wind propulsion technology. While from the technology provider's perspective, only two providers' products are close to marketability, and all the other dozens of green propulsion technologies/concepts could be available but not before 2030. In the following, some facts regarding the green energy propulsion concepts will be given first. Then, technological barriers associated with the green propulsion technologies will be presented. Finally, the economic barriers connected with their marketability will be discussed. It should be noted that this section focuses on the barriers of wind propulsion technologies, which share similar problems with other types of green technologies concepts.

Technology Factors

Various concepts of green shipping technologies are under development, and the aim of these technologies is to optimize the utilization of

renewable green energy during a ship's sailing period. Taken the most promising wind propulsion technology, for example, for the conventional sail propulsive ships, it is important to evaluate/optimize the aerodynamic and energetic features of the designed wind propulsion technologies. That is the central issue because all technologies must provide enough power/propulsion forces to ships to make these concepts profitable to be installed onboard ships. Furthermore, when introduce these concepts to provide auxiliary propulsion in a merchant ship, some kinds of automation should be of high demand to be self-adjusted to various wind conditions without causing any extra work load of the crew. Moreover, equipment should be considered for conditions that the green shipping technologies are not allowed to be operated, e.g., in harsh weather conditions or harbor. The propulsion from these technologies should be minimized to keep the ship safe in those conditions. It should be noted that different concepts are associated with different degrees of complexity. For wind propulsion technologies, an inventory study by Nelissen et al. (2016) indicates their potential energy gains for tankers and bulk carriers listed in Table 1. In this study, five different well-recognized wind propulsion concepts are compared: Flettner rotor, sails/wingsails, towing kites, and wind turbines, for four potential ships. The sailing information of these ships are obtained from AIS-recorded ship passage profiles, and the wind technologies characteristics are based on the current state of development. However, the results from the inventory analysis to a large extent depend on the assumptions and parameters defining the wind propulsion devices. The analysis simply provides a preliminary results/energy saving that can be realized by such concepts. In the final marketing stage, a specific technology may be further optimized with respective to each individual ship and trade factors like investment cost, ease of installation/operation, and others.

It should be noted here that for those big ships, only 1% of the energy saving can mean hundreds of thousands of USD fuel saving every year. As shown in Table 1, except for the wind turbine concept (this idea is to install a commercial

Technical and Economical Barriers on Green Energy Utilization in Shipping, Table 1 Average energy savings using different wind propulsion technologies based on the AIS information of each ship type for their average operation profiles (Nelissen et al. 2016)

	Flettner rotor (%)	Wingsail (%)	Towing kite (%)	Wind turbine (%)
Large bulk carrier	17	18	5	2
Small bulk carrier	5	5	9	1
Large tanker	9	9	3	1
Small tanker	5	5	9	1

windmill onboard a ship to provide electricity and possible propulsions to ships.), all other concepts show big potential to be exploited in the shipping market. Furthermore, the relative energy savings from auxiliary propulsion concepts are higher for ships sailing at lower speed. This is because first the lower speed sailing ships may require the much lower power and, second, the energy gains from those green shipping propulsion concepts are also higher due to relatively higher auxiliary propulsive power. In addition, larger vessels can be installed green shipping propulsion devices of large power capability, leading to higher auxiliary power capture (energy saving).

Even though the above inventory study indicated large potential energy saving by installing such green shipping propulsion devices, it is still not recognized as mature technologies acceptable for current shipping companies to make practical investment. The barriers that hinder market installation are very complex. It can be categorized into technical and economic barriers, which will be further described in the following two subsections.

Technical Barriers

In principle, most of technical barriers are associated with the factor that those concepts are often provided with lack of reliable evaluation

procedure to convince customers regarding their actual gains (energy saving) against the practical investment. This kind of information requires reliable life cycle cost-benefit evaluation of those green shipping concepts. This evaluation needs multidiscipline knowledge and tools such as naval architecture, sea environments, mathematics and programming, etc. to develop the generic analysis method and optimization tool. In addition, to ensure ship safety with encountered extreme storm conditions, it is important to understand and simulate the physical phenomena of aerodynamic (wind forces from sails) and hydrodynamic (wave forces on ships) interactions. Good understanding of all these disciplines will also boost further development of the concept, as well as to educate our students as future naval architects. In summary, the key technical barriers can be related to reliable data on a ship and devices' mutual performance, operability, economic potentials and safety, as well as incentives from shipping markets/regulations to improve ships' energy efficiency and reduce CO₂ emissions from ships.

Design and Installation Challenges

The work efficiency of the green shipping concepts is of the greatest interest for their practical installation and practical applications. In general, to generate large propulsion force to provide auxiliary propulsion to merchant ships, the green shipping technology devices should be appropriately big enough. Consequently, the investment to design, construction, and installation is usually a big amount of cost. In addition, some of these technology devices should be designed specifically for individual ships to obtain larger energy gains than their actual investment for practical installations. For wind propulsion concepts, the aerodynamic performance of these devices was extensively studied via both wind tunnel tests and even the CFD (computational fluid dynamics) analysis, e.g., Flettner (1925), Blackford (1985), Argatov et al. (2009), Johnson and Andersson (2011), Cooke Association (2014), etc. However, the reliability of those established performance

models of these devices obviously requires further development. For example, some devices have a high weight and center of gravity, which may affect a ship's seakeeping behavior and also a high installation cost. In unfavorable winds, wind propulsion devices will create large drag forces, which may lead a ship to dangerous conditions of large ship heeling. Furthermore, since large wind devices have to be operated by cranes, for conditions that the height of devices is large enough to affect a ship's normal navigation, such as passing a bridge with marginal height clearance, the fold/launch equipment will significantly increase the installation cost (Bose 2008).

Furthermore, due to the severe working environments (high moisture and salty) of these green shipping devices, long-term maintenance cost during their operations should be seriously considered. For ship owners, the preferred payback time of their investment is 5–10 years. But usually the current payback time is often longer than that. For example, according to O'Rourke (2006), in order to install the most "simple" wind device such as a skysail, the cost of retrofitting a cargo ship with a row of masts, and strengthening the ship to dissipate stress due to the sail, was estimated to be approximately €10 m, around 15 years to recoup their costs through fuel savings. In addition, usually large amount of maintenance cost are associated with the initial installation investments of such expensive devices. The maintenance sometimes could double the payback time. It should be noted that the fuel saving cost is also strong related with future market oil price and future fuel cost scenarios for reduction of shipping emissions (Eide et al. (2011) and Faber et al. (2009)), which may shorten the payback time and make the wind-assist concepts more attractive to the shipping market.

For example, due to large forces generated by the wind devices, ships' structural components should be strengthened to enhance ship structural safety. For example, following the success of wingsail implementation on the tanker Shin Aitoku Maru, another vessel installed a similar wingsail onboard went wrong during high wind in a Japanese harbor. The big forces partly from the wingsail broke moorings of the vessel, which

was blown across the harbor. She seriously damaged other vessels in the harbor as well as the harbor itself. The accident discredited wingsail safety and indicated further technical development of such wingsail applications.

Operational Challenges

In addition to the design and construction of wind-assist devices, operation of the ship fitted with these devices and associated handling/control system requires robust performance prediction models to describe ship and wind devices' response in real ship operations. It is known that numerically modelling complex ship motions in real sea environments has always been one of the greatest challenges for the hydrodynamic analysis in the maritime industry. While the modelling of wind-assist ship performance requires even more sophisticated aerodynamic and hydrodynamic coupling analysis.

It is known a sailing ship sails with a constant heel and leeway angle to make use of wind propulsions to power the ship, while current merchant ships are mainly powered by marine engines connected with well-designed propellers. The installation of auxiliary wind propulsion devices will add at least two extra wind forces to ships depending on the number and arrangements of the devices. Wind forces, which should be big enough to ensure device-working efficiency (note that large wind-assist devices may block a ship's operation view on the bridge, which is quite essential in port operations), will significantly affect a ship's response behavior such as ship resistance, propeller efficiency, ship maneuverability, ship motions, etc. Furthermore, drag and lift forces generated from wind-assist devices have to interact with the thrust force provided by a ship's propellers. The ship's sailing course might be also affected by such interactions. To achieve highest work efficiency of these devices, an optimization procedure should be developed to guide the operation of the ship. It is often referred to routing optimization (optimum voyage plan) systems. In this case, routing optimization is a key factor to help captains to operate their ships to

maximize the utilization of wind propulsion through the iterative consideration of wind device performances and the ship's main power management to achieve the expected time of arrival with minimum fuel cost. In the following section, the characteristics of routing systems available in the current shipping market will be summarized in order to give a state-of-the-art description of such a service. It is expected to indicate further development requirements and directions to consider the wind-assist device in the future routing systems.

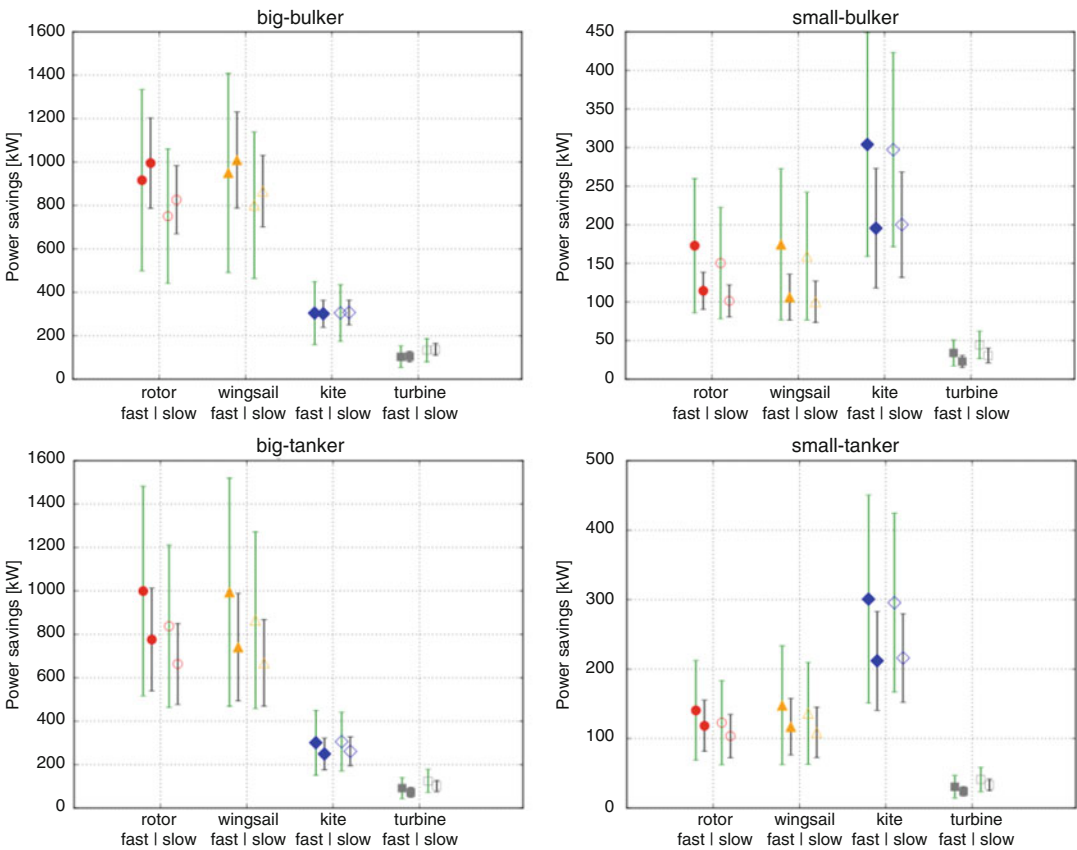
Moreover, the fuel savings by implementing these devices could be even maximized through the optimized planning of ship routing for each individual ship voyage. Currently, most of ocean-crossing vessels are instrumented with routing plan system, where routing optimization tool is used to assist ship operation in a more optimal way based on weather forecast information. The routing tool has the potential to save up to 5% of fuel cost and reduce 50% structural fatigue damage (Mao et al. 2012). The routing optimization system is also one of the key factors to maximize the use of wind propulsion in practical shipping. (Spaans 1985) investigated different optimization algorithms for the utilization of wind-assist devices to reduce energy cost of shipping. However, there is no routing tool, which can consider lift and drag wind forces for the routing optimization.

The Uncertainty of the Cost Efficiency

In general, there are large uncertainties included in the estimation of fuel savings by implementing these devices. Most often, these energy-saving devices need large amounts of investments for their implementations. One important uncertainty is due to lack of reliable methods/models to predict the performance (aerodynamic and hydrodynamic coupling analysis) of these devices in the real operation conditions, e.g., WASP (1985), WINDTECH (1985), etc. Another reason is that the estimation of fuel savings should rely on the encountered sea environments (trade routes, wind, wave, etc.) of a ship's future sailing service.

In addition to the nontransparent data and models for energy-efficient assessment, other barriers that hinder the implementation of the energy-saving measures in the shipping market are categorized by Jafarzadeh et al. (2012), which was carried out based on extensive market survey by Faber et al. (2009), Cagno et al. (2013), etc. As the case study mentioned in Nelissen et al. (2016), Fig. 1 presents uncertainties of energy saving by implementing four wind propulsion concepts for four different ships. In most cases, the scatter of potential energy gains is quite big. The technology providers should develop methods to evaluate energy saving provided by their technologies in a more accurate basis to convince potential customers in the shipping market.

In addition, services of implementing the measures within machinery system are often provided by large marine engine companies. The main barriers for such implementations are also connected with economic concerns from ship owners' perspective, or organizational barriers that the shipping company may have different priorities in comparison with the measures' implementations. Information barriers are of essential for the application of such measures, since the data and analysis provided by engine companies are often lack of transparency. The consequence is that larger percentages of energy savings are claims than it is actually observed in the market. The information barriers can be also found in retrofitting energy-saving



Technical and Economical Barriers on Green Energy Utilization in Shipping, Fig. 1 Average absolute power savings (solid symbols show savings averaged over sample

routes, empty symbols averaged over sample fleet voyage profiles from AIS tracks)

devices, such as installing a duct to harmonize and stabilize the flow distribution entering into the propeller to increase propulsion efficiency, reconfiguration of propellers, etc. On the other side, the lack of information can be also induced by the lack of technical solutions to prove the efficiency of the energy-saving devices.

Currently, the technical maturity of these measures should be further developed to ensure the confidence to estimate their investment and payback time, for example, the utilization of wind power for ship auxiliary propulsions. Even though large potentials can be expected from such measures, their arrangement onboard ships, installations, maintenances and its impacts to a ship's overall safety and maneuverability, as well as the optimum utilization of the wind forces provided by them require a complex system with different levels of models and technologies to support the whole concept. In addition to retrofitting or updating sub-systems in the existing ships, optimal ship operation managements, such as slow steaming, optimization voyage plan, etc., are rather cheaper but straightforward, efficient, and most implemented measured in the shipping industry.

Because of fast advancement of information and communication technology, the increased reliability of weather forecast provided by meteorological institutes, large amounts of free access to marine weather measurements, robust communication between onshore operation office and ship bridges, and more energy saving from the optimal ship operation will be realized through the development and implementation of optimization voyage plan system by integrating different levels of measures, such as wind power devices.

Market Potential and Economic Barriers

According to the inventory study carried out by Nelissen et al. (2016), only for the utilization of wind power concepts as auxiliary propulsion, the maximum market potential installation onboard commercial ships is estimated to be around 3700–10,700 devices/systems until 2030. It contains the installation for both retrofits and installations on newbuilds, and the precise number of

installations depends on the future bunker fuel price, and the cost reduction of these technologies can be achieved because of economies of scales through mass production. The use of these wind propulsion systems would help to reduce CO₂ emissions to around 3.5–7.5 Mt in 2030.

Even though there is a large market potential for the green shipping concepts, the diffusion process may not be perfectly mature in 2030 yet, and it is expected to occur when more newbuilds have entered the market with observed profits from their installations. Before the mass penetration into the market, in addition to the technical barriers, economic hindrances are also referred to the interorganizational barriers for the implementation of energy-efficient measures. For example, even within a same company, top managements may have different motivations from technical advisors to update their conventional designs to the more energy-efficient ship concepts, which are often associated with risk of the actual energy gains through the new implementations. In some cases, the organizational barriers are also combined or due to the technical maturity of certain measures and methodologies. For example, it is well recognized that a ship's optimization design should be carried out for her actual operational conditions and future service weather environments, while the construction contracts between shipyards and ship owners are normally referred to the calm water condition with specific service speed. All the sub-systems should be arranged and optimized based on requirements for the ship operated in calm waters, while ships are rarely sailing in such conditions. The reason behind is that the mature information and technology for such condition is available to judge the success of the ship construction project. Furthermore, the incentive for current shipping companies is mainly associated with the compliance to environmental regulations to improve the energy efficiency in ships. The factors of making actual profits through the installation of energy efficiency technologies are not solid/mature enough to convince the shipping market.

To breach the marketing problem, a standardized method to assess green energy propulsion

systems should be developed and validated with test cases. Only if assessable information produced through the standardized approach by an independent party, ship owners and investors could be convinced for practical investment and public funds for practical installation. On the other side, it will motivate the further development of the technology and provide valuable information regarding actual gains through the development. When developing a standardized method to assess practical gains by the green shipping concepts, the consistency of the regulation incentives, such as the Energy Efficiency Design Index (EEDI), technical guidance for the conducting performance tests of those technologies should be carefully considered. It should be noted to the market that the determination of the actual fuel savings is preferable.

Cross-References

- [Energy Efficiency Regulations from IMO](#)
- [Future Trend on the Design, Building, and Operation of Ships for Energy Efficiency](#)
- [New Technologies in Auxiliary Propulsions](#)

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Technical Design

- [Concept Design](#)
- [Design of Submersibles](#)
- [Design Spiral](#)
- [Detailed Design](#)
- [Empirical Design](#)
- [Preliminary Design](#)

Temperature

- [AUV/ROV/HOV Hydrostatics](#)

Tension Leg Platform

- [Shallow Foundations](#)

Tension Leg Platform (TLP)

► [Decommissioning of Offshore Oil and Gas Installations](#)

Tension Leg Platforms (TLP)

► [Steel Pipelines and Risers](#)

Tension Leg Platforms (TLPs)

► [Christmas Tree](#)

Tension-Leg platform

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Synonyms

[Capex](#) (capital expenditure); [ETLP](#) (extended tension leg platform); [FPSO](#) (floating production storage and offloading); [MOSES TLP](#); [SeaStar TLP](#); [TLP](#) (tension leg platform)

Definition

A *tension leg platform (TLP)* or *extended tension leg platform (ETLP)* is a vertically moored floating structure normally used for the offshore production of oil or gas and is particularly suited for water depths greater than 300 m (about 1000 ft.) and less than 1500 m (about 4900 ft.). Use of tension leg platforms has also been proposed for wind turbines (https://en.wikipedia.org/wiki/Tension-leg_platform).

Introduction

With the increase of water depth, a tension leg platform (TLP) was developed based on the semisubmersible platform in order to optimize the heave motion of the platform. The tension leg platform consists of the upper block, the floating body, the tension leg, the top tension wellhead riser, the catenary riser (external/internal input), and the pile foundation. Several tendons composing the tension connected to the foundation of the subsea pile serve to tighten and fix the platform and control the movement of the floating body within a certain range. TLPs are divided into two types, traditional and new ones, and subdivided into four types, conventional TLP, Seastar TLP, MOSES (MOSES TLP), and extended TLP (ETLP).

One of the most important characteristics of TLP platform is that its buoyance is larger than its weight, which leads to the large tension in the tendons. Therefore, TLP platform can be regarded as an inverse pendulum, which brings TLP platform not only outstanding motion characteristics but also strict requirement on onboard weight control.

Distribution Worldwide

According to the latest statistics of *Offshore Magazine*, there are 26 TLP platforms in the world, including 11 conventional TLPs and 15 new generation TLPs; 16 of them are in the Gulf of Mexico, 1 each in the UK, Brazil, and Indonesia, and 2 each in Norway and Angola and Equatorial Guinea; of the 25 platforms, 7 are wet tree oil production, 1 has dry and wet tree oil production, and 17 are completely dry trees.

Since 1984, there have been many TLP platforms that have been put into production and operation in the world. They have excellent working records, which has strengthened the confidence for industry's structure of the emerging offshore platform of TLP. During its development, researchers and engineers from all over the world have accumulated rich experience in design and application and technical data, laying a solid

foundation for the development of the TLP platform.

TLP platform has a wide range of working water depths and oil production modes. Different countries have formed different oil production modes according to the characteristics of their own sea areas. At this moment, the “American model” and the “Brazil model” are the most typical ones. The “American model” is a semi-sea semi-continental type, which is like the engineering model of “floating drilling platform (TLP/SPAR) – underwater wellhead/underwater production system – submarine pipe network.” The Brazilian model belongs to the all-sea model, that is, the oil and gas field development engineering model of “semisubmersible platform – underwater wellhead/underwater production system – floating production storage and offloading unit (FPSO).” In addition, the South China Sea oil production model can draw on the successful experience of the USA and Brazil to form an oil field development model of “floating drilling platform, FPSO, and underwater wellhead/underwater production system” (Figs. 1 and 2).

History of TLP Development

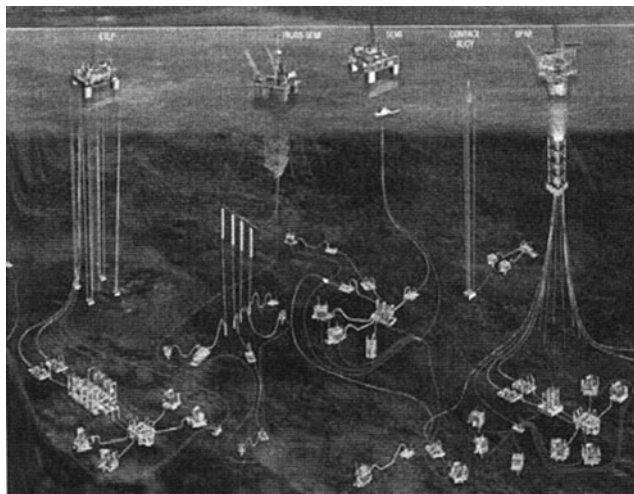
In 1954 American R. O. Marsh proposed a TLP scheme with a cable group fixed by tilting

mooring in the first place, which immediately attracted great attention from offshore engineers. In 1962, the British Petroleum Development Corporation built a 124 t triangular TLP test platform called “Triton” at a site with water depth of 30 m in the Scotland coastline. It is proved that this platform has excellent motion performance in waves. In 1974, the US Deepwater Petroleum Technology Company placed a 650 t TLP test platform named “Deep Oil X-1” at a depth of 60 m in the waters around California, conducting 5 years of experimental research and theoretical analysis on its movement. At the same time, the marine engineering research institutions in Japan, Italy, Norway, the Netherlands, and other countries have invested a lot of manpower and resources to carry out a large number of investigations, on TLP’s overall performance, main scale optimization, tension leg tension characteristics, and specific links in the process of construction and installation.

Based on nearly 30 years of experimental and theoretical investigations, in August 1984, the world’s first practical TLP was completed and put into operation by Conoco in the water depth of 157 m in the Hutton oil field in the North Sea of the UK. Subsequently, in November 1989, Continental Petroleum Corporation built a tension leg wellhead platform at a depth of 528 m in the Jolliet oil field in the Gulf of Mexico. In 1992, Saga Oil Company of Norway built a large

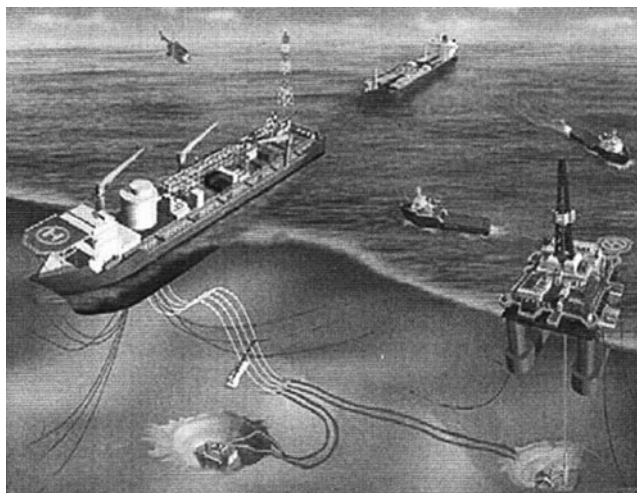
Tension-Leg platform,

Fig. 1 US oil and gas development model



Tension-Leg platform,

Fig. 2 Brazil oil and gas development model



TLP in the Snorre field in the North Sea of Norway with a water depth of 340 m. In June 1995, Conoco installed the world's first concrete TLP at a depth of 330 m in the Heidrun oil field in the North Sea. In 1996, Shell and BP Exploration installed the Mars tension leg platform in the Canyon 807 block in Mississippi, USA, to rewrite the water depth record to 880 m. The first SeaStars TLP was built in 1998, the first MOSES TLP was built in 2001, and the first ETLT was installed in 2003.

The development status of TLP platform in recent years is as follows (Table 1).

Structural Components of TLP

Description

The main body of the traditional TLP platform is generally rectangular or triangular. The upper body of the platform is above the water surface. The lower body is connected to the upper body by four or three columns which are the cylindrical structure. The main function of these columns is to provide the necessary structural rigidity to the platform body. The buoyancy of the platform is provided by a sinking pontoon located below the surface of the water, and the head and tail of the pontoon are connected to the columns to form a ring structure.

The tension leg is composed of 1–4 tension tendons. In terms of these tendons, the upper end is fixed on the platform body, and the lower end is connected with the submarine base template or directly connected to the top of the pile foundation. Sometimes in order to increase the lateral stiffness of the platform system, a diagonal mooring system is also installed as an aid to the vertical tension leg system. The foundation on the seabed is fixed in place, mainly in the form of pile foundation or suction foundation. The central well is located on the upper body of the platform and can support the dry tree system. The production riser is connected to the production equipment through the middle well and is connected to the subsea oil well (Figs. 3 and 4).

An TLP platform normally has five parts of structural composition:

1. Upper platform: The upper body of the platform refers to the part above the lower deck of the TLP. It is equipped with production, living equipment, and auxiliary facilities. The shape of the platform can be triangular, quadrangular, and pentagon, but majority of TLP platforms use quadrilateral shape. The main function of upper platform is to support the production module, accommodation module, helicopter deck, etc.
2. Column: The columns are mostly larger diameter cylinder. The number of columns depends

Tension-Leg platform, Table 1 Statistics of TLP in service

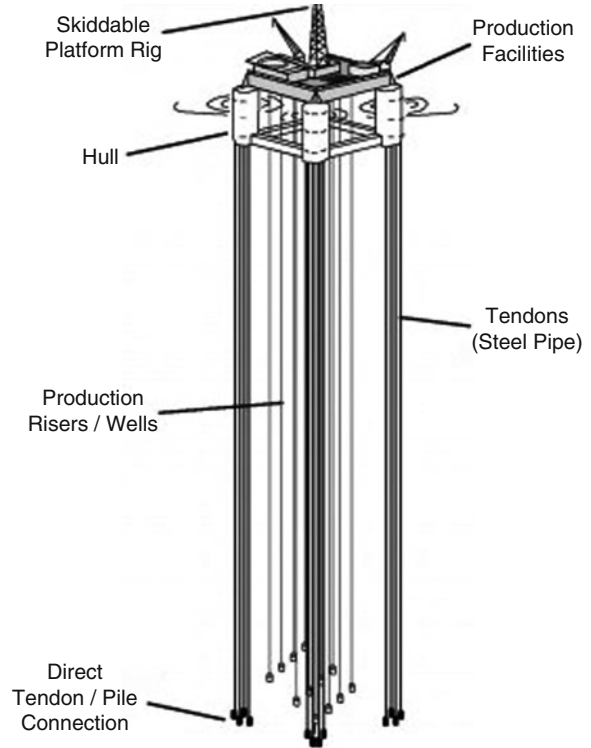
Name	Production time	Owners	Operation sea	Water depth (m)	Foundation	Type
Hutton	1984	Conoco	North Sea	157	Gravity	CTLP
Jollot	1989	Conoco	Gulf of Mexico	542	Pile foundation	CTLP
Snorre A	1992	Saga	North Sea	335	Gravity suction anchor	CTLP
Auger	1994	Shell	Gulf of Mexico	872	Pile foundation	CTLP
Heidrun	1995	Conoco	North Sea	350	Gravity suction anchor	CTLP
Mars	1996	Shell	Gulf of Mexico	894	Pile foundation	CTLP
Ram/Powell	1997	Shell	Gulf of Mexico	980	Pile foundation	CTLP
Morpeth	1998	British Borneo	Gulf of Mexico	518	Pile foundation	SeaStar
Allegheny	1999	British Borneo	Gulf of Mexico	1006	Pile foundation	SeaStar
Ursa	1999	Shell	Gulf of Mexico	1158	Pile foundation	CTLP
Brutus	2001	Shell	Gulf of Mexico	910	Pile foundation	CTLP
Prince	2001	EIPaso	Gulf of Mexico	454	Pile foundation	MOSES
Matterhorn	2003	Total Fina Elf	Gulf of Mexico	860	Pile foundation	SeaStar
West/Seno A	2003	Unocal	Indonesia	975	Pile foundation	CTLP
Kizomba A	2004	ExxonMobil	West Africa	1250	Pile foundation	ETLP
Marco Polo	2001	EI Paso	Gulf of Mexico	1311	Pile foundation	MOSES
Magnolia	2004	Conoco	Gulf of Mexico	1423	Pile foundation	ETLP
Kizomba B	2006	ExxonMobil	West Africa	1180	Pile foundation	ETLP
Oveng	2006	AHEGI	Gulf of Mexico	274	Pile foundation	MOSES
Okume/Ebano	2006	AHEGI	Gulf of Mexico	503	Pile foundation	MOSES
Neptune	2007	BHP Billiton	Gulf of Mexico	1311	Pile foundation	SeaStar
Shenzi	2009	BHP Billiton	Gulf of Mexico	1333	Pile foundation	MOSES
Bigfoot	2011	Chevron	Gulf of Mexico	1615	Pile foundation	ETLP

Tension-Leg platform,
Fig. 3 TLP platform (Papa Terra P-61)



Tension-Leg platform,**Fig. 4** TLP structure.

(Picture from <http://www.globalsecurity.org/>)



on the shape of the upper body. In order to ensure the strength, some columns are also provided with cross braces and diagonal braces. The main function of the column is to provide buoyancy to the structure, to ensure that the platform has sufficient stability to support the upper deck structure.

3. Pontoon: The pontoon can be divided into three types according to the structure type: integral type, combined type, and sinking type. The main function of the pontoon is to provide buoyancy for the platform, together with the column to ensure the stability and floating state of the platform.
4. Tension leg: The tension leg connects the platform body and the anchoring system of the seabed, which ensures the relative fixation of the platform structure position, controls the motion response of the platform, and ensures the platform has good motion performance. The tension leg system consists of a plurality of sets of tensioned steel cables. The number of sets is related to the shape of the upper body of the platform. Each set of cables is composed

of a plurality of tendons, and the tension generated therein is balanced with the remaining buoyancy of the platform. The tension leg system not only controls the relative position of the platform and the wellhead but also plays a decisive role in structural safety.

5. Anchoring foundation: The anchoring foundation is an important part of the TLP platform, which plays a fixed platform and precise positioning. There are two types of anchoring foundation which are gravity foundation and pile foundation. In addition, there is some other type of anchoring foundation, for instance, the TLR of the Snorre oil field in the North Sea has adopted the concrete suction anchor fixing system.

Tendon System

Introduction to the Tendon System

The tension leg platform (TLP) is a typical compliant platform that is connected to the sea floor by several tendons. The tendon system is an important part of the tension leg platform (TLP) in order

to guarantee the platform is securely anchored to the seabed foundation. The tendon system is always in tension and its axial rigidity is large, which limits the heave, pitch, and roll motions of the platform to a small extent. Its horizontal compliance limits the platform’s sway and roll to an acceptable range of operation and ensures that the platform body in harsh waves is still capable of uninterrupted production operations under environmental loads such as currents and winds. The tendon system consists of multiple tendons and load measurement systems, inspection, or monitoring devices. A single tendon consists of three main parts: a platform interface, a subsea interface, and a body that connects the two interfaces. Each part can contain multiple components, and the form of each component varies depending on

the specific design. Figure 5 shows the tendon structure.

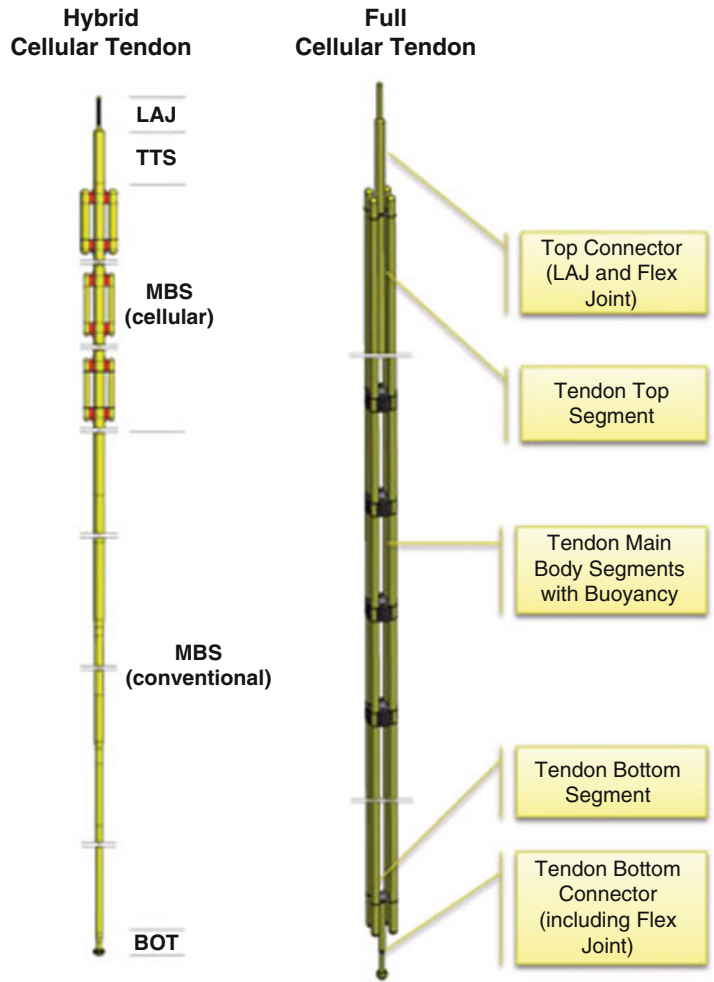
Design of the Tendon System

The tendon system design should ensure that the tendon system meets operational, installation, material, inspection, stress, and fatigue requirements. The tendons designed can be permanent or removable. Additionally, regardless of whether the tendons are permanent or not, the tendon system should be designed to have the potential to remove a tendon.

The selection process of tendons in the tendon system is as follows:

- 1. Calculate the displacement of the platform at this draft according to the main scale of the

Tension-Leg platform,
Fig. 5 Tendon structure.
(Picture from <http://www.offshore-mag.com/>)



tension leg platform and the corresponding draught information.

2. According to the calculated displacement of the platform, determine the number of tendons and the tension coefficient (tension coefficient = total pretension/platform displacement, for example, 30%) and then calculate the pretension required for each tendon.
3. The selection of pipes with a variety of outer diameters can be made according to classification society rules, for example, API SPEC 5L "Pipeline Steel Pipe Specification" (43rd Edition), to calculate the weight of the pipe under different wall thicknesses and select the pipe that meets the demand of $0.8 \leq \text{floating weight ratio} \leq 1.0$.
4. The selection of an appropriate outer diameter and wall thicknesses of the tendon must also be made according to classification society rules, for example, API RP 2T "Recommended Practices for Planning, Design and Construction of Tension Leg Platforms" (3rd Edition) and API RP 2RD "Recommended Practice for Design of Floating Production Systems and TLP Risers" (1st Edition).

Tendon System Installation

In the installation process of the tension leg platform, the tendons are firstly connected with the seabed. When the connection between all the tendons and the seabed is completed, the main body of the tension leg platform is transported to the working sea area to connect with the tendons. After the tension leg platform tendons are connected with the seabed pile foundation, the tendon support buoy (TSB) is required to provide sufficient buoyancy for the tendons to ensure that the tendons do not tip over before the platform reaches the working waters. After the ribs are connected to the main body of the platform, the TSB is released. If the buoyancy provided by the pontoon is insufficient or faulty, the ribs will collide with each other to invalidate the ribs or even overturn the entire tendon, so that the installation of the tension leg platform cannot continue (Fig. 6).

Influence of the Tendon System on the Hydrodynamic Property of the Platform

The nonlinear recovery stiffness provided by tendon system plays a very important role in dynamic responses of TLP platform. The six-degree-of-freedom motion of TLP can be easily divided into two categories, one is mainly controlled by the mooring system stiffness, including the roll, heave, and pitch motions in the vertical plane; the other is mainly determined by the geometric form of the structure and the horizontal component of the mooring tension, including the surge, sway, and yaw motions in the horizontal plane. Due to the large stiffness generated by buoyancy and gravity difference, the natural period of motion on the vertical plane of TLP is relatively small, and the natural period of motion on the horizontal plane is generally large. The natural period of the heave motion is generally within 2–4 s. The horizontal system stiffness is only the small angle positive component of the pretension. The natural period of the longitudinal and vertical motion is generally within 100–200 s. The unique structural form of TLP largely avoids a large displacement caused by the wave excitation load resonance.

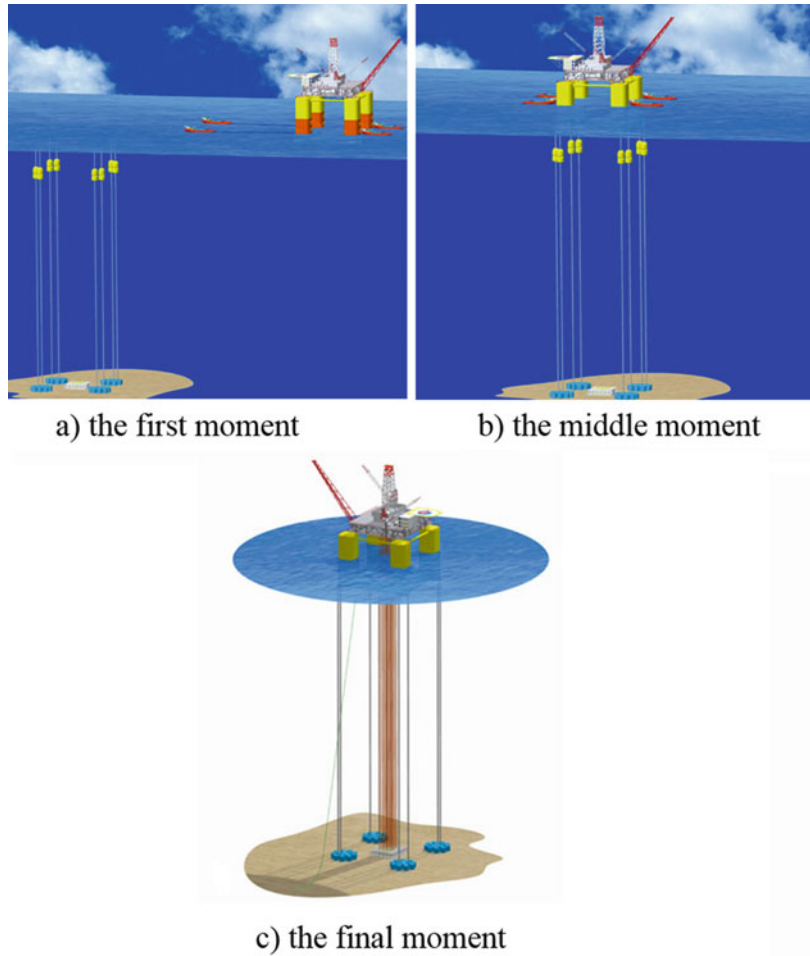
The tendon system makes the natural periods of TLP's motion far away from the resonance periods zone of the wave spectrum. Under the effective constraint of tension tendons, the tension leg platform can avoid the resonance frequency of wave energy concentration, which makes the platform significantly improved in longitudinal, roll, and heave performance. It greatly improves the application range, comfort, and safety of the platform. The increase of the pretension can effectively reduce the motion response of the platform and the angular displacement response of the tendons. While maintaining the draught, the increase in the number of tendons can simultaneously increase the horizontal flexibility and vertical rigidity of the platform movement.

Mini TLP

Mini TLP appeared in the early 1990s. It inherits the advantages of traditional TLP's excellent hydrodynamic performance and good economic

Tension-Leg platform,

Fig. 6 Installation process of the tendon. (Picture from <http://www.cotecinc.com/>)

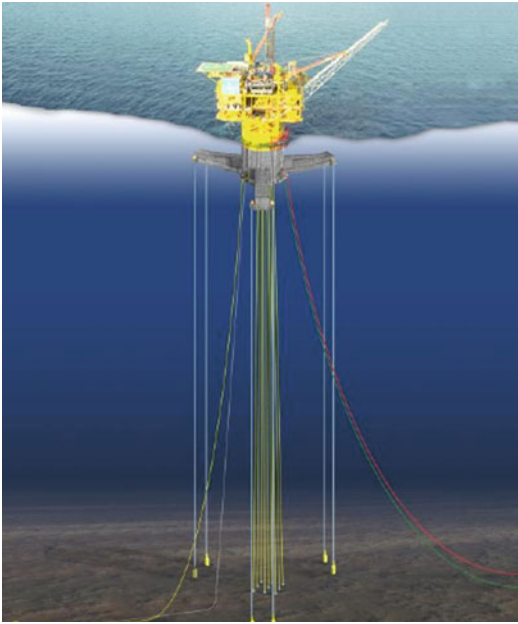


benefits and optimizes the structural form, which makes it low in cost, small in size, flexible, and environmentally friendly. The utility model has the advantages of small hydrodynamic load and stable operation in a deepwater environment. Therefore, it is more suitable for deep-sea operation, harsh marine environment, and small and medium oil field development in deep-sea area, due to its reduction of construction cost and oil field development. The Mini TLPs currently in production practice are mainly divided into two categories, namely, SeaStar TLP and MOSES TLP.

SeaStar TLP

The main body of the platform breaks the traditional type of T-concave 3-column or 4-column

structure, using a very unique single-column design. This cylindrical structure is called the central column. The central column passes through the horizontal plane. The upper end of the column supports the platform deck. At the lower end of the column, the three horizontal cross-section buoys are fixed by internal horizontal and diagonal pull-ups. Each float extends outward forming the cantilever beam structure. The angle between each other on the horizontal plane is 120° , forming a radial shape, and the end cross-section of the pontoon gradually reduces. The three buoys provide buoyancy to the platform body and are connected to the tension leg system at the outer end. The central column includes a casing wall, a moon pool, or a passage. The interior space of the platform is divided into a series of



Tension-Leg platform, Fig. 7 SeaStar TLP

watertight spaces. The six tension legs extend vertically from the joint at the end of the pontoon to the submarine pile foundation. The buoyancy of the pontoon pulls the tension leg, and the buoyancy of the center column supports the deck load to ensure the stability of the platform against subversion. Figure 7 shows the basic composition of the SeaStar Mini TLP.

The SeaStar TLP platform inherits the superior motion characteristics of the traditional TLP platform, with small fluctuations in heave, roll, and pitch. The single cylinder design eliminates crushing and pickup loads in traditional TLP platforms, and the deck design is simplified and modular. The small cylindrical structure reduces the influence of the water line surface dynamics, so that the motion sensitivity problem of the upper equipment is reduced. The small cylindrical structure also saves construction and installation costs, shortens the construction period, and is very suitable for developing large marginal oil fields. However, the SeaStar TLP platform also has some shortcomings. The single cylinder design makes its towage ability or free-floating stability very poor. It is also sensitive to the quality of the superstructure, which makes the superstructure

very limited. When the main body and the superstructure are installed, suspension assistance is required. The entire system is rigid and has extremely high requirements for tension legs.

MOSES TLP

Deepwater tension leg platforms (TLP) are normally unsuitable for small- or medium-sized deepwater fields because of their inherently high Capex costs. Thus, a new, small low-cost TLP design was developed that can be delivered in relatively short timescales and be provided to operators on a lease basis. The minimum offshore surface equipment structure (MOSES) was designed for small fields located in moderate/deepwater depths (Wybro et al. 1995).

The main body of the MOSES TLP still follows the design of the square column structure of the traditional TLP. The platform buoyancy is mainly provided by a floating cabin located in the platform base. The platform base is below the water surface and has a special shape. The center of the base is a square body, and each edge extends diagonally to form a cantilever beam structure. The longitudinal section of the beam is triangular, and the tension leg system is attached to the edge of the four cantilever beams. The column is integrated with the base and is located on the four corners of the top surface of the base. At the same time, the four columns arranged centrally have a certain shadowing effect, and to some extent, the environmental load on the platform is reduced. Moreover, the riser system is connected along the outside of the platform to an open space that is installed at one end of the platform away from the center. This design reduces the possibility of collisions in the riser system. Figure 8 shows a MOSES TLP platform.

MOSES TLP continues to use the traditional four-corner platform structure design to provide greater support for the upper body of the platform and improve the stress on the deck, thus reducing the construction cost of the upper body. The column is thinner than the traditional TLP platform, and the force area at the waterline surface is small, which reduces the wave load on the platform. The modular design of the main body saves a lot of trouble in the construction process and can be manufactured using the standard production line



Tension-Leg platform, Fig. 8 MOSES TLP

of the shipyard. At the same time, the special design of the platform body also greatly reduces the motion response of the platform in the “fatigue zone,” which can reduce the pretension in the tension leg system of MOSES TLP, greatly simplifying the design of the tension leg system. The tension leg system of the platform is manufactured using standardized components, reducing manufacturing costs. And for the first time, a passive spring riser tensioning system is used, which replaces the traditional hydraulic and pneumatic tensioning system, and its safety and stability are greatly improved. In addition, the cost of the whole system is lower, the structure is simpler, and the influence from the environmental load is reduced. In addition, MOSES TLP is simple in structure and flexible in installation. It can be applied to the installation of a variety of deep-sea equipment.

Cross-References

- [Concrete Platform](#)
- [Mooring System](#)

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Test

- [Experimental Investigation of Offshore Renewable Energy](#)

2,3,5,6-Tetrachloro-4-methylsulfonyl (TCMS)

- [Net Structures: Biofouling and Antifouling](#)

The Arctic Pole Pipeline

- [Arctic Pipeline](#)

The Global Maritime Distress and Safety System (GMDSS)

- [Polar Offshore Engineering](#)

The Hamburg Ship Model Basin (HSVA)

- [Ice Tank Test](#)

The Institute for Ocean Technology (IOT)

- [Maneuverability of Polar Vessel](#)

The Lumped Mass Method

► [Dynamic Analysis Method](#)

The North Pole Pipeline

► [Arctic Pipeline](#)

The Rod Theory

► [Dynamic Analysis Method](#)

Thermal Insulation

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Synonyms

[Heat preservation](#); [Insulation work](#); [Keep warm](#);
[Preserve heat](#)

Definition

Thermal insulation refers to laying thermal insulation layer on the surface of heat supply pipeline and its accessories to reduce the invalid heat loss of heat medium in the process of transportation and maintain certain parameters of heat medium to meet the needs of users.

Background

As oil and gas exploration enters the deep sea mode, the low-temperature and high-pressure environment in the deep sea is more conducive to the formation of hydrates and wax crystals,

thereby blocking the submarine pipelines. Once the hydrates or wax crystals are formed in the pipelines, the transport capacity of the pipelines is affected. It will cause major engineering accidents. Insulation measures for submarine pipelines to prevent the formation of hydrates and wax crystals have also been continuously studied since oil and gas exploration.

The type of insulation can be divided into passive insulation and active heating.

Passive Insulation

Passive insulation includes dry insulation and wet insulation. Passive insulation is achieved by using the properties of the insulation material and the pipe structure to achieve its insulation effect. For submarine pipelines in deep water, the way to maintain insulation by virtue of the properties of the materials is extremely demanding, including materials that need to meet high strength, high hardness, high hydrostatic pressure, and creep resistance, and is quite expensive. In the process of passive insulation, chemical reagent injection methods, usually methanol and ethylene glycol, are often used to inhibit hydrate formation and wax deposition, but the injection of chemical reagents requires a rigorous calculation, because the injection of chemical reagents tends to reduce the amount of flow rate. For passive insulation, the most deadly drawback is that in the process of shutting down and opening the well, it will promote the formation of hydrate and wax crystals due to passive insulation.

At present, most of them adopt measures such as heating on the offshore platform and combining double-layer insulation pipes and chemical injection reagents and pigging to ensure the safety of conventional and short-distance oil pipelines. This requires not only a large electric heating device but also for heavy oil or long-distance oil and gas transportation. The abovementioned deficiencies can be well solved by using submarine pipeline active heating technology. Especially in the deep water long-distance return system or in the cold Arctic region, the advantages of active heating technology are more prominent.

Active Heating

Submarine pipeline active heating technology can be divided into hot fluid and electricity according to the different heat sources used, as shown in Fig. 1. Active heating technology has been applied to subsea pipelines in multiple oil and gas fields.

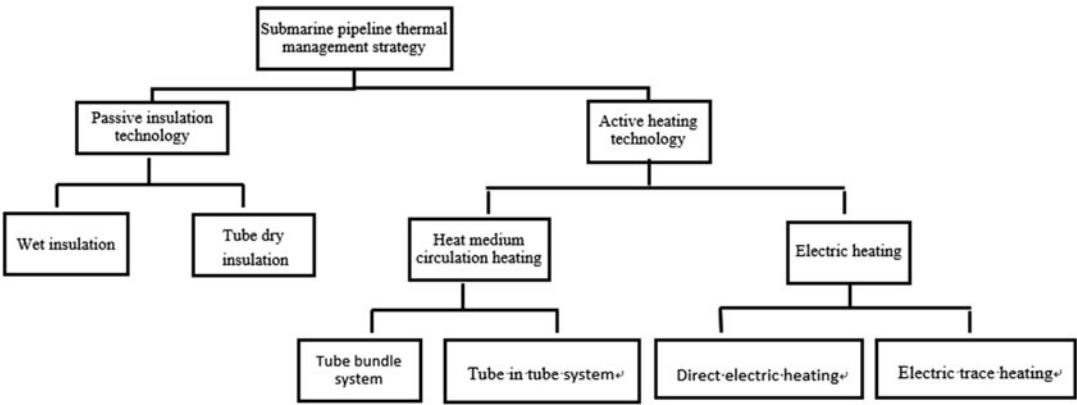
Hot Fluid Heating

Thermal fluid heating is a traditional method of heat tracing. The method is to ensure the normal transport temperature of the medium in the oil and gas pipeline by the heat exchange between the hot

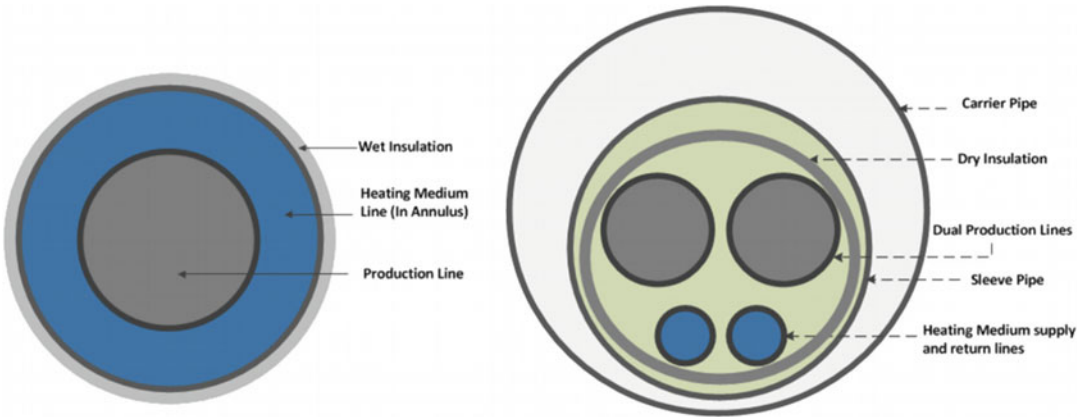
fluid and the medium in the oil and gas pipeline and the external thermal insulation layer.

Depending on the type of pipeline structure, the hot fluid can flow in the tube loop cavity or the bundle tube heat pipe, which is called direct heat tracing and indirect heat tracing, respectively, as shown in Fig. 2. Commonly used hot fluids are sea water, production water, aqueous glycol solution, steam, and the like. The heat fluid heat source may be a heating device such as a boiler or a waste heat recovery system.

The hot fluid is directly heated, and the insulation effect is poor due to the wet insulation layer. For the indirect heat tracing of the hot fluid, the



Thermal Insulation, Fig. 1 Submarine pipeline thermal insulation technology



Thermal Insulation, Fig. 2 Outline of direct and indirect hot fluid circulations systems (McDermott and Sathananthan 2014)

thermal insulation effect is better because the dry heat preservation is adopted. However, current cluster pipes require specific construction sites, and the installation method and application depth are limited. Thermal fluid heating is a good choice when there is sufficient residual heat available on the platform.

At present, the Britannia oil and gas field in the North Sea, the King oil and gas field in the Gulf of Mexico, and the Puteri oil and gas field in the Malaysian waters are all directly heated by hot fluid; the Gullfaks South, Asgard, Skene, and Bacchus oil and gas fields in the North Sea are all using thermal fluids.

Direct Electric Heating

Direct electric heating is to directly pass current on the pipe wall, and the pipe is electrically conductive as a conductor, which generates heat due to the alternating current resistance of the pipe material, as shown in Fig. 3. The heat generated

by direct electric heating is related to the current and the AC resistance.

According to the different types of pipeline structure, direct electric heating can be divided into wet direct electric heating and direct tube electric heating. Wet direct electric heating is for single-layer pipes, and the cables are connected to the two ends of the pipe. The direct electric heating of the tube in the tube is for the double-layer tube structure, and the cables are respectively connected with the inner and outer tubes and can be divided into two types: end feeding and intermediate feeding according to different wiring positions.

The direct electric heating adopts the wet heat preservation, the heat preservation effect is poor, and the dry heat preservation has better heat preservation effect. Since direct electric heating is directly connected to alternating current on the pipe wall, there is a risk of AC corrosion, special design of anticorrosion is required, and regular



Thermal Insulation, Fig. 3 Schematic diagram of direct electric heating pipeline (<https://www.techreleased.com/wartsila-to-supply-direct-electric-heating-system-for-african-offshore-oilfield/>)

inspection and maintenance are carried out. At present, the direct electric heating type has low electrical efficiency and has not yet been validated for long-term use.

At present, Alve, Norne, Asgard, Huldra, Kristin, Urd, Ormen Lange, Tyrihans, Morvin, Skarv, and Skuld oil and gas fields in the North Sea and Olowi oil and gas fields in West Africa use wet direct electric heating. The Serrano and Oregano oil and gas fields in the Gulf of Mexico use the direct electric heating (end feed) type of the tube, and the Habanero and Na Kika oil and gas fields use the direct electric heating (intermediate feed) type of the tube.

Electric Heating

The working principle of electric heat tracing is to dissipate a certain amount of heat through the heat tracing medium and supplement the loss of the heat tracing pipe through direct or indirect heat exchange to achieve the work requirements of heating, heat preservation, or anti-freezing. The electric heat tracing is to heat the heating cable/heat tracing tube by the electric and magnetic effects of the cable/heat tracing tube, thereby transferring heat to the heat tracing pipe, supplementing the heat lost by the tracing fluid in the process flow, so as to maintain the most

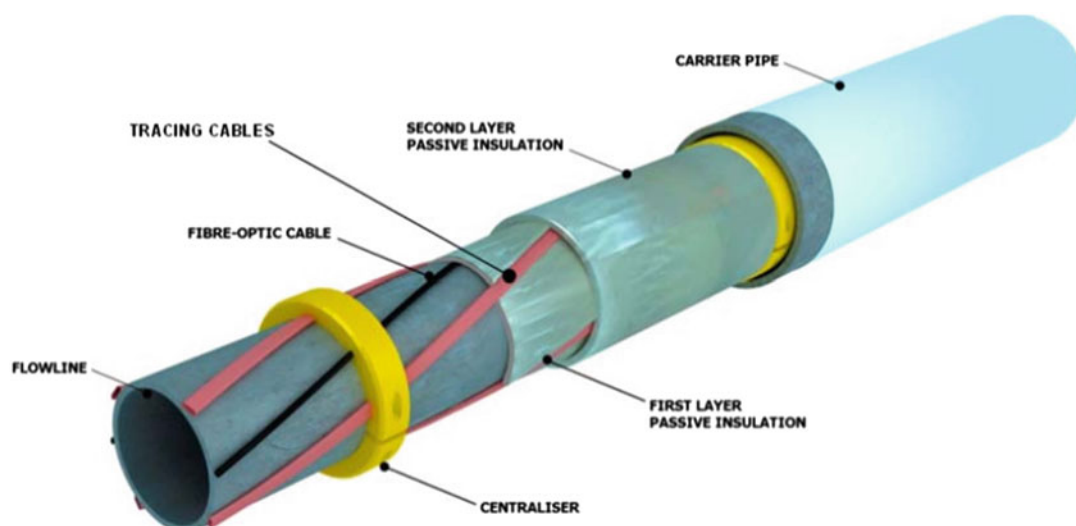
reasonable process temperature of the flowing medium.

According to the different heat generation mechanism of electric heat tracing, it can be divided into electric heating cable heating, skin effect electric heat tracing, and electromagnetic induction electric heating. At present, the first two types of heating type already have subsea pipeline engineering applications, and the electromagnetic induction electric heating has no submarine pipeline engineering application and is still in the development stage.

Electric Heating Cable

Electric heating cable heating usually wraps the heating cable around the inner tube of the double tube, as shown in Fig. 4. Set a certain number of heat tracing cables based on the required heat, and set up a backup heating cable.

Electric heating cable heating generally adopts dry heat preservation, which has better heat preservation effect and higher thermal efficiency. The electric heating cable is relatively uniform in longitudinal and circumferential heating of the pipeline and has good operability. It can accurately control and adjust the heating energy and can monitor the heating temperature through the optical fiber. However, when the electric heating cable



Thermal Insulation, Fig. 4 Schematic diagram of electric heating cable (Candelier et al. 2015)

is laid by the coiling method, the diameter of the pipeline is limited by the capacity of the pipe laying vessel.

In addition, by combining the heat tracing cable with the flexible bundle pipe, an electric heat tracing flexible bundle pipe that can be applied to the riser pipe segment is formed, as shown in FIG. It can be used for active heat tracing of dynamic risers. However, the insulation effect is poor, similar to wet insulation (Fig. 5).

At present, the Islay oil and gas field in the North Sea has adopted electric heating cable heating; the Dalia oil and gas field in the West African waters and the Para Terra oil and gas field in the Brazilian waters have adopted flexible cluster tube electric heating cable.

Skin Effect Electric Heating

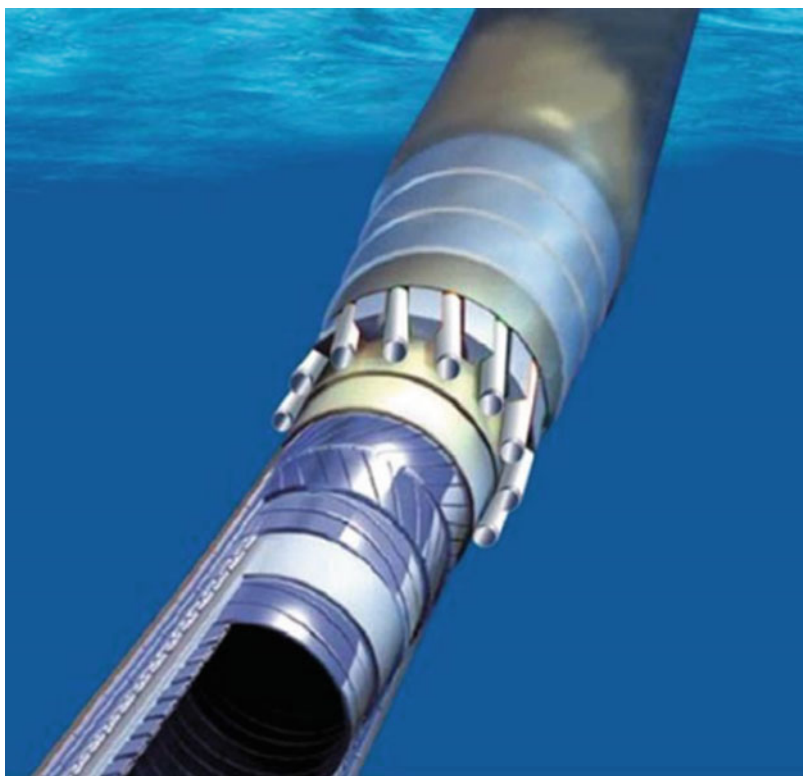
Pipe skin effect electric heat tracing technology is a new metal pipe heating method, referred to as SECT (skin electric current tracing) method. The skin effect electric heat tracing system is mainly composed of a process pipe, a heat tracing pipe

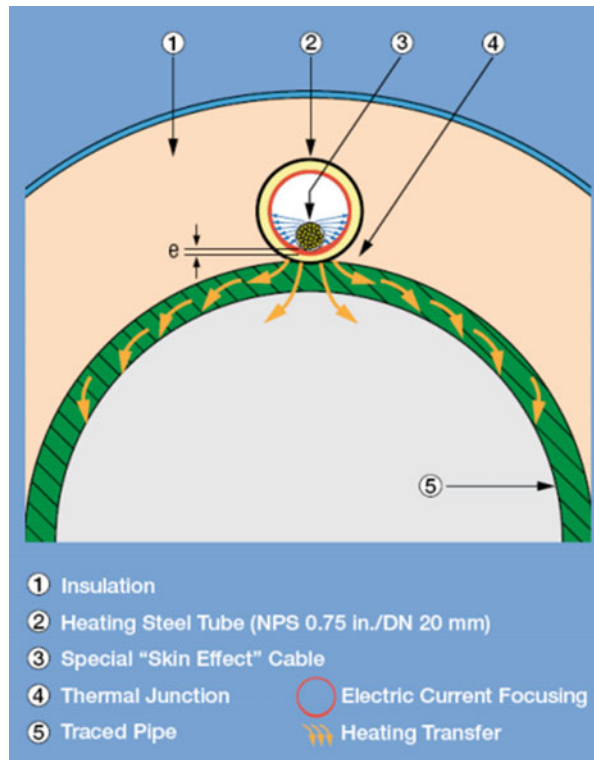
and a heat-resistant skin cable, an insulation layer, and a protective casing. The heat tracing tube is a ferromagnetic steel tube, and the heat-resistant skin cable is placed in the heat tracing tube, and the outer layer is an insulating layer and a protective outer casing. The structure is shown in Fig. 6.

According to the tube diameter and the heating temperature, the skin effect heating is divided into single-tube, double-tube, and triple-tube heating. The heat-resistant skinning cable is placed in the heat tracing tube, and the core wire of the cable is connected to the heat tracing tube at the end of the heat tracing tube. At the power supply end, the heat pipe and the skin cable are respectively connected to the AC power source. When the AC current flows through the heat-resistant skin cable and the heat-treating tube, due to the ferromagnetic characteristics and the proximity effect of the heat-treating tube, the skin effect phenomenon forces the current only in the inner wall of the heat tracing tube flows to generate thermal energy. The external surface of the heat tracing

Thermal Insulation,

Fig. 5 Schematic diagram of electric heat tracing flexible bundle pipe (Baker 2014)



Thermal Insulation,**Fig. 6** Skin effect electric heat tracing structure type (Ansart et al. 2014)

tube has no current, so the heat tracing tube can be safely grounded.

In 2008, the Weinan oil and gas field applied the skin effect electric heat tracing to the submarine pipeline. After the skin effect electric heating system was put into use, the pipe outlet temperature increased by about 5 °C (Denniel et al. 2004).

Electromagnetic Induction Heating

The basic structure of the electromagnetic induction electric heat tracing system is similar to the skin effect electric heating tracing, and the difference lies in the way of the remote connection. For electromagnetic induction heating, the three heat cables at the far end are connected to each other and grounded to form a neutral point. An alternating magnetic field is formed when the alternating current flows in the cable located at the center of the heat tracing tube, causing an induced circulating eddy current in the heat tracing wall. This circulating eddy is eventually dissipated as heat due to the alternating current resistance and hysteresis effect of the heat tracing tube. This heat is

transferred to the main line through the heat tracing tube to heat the fluid.

At present, no case of electromagnetic induction electric heating is applied to submarine pipelines.

A general comparison table of active heating technologies is presented here below (Fig. 7).

After comprehensive comparison and analysis, it is suggested that the following items should be considered in the selection and design of heat tracing modes for submarine pipelines:

1. The heat preservation effect of the insulation layer is directly related to the heat tracing efficiency. In general, the effect of wet insulation is obviously lower than that of dry insulation. Generally, when designing the heat tracing system, the influence of the insulation system on the heat conduction of the whole submarine pipeline should be considered comprehensively.
2. The heat tracing efficiency of hot fluid tracing is low, and the corrosion problem caused by hot fluid to heat tracing pipeline should be considered. When there is enough waste heat

		Hot fluid heating	Wet direct electric heating	Direct tube electric heating	Electric heating cable	Skin effect electric heating
DESIGN	Pipe type	Pipe in Pipe	Wet Insulated Pipe	Pipe in Pipe	Pipe in Pipe	Pipe in Pipe
	Weight	Heavy pipe	Light pipe	Heavy pipe	Heavy pipe	Heavy pipe
	U-Value (W/m ² K)	Wet:3-6 Dry:0.6-6	Wet:3-6	Dry:0.6-2	Dry:0.6-2	Dry:0.6-6
HEATING EFFICIENCY	Heating Efficiency	40-60%	50-70%	95-100%	90-100%	90-100%
	Electrical Efficiency	N.A.	30-60%	50-70%	90%	Close to 100%
	Max single heated length	Limited by pressure drop in water circulation system	Longest length installed to date: 28km	Longest length installed to date: 17km	20-50km due to cables rating limitations	Single power supply can heat tracing for 20 km in one direction
CONSTRUCTION & INSTALLATION	Installation Method	S&J Lay only	S&J Lay Reel lay	S&J Lay Reel lay	Reel lay	S&J Lay Reel lay
	Max Water Deep	High for small diameter pipes	0-400m	High for small diameter pipes	Tracing cables qualified for 1500m WD max	High for small diameter pipes
	Max Diameter	About 12" inner pipe max	6" to 30" Corresponding to S-Lay Capabilities	About 12" inner pipe max	About 12" inner pipe max	About 12" inner pipe max
OPERABILITY	Continuous Heating	Yes	Yes in Shallow Water. Not qualified in Deep Water	Not qualified	Qualified	Yes
	Precise power adjustment	No	No	No	Yes	Yes
	Temperature Monitoring	Only at the inlet and outlet	Only DTS monitoring in the power cable	Fiber Optic Measures	Fiber Optic Measures	Fiber Optic Measures
COST	Engineering and process	Cheap Wet Insulation. Expensive Dual pipe structure	High rating piggy back cable + 2 Dynamic umbilical	Expensive Dual Pipe structure + 2 Dynamic Umbilical	Expensive dual pipe structure + High performances Insulation	Expensive dual pipe structure

Thermal Insulation, Fig. 7 Active heating technologies – comparison table (Esaklul et al. 2003)

available on the platform, hot fluid tracing has great advantages (Nebell et al. 2013).

3. The electric efficiency of direct electric heating is low, so it is necessary to design special

anticorrosion and carry out regular inspection and maintenance. At present, most of the direct electric heating pipelines are short-term use, rarely long-term use.

4. It is suitable for long-term use because of its small temperature gradient and long thermal stability time. The heat (electric power) required by electric heat tracing is obviously lower than that of direct electric heating.

Prospect

Active heating technology and passive insulation technology can make subsea pipeline operation more flexible and reduce the sensitivity of underwater production systems to transient operating conditions, especially under deep water and high hydrostatic pressure conditions.

Electrically trace heated PiP technology is the next evolution of active heating systems that combine high-performance thermal insulation with a low power design making it a very attractive option in comparison to conventional DEH systems. In the future it is expected that the major industry drivers will be toward the development of smart oil fields of the future which will require continual development and innovation of active heating systems technology.

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Thermoplastic Hoses

- [Umbilical Cable](#)

2-Thiocyanomethyl-thiobenzo-Thiazole (TCMTB)

- [Net Structures: Biofouling and Antifouling](#)

3DOF, Three Degrees of Freedom

- [Numerical Simulation of Ice-Going Ships](#)

Three-Dimensional (3D)

- [Ship Hull Lofting and Marking-Off](#)

Three-Dimensional Hydroelastic Analysis Method

- [Hydroelasticity Theory](#)

Thruster-Assisted Mooring

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Synonyms

[Dynamic positioning \(DP\)](#); [Mooring](#); [Position Mooring \(PM\)](#); [Ship motion control](#); [Vessel positioning](#)

Definition

A **thruster-assisted mooring (TAM)** system has different control functions for station keeping and motion damping for a moored offshore vessel with assist from thrusters. It consists of a conventional passive mooring system and a so-called dynamic positioning (DP) system. Actually, it is not the simple superposition of the two systems. The thrusters are used to provide damping and some restoring to the vessel motion and withstand environment loads. In addition, if line breakage occurs, thrusters need to produce more force to compensate. The mooring system absorbs the main loads to keep the vessel in position.

Scientific Fundamentals

Development of Thruster-Assisted Mooring

When offshore operations are discussed, it is commonly thought of as the various tasks related to oil and gas production and exploration at sea. Oil production is probably one of the most technologically advanced and challenging operations on the ocean. According to the type of operation, a structure might need to stay at the working spot on the ocean for everything from shorter day missions and a few months of exploration to a several years-long production period.

A common solution for long-lasting operations on relatively small water depths is to build a solid structure which can stand firmly on the seafloor. It

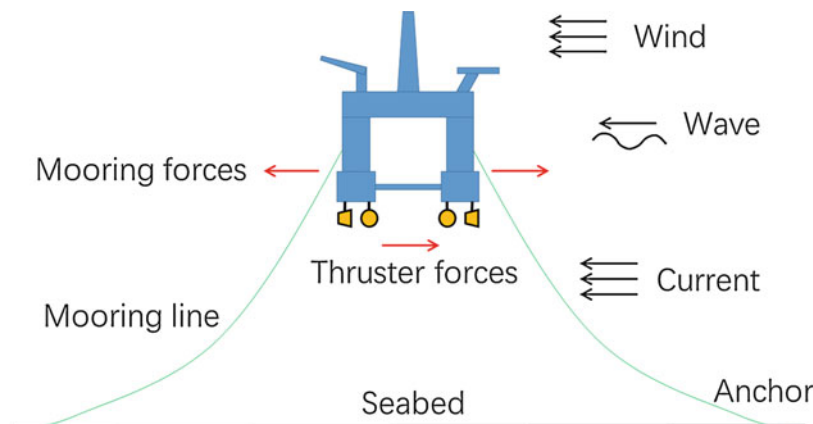
can keep the platform steady without any energy consumption, but it is less flexible and requires more time for installation to get in position.

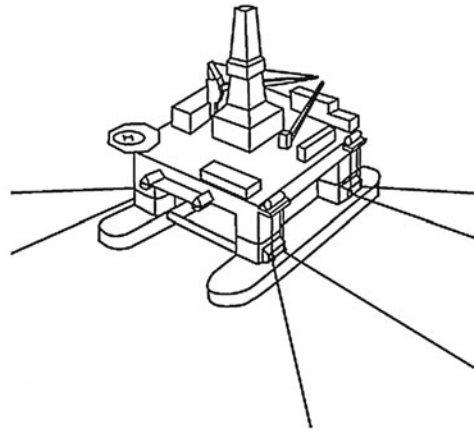
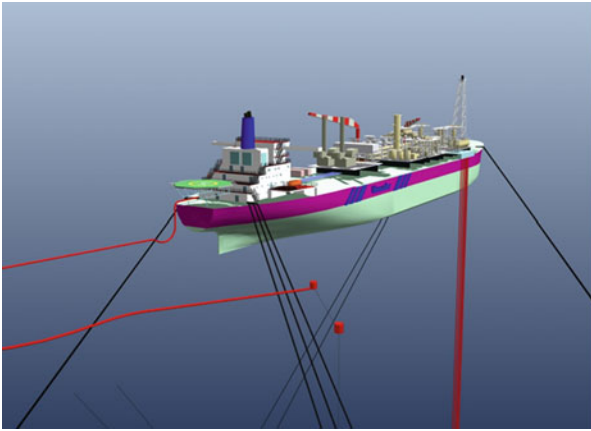
So it is necessary to use a floating structure in deeper water. There is also an advantage in general that it requires less amount of effort needed for installation. For some shorter-lasting operations, the easy-come-easy-go floaters are more cost-effective. These can be kept in working position by the use of anchors and mooring lines. The drawback of this solution is the vulnerability when the waves and other periodical loads act on the floating structure, leading to potentially large oscillations due to the spring effect from the mooring lines and the relatively low damping of the system.

By introducing a control system and thrusters, the floating structure can be kept in target position automatically, while active damping can be used to prevent large horizontal offsets. It led to the introduction of dynamic positioning (DP), making it possible to stay in place without any time-consuming external installations and with no need of seafloor installations. Figure 1 shows the basic conception of TAM system on a semisubmersible platform.

In recent years, thruster-assisted mooring (TAM) systems have become more commercially available, especially in the oil industry. For semisubmersibles, it is suitable to use spread moored systems. For FPSOs, both turret mooring systems and spread mooring systems are used according to the working conditions.

Thruster-Assisted Mooring, Fig. 1 Basic conception of TAM system





Thruster-Assisted Mooring, Fig. 2 Spread mooring systems. (<https://www.bluewater.com>, <http://www.dredgingengineering.com>)

Basic Technology and Theory of Thruster-Assisted Mooring

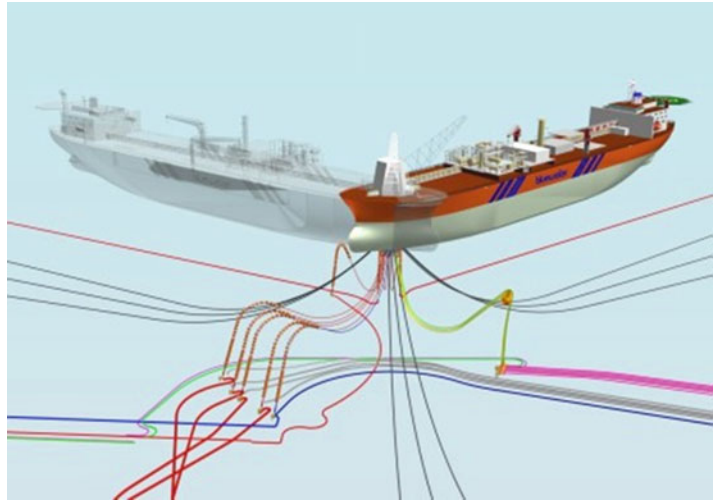
Mooring system is one of the key systems in TAM system. A mooring system refers to a set of mooring lines connecting a floating vessel to the seabed to keep it in position. Traditionally, anchoring of a ship uses one mooring line, but it's not an optimal scheme because the ship is likely to move around the anchor point and oscillate in yaw under the wind and current loads. Researcher proposed a better solution, which is spread mooring. Several mooring lines are connected to different locations of the floating vessel. It can make the ship stay in one direction during the operation time, but on the other hand, it is not very flexible when it comes to the change of vessel yaw angle. Figure 2 illustrates the spread mooring systems on FPSO and semi.

A good solution for mooring systems is turret mooring. All mooring lines connect to what is called a turret, which is located inside or outside the ship. Positioning the turret within the front half of the hull is most commonly used. The mooring lines are then spread into different directions. By this method similar restoring capability in surge and sway can be provided. The turret is usually free to rotate, making the vessel able to move in yaw easily. Being able to rotate the vessel to minimize environmental loads, referred to as weathervaning, is therefore an advantage (Fossen 2011). Figure 3 shows the weathervaning around the turret mooring system.

The first use of mooring systems for position keeping in offshore operations was in 1953 (Fay 1990). But when the development goes into deeper water, the drawbacks become obvious. Then the dynamic positioning system was developed. This idea means the passive mooring system could be replaced by thrusters which can dynamically compensate the environmental loads. However, since DP system requires a large amount of energy during a long working period, the conventional mooring systems still has some economic advantages. So the idea behind thruster-assisted mooring systems is to combine the advantages from both fields, where the mooring system is to withstand the mean environmental loads, while DP system contributes to improve the system dynamic performance that the mooring system is not capable of. For more on automatic thruster-assisted mooring systems, see, for instance, Strand et al. (1998).

In 1961 the vessel CUSS 1 maintained position in an accuracy of 180 m radius with manual controlled thrusters. Later in the same year, a ship named Eureka was equipped with the first computer-based dynamic positioning system. From then on, there has been a steady development of DP systems in all fields related. Kalman filtering and optimal control theory are milestones in the evolution of DP control system. Besides, the researches on hydrodynamics, thrust allocation, and interaction effects all get approaches.

Thruster-Assisted Mooring, Fig. 3 Turret mooring system and weathervaning. (<https://www.bluewater.com/>)



In the 1990s, DP system started to use smarter methods to obtain a better control performance and DP capability. This kind of development was propelled by the demand of a control system which could take both system dynamics and limitations into account at the same time. Model-based design is an advanced method, upon which several functions have been built later. A model-based control system design was presented by Sørensen et al. (1996). The model-based passive nonlinear observer was introduced in Fossen and Strand (1999). The nonlinear passive observer yields global exponential stability. However, the linear Kalman filter theory does not. Strand and Sørensen (2000) proposed a roll and pitch damping controller and proved to damp horizontal motions by exploiting the hydrodynamic couplings between surge and pitch and sway and roll degrees of freedom.

The theory of hybrid control was first implemented in planes and cars. During the course of previous years, hybrid control has also been applied in DP control system. The demand for total system stability, including the switching logic, places special requirements on the DP control system design. Hybrid control is studied in Nguyen and Sørensen (2009b). A hybrid control strategy was proposed by Brodtkorb et al. (2018). This concept allows a structured way to develop a control system with a bank of controllers and observers improving DP performance in stationary

dynamics, changing dynamics including enhancing transient performance, and giving robustness to measurement errors.

Other advanced control methods have also been applied to DP controller design. A Takagi-Sugeno fuzzy dynamic positioning controller was designed for an unmanned marine vehicle in network environments by Wang et al. (2018). Du et al. (2018) proposed a robust adaptive control method for DP ships with unknown model parameters and unknown time-varying disturbances.

The mooring system is a relative low-cost solution. In shallow water and calm weather conditions, mooring systems provide steady station keeping performance for most operations at sea. Dynamic positioning, on the other hand, is a relative high-cost solution compared to passive mooring only. The third option is a merge between mooring and DP. It's the so-called thruster-assisted mooring, position mooring, or DP-assisted mooring. For TAM systems, the mooring lines form a passive elastic system and provide most of the horizontal damping. The thrusters must assist the passive mooring system for environmental loads or operational requirements to obtain an accurate positioning.

Strand et al. (1998) presented the first publication of the combination of thruster assistance and mooring. Position keeping based on adjusting the mooring line lengths using winches was proposed later. Imposing a structural reliability criterion on

riser angles was studied for the floating vessel motion influences the riser dynamic responses. Based on the proposed structural reliability criterion, Berntsen et al. (2006) presented a nonlinear feedback control law by using backstepping techniques. The structural reliability criterion was originally used in the field of construction. Implementation of the algorithms combined with structural reliability consideration could minimize the risk of mooring line failure to a large extent. Thruster-assisted mooring was first merged with hybrid control in Nguyen and Sørensen (2009a). And the hybrid control method is believed to be one of tomorrow's good solutions for both DP and TAM. In Fang et al. (2013), a setpoint chasing method based on mooring line tension was proposed. Reaching the desired setpoint is handled by a control law, e.g., PID.

A DP or TAM system relies on a complex system of different components, as shown in Fig. 4. The key component of the whole system is the control system, which generates thrust command vector given the various input parameters it is provided, as shown in Fig. 5. The most important input information is the vessel position and heading, but in most cases it also receives a variety of other input variables in order to improve control performance, like vessel velocities, wind speed and direction measurements, accelerometer data, and gyro information. The resulting force command is

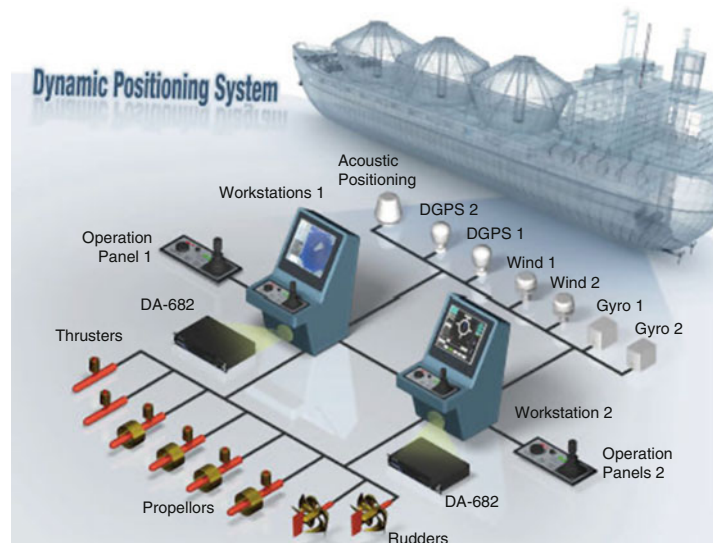
then passed to a thrust allocation unit, converting it into individual thrust force and direction commands to each thruster. Thrusters on the vessel have their own control system, making sure the right power is applied to the thruster it controls. The power system provides power for other systems of the whole one.

In order to achieve a good estimate of vessel motion, usually it is equipped with multiple motion reference systems. These reference systems are then evaluated to find the most accurate motion estimate. The redundancy configuration is depended on the DP class requirement. Different vessel needs different DP class to hold its position, and it doesn't mean good if each vessel is equipped with the highest class of DP system.

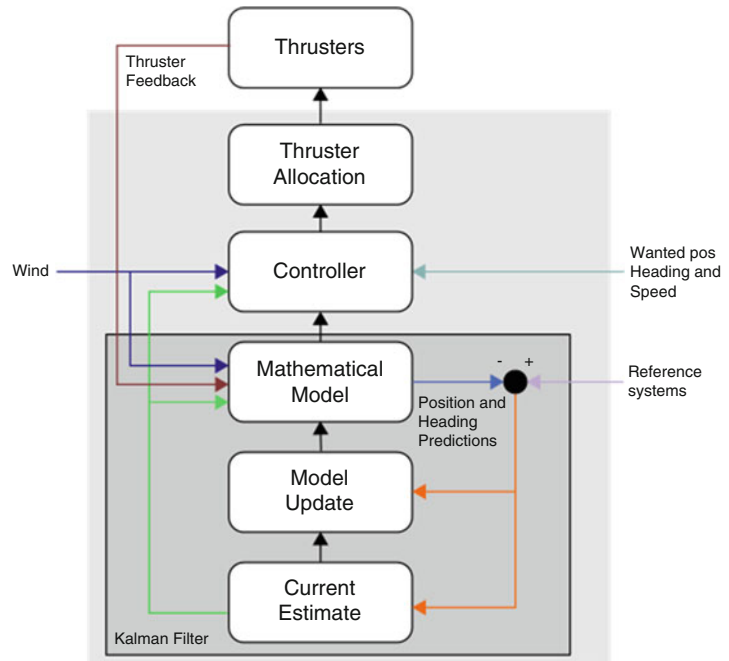
It is necessary to filter the motion signals to obtain a smooth position reference to provide to the control system due to sensor noise and other factors causing inaccuracy. This task is implemented by an observer, which can be, for instance, a Kalman filter or a nonlinear passive observer (Fossen 2011).

The controller can use the accelerometer or wind measurement data for feed forward, meaning it can take advantage of them to predict future position offsets or part of the environmental loads by measuring some of the forces acting on the vessel at the present time. These techniques are however limited by inaccuracy mainly due to

Thruster-Assisted Mooring, Fig. 4 A typical DP system. (<https://www.moxa.com>)



Thruster-Assisted Mooring, Fig. 5 Control system. (https://en.wikipedia.org/wiki/Dynamic_positioning)



sensor disturbances. For instance, the significant local variations of the wind lead to inaccuracy of wind load signal.

In addition to the filtered sensor measurements, the controller is provided with a reference position from the DP operator. This is the desired position for the vessel, meaning the position the controller is supposed to keep the vessel as close as possible to.

For TAM systems the position is determined by the mooring, and the control objectives can be a bit different. For more on marine control systems design, see, for instance, Fossen (2011) and Sørensen (2011).

Control Objectives, Model, and Method

The main objective of a DP system is to keep the floating vessel within a certain radius of a target point. DP is used to prevent failure of risers and other drilling equipment; to stay around the operation spot, for instance, lifting, lowering, and installing; or just to prevent from horizontal position offset when idling. Control objectives are different depending on the kind of positioning requirements. Figure 6 illustrates the forces on a DP vessel.

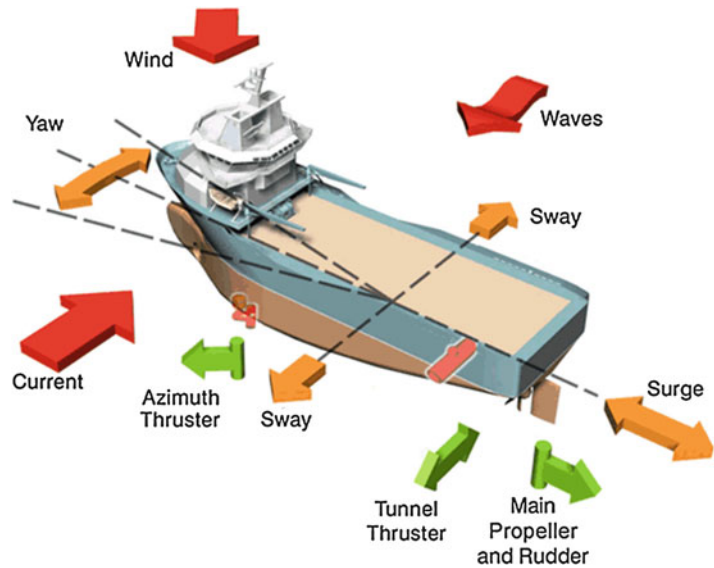
In addition to stabilizing the vessel in position, DP system is desired to provide additional damping forces. These forces could compensate for low-frequency wave loads and oscillations due to wind, current, etc.

For TAM systems, the damping action is a primary part of the control system functionality. Among the whole system, the mooring lines will withstand the mean loads in surge and sway. The elastic behavior of the mooring system is similar to that of a spring. Therefore, it can be simulated with a proportional controller. To control the vessel position, a derivative term is needed, which can be provided by the thrusters activated by the control system.

If the constraint in yaw cannot be provided by the mooring system, the control system will in most cases be used to control the heading angle. It is useful when the offshore operation requires the ship to keep its heading orientation or just to reduce unnecessary accelerations in yaw. In addition to this, an optimal heading angle can also be maintained by the thrusters to compensate environmental loads. Especially for ships that are sensitive to wind and current force direction, there will be a significant difference in resulting loads

Thruster-Assisted**Mooring, Fig. 6**

Forces on a DP vessel. (<https://www.kongsberg.com/>)



when the load is acting in surge or sway. Therefore, it can reduce the mooring line tensions by dynamically chasing the optimal heading for handling the weather.

Another practical feature for a TAM system is mooring line failure detection and compensation. It means that the control system activates thrusters to prevent further damage on the vessel in the case if one mooring line is broken. Mooring line failure can be detected by sensors on the positioning system, and if the control system receives the signal of line breakage, the thrusters can be immediately activated to generate force to cover the broken line with proper thrust and direction. With the interference of thrusters, another line breakage may be prevented, and the operation can be shut down in a safe environment, and everything could be controllable. Line breakage can also be prevented by using the thrusters to produce force to reduce mooring line tensions when the stresses become too high in the case of extreme conditions.

Therefore, a TAM system is mostly used for the control objectives of damping action (surge and sway), heading control, and mooring line failure compensation and prevention.

Commonly there are two kinds of method for modelling a dynamic system in the frame of

control theory, depending on the purpose of the model.

Usually the process plant model is used to describe the system dynamics as accurately as possible. In this model different nonlinear phenomena would be included as well as the disturbances and noises. This model can be used for numerical simulations and analysis of stability and performance. It can also replace model tests of the system (Nguyen and Sorensen 2009a).

Actually, a vessel with mooring system at sea is subjected to different environmental loads. Combining the cable dynamics, the vessel motion itself is complex. In order to study a dynamic system behaves under any condition, a good description of the system is needed. Therefore, the model will be relatively large and complex. There are most likely several nonlinear phenomena that need to be considered at the same time.

A process plant model is therefore not suited for time-domain analytical computations, which means it cannot be used for the design of control system. That is when a control plant model is applied, which is a simplified version of the process plant model. It is less computationally intensive and easier to work with both numerically and analytically.

TAM systems are different from DP systems, whose main purpose is to keep a floating vessel on

a target position by activating the thrusters on the vessel. In calm weather conditions, the thrusters are mainly used to keep the yaw angle of the vessel for a turret moored vessel, whereas the mooring system maintains the vessel position within an allowable region. However, in harsh weather conditions, the thrusters are not only used to maintain vessel orientation but also to compensate environmental loads in order to keep the line tensions of mooring system under safety limits. The mooring system doesn't consume any fuel making it cost-efficient as well.

It can be assumed that a thruster-assisted mooring system consists of a mooring system and a special DP system. The mooring system is a passive system. But the thruster system is active. So many studies on TAM systems are mainly focused on DP control method. It should be pointed out that vessel mass, center of gravity, and velocity always change by time. In addition, the vessel motion is nonlinear considering the loads on the vessel. So, the control system can be regarded as an uncertain nonlinear system. Recently, nonlinear control theories are proposed in order to handle the inherent nonlinear characteristics of the vessel dynamics, for example, the output feedback control, the nonlinear model predictive control, the adaptive fuzzy control, and the adaptive neural network control. A nonlinear control algorithm based on mooring line structural reliability has been presented by Wang et al. (2014) in which the reliability index is used to quantify the failure probability of mooring lines. The reliability index is set as an intrinsic term of the controller function, the mooring system keeps the position of the floating vessel within an allowable region, while thrusters maintain the structural reliability index of mooring lines within allowable limits. The controllers are built according to the backstepping theory. However, backstepping requires repeated differentiation of the virtual control function, which increases the computational cost. This problem can be solved by applying dynamic surface control which uses first-order filtering of the virtual control.

Vessel with TAM system is assumed to be an uncertain nonlinear system. In order to solve the problem, artificial intelligence algorithms can be used, for example, the fuzzy logic method,

artificial neural networks, and fuzzy neural networks. Niu et al. (2017) used the RBF neural networks to make the approximation of the uncertainty in nonlinear pure-feedback switched time-delay systems, and the fuzzy control systems have also been utilized in various nonlinear systems such as stochastic nonstrict-feedback systems, networked control systems, non-Gaussian stochastic distribution control systems, and Takagi-Sugeno fuzzy systems to compensate some of the uncertain nonlinear effects. Chen et al. (2015) presented an adaptive control approach for MIMO robotic systems with unknown control signs and time-varying nonlinearities. The Nussbaum gain technique is used to estimate the unknown actuator directions. However, the difficulty of using this approach is that the Nussbaum gain has been extended from one dimension to multiple dimensions, which makes the controller design and stability analysis more difficult. To solve the problems, Tuo et al. (2018) designed a novel adaptive robust controller for TAM system of the moored FPSO vessel with model uncertainties and unknown control coefficients.

Key Applications

Thruster-assisted mooring systems are applied to anchored FPSOs and semisubmersibles. They can be used to keep vessel position, maintain vessel heading angle, chase the optimal environmental direction, and compensate the dynamic environmental loads or sudden loads due to mooring line failure.

FPSO

An FPSO is usually positioned by mooring lines in deep water. With a turret it can be rotated according to the environment load direction. If the offshore operation needs a fixed vessel heading and position, the TAM system is online. This can be offloading the oil to a shuttle tanker. TAM or PM system allows for larger operating windows and increases structural reliability in extreme conditions, both for spread and turret moored vessels. The mooring system decreases the level of thrust needed, while the thrusters increase position keeping accuracy.

Semisubmersible

Increasingly newly built semisubmersibles are being equipped with advanced mooring systems and some level of thrust capability for improved station keeping ability. The operation modes range from mooring only in relatively shallow water, thruster-assisted mooring in deep water, and DP mode in ultra-deep water conditions.

TAM has been used in the design of station keeping systems for semisubmersibles. The thrusters (or DP) are used for heading and position control and for reducing the mooring load. One of the principal reasons for using TAM is it can reduce overall project costs by reducing the mooring system in terms of either fewer mooring lines or a lighter overall system, especially for deep water projects.

For a TAM system, the thrusters may only be active under relatively severe weather conditions such that therefore the operation and maintenance cost for the DP system could be relatively lower than that of a DP only operation. With exploration and production moving into increasingly deeper water, TAM has become one of the alternative concepts for vessel motion control.

Cross-References

- [Dynamic Positioning in Ice](#)
- [Hydrodynamics for Subsea Systems](#)
- [Ship Motion Control](#)

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Tidal and Ocean Current Turbines

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Introduction

Tides and Currents of Oceans

Over 70% of the Earth is covered by oceans in the world. These oceans are a vast renewable energy



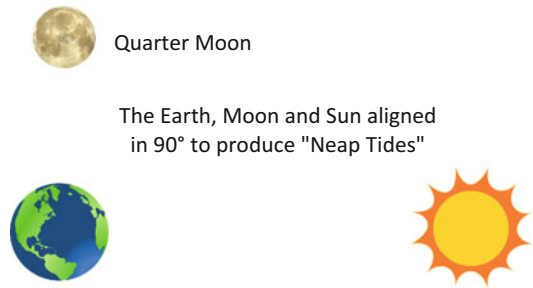
Tidal and Ocean Current Turbines, Fig. 1 Full Moon (New Moon) schematic diagram

source. Energy stored in oceans can be mainly categorized as thermal energy; kinetic energy due to tides, waves and currents; and chemical and biological energies. Tides are caused by planetary motion to produce the flow of seawater. Because the Earth, its Moon, and the Sun rotate around each other, the gravitational movement of the Moon and the Sun with respect to the Earth (see Fig. 1) causes huge amount of water to flow around the Earth's oceans, creating periodic shifts in these moving bodies of water. These vertical shifts of water are called "tides" (Alternative Energy Tutorials 2018).

When the Earth and the Moon's gravity lines up with each other, the influences of these two gravitational forces become very strong and produce a movement of water toward the shore creating a "high tide" condition. When the Moon is in perfect alignment with the Earth and the Sun, the gravitational pull of the Moon and Sun together becomes much stronger than normal with the high tides becoming very high and the low tides becoming very low during each tidal cycle. Such tides are known as spring tides (maximum). These spring tides occur during the Full or New Moon phase.

Similarly, when the Earth and the Moon's gravity are at 90° to each other as shown in Fig. 2, the influences of these two gravitational forces are weaker, and the water flows away from the shore as the mass of water moves to another location on the Earth, creating a "low tide" condition.

This ebbing and flood tides happen twice during each period of rotation of the Earth with stronger weekly and annual lunar cycles superimposed onto these tides. The other tidal situation arises during neap tides (minimum) when the gravitational pull of the Moon and the Sun are against each other, thus cancelling their effects.



Tidal and Ocean Current Turbines, Fig. 2 Quarter Moon when the Earth-Moon and Earth-Sun are aligned in 90°

The net result is a smaller pulling action on the seawater creating much smaller differences between the high and low tides, thereby producing very weak tides. Neap tides occur during the Quarter Moon phase.

The spring tides and neap tides produce different amounts of potential energy in the movement of the seawater as their effects differ from the regular high and low sea levels, and we can use these tidal changes to produce renewable energy. The constant rotational movement of the Earth and the Moon with regard to each other causes huge amounts of water to move around the Earth as the tides go in and out. These tides are predictable and regular resulting in two high tides and two low tides each day with the level of the oceans constantly moving between a high tide and a low tide and then back to a high tide again. The time taken for a tidal cycle to happen is about 12 h and 24 min (called the "diurnal cycle") between two consecutive high tides. The main big advantage of this is that the tides are therefore perfectly predictable and regular unlike wind energy or solar energy, allowing miles of coastline to be used for tidal energy exploitation, and the larger the tidal influence, the greater the movement of the tidal water and therefore the more potential energy that can be harvested for power generation.

Tidal and Current Energy Resources

It is estimated that the total ocean energy reserve in the world is 76 billion kW and the corresponding explorable ocean energy is 15.7 billion kW (Xiang 2010). According to a recent

renewable energy report (Paris: REN21 2017), the total explorable wind power in the world is about 400 TW, and the wind power generation capacity is about 487 GW. Therefore, the wind power utilization rate is extremely small, of about 0.1%. The explorable tidal power in the world is much smaller, about 120 GW, and the tidal power generation capacity about 536 MW, giving a very small utilization rate of 1.2%.

Tidal and Ocean Current Turbines (TOCTs)

The motion of the tidal water, driven by the pull of gravity, contains large amounts of kinetic energy in the form of strong tidal currents called tidal streams. The movement of the seawater can be harnessed in a similar way using waterwheels and turbines to generate hydroelectricity. As the

seawater can flow in both directions in a tidal energy system, it can generate power when the water is flowing in and when it is ebbing out. There are several technologies to harness the kinetic energy from tidal, ocean, and river currents. These are introduced as the following.

Different Types of Tidal and Ocean Current Turbines

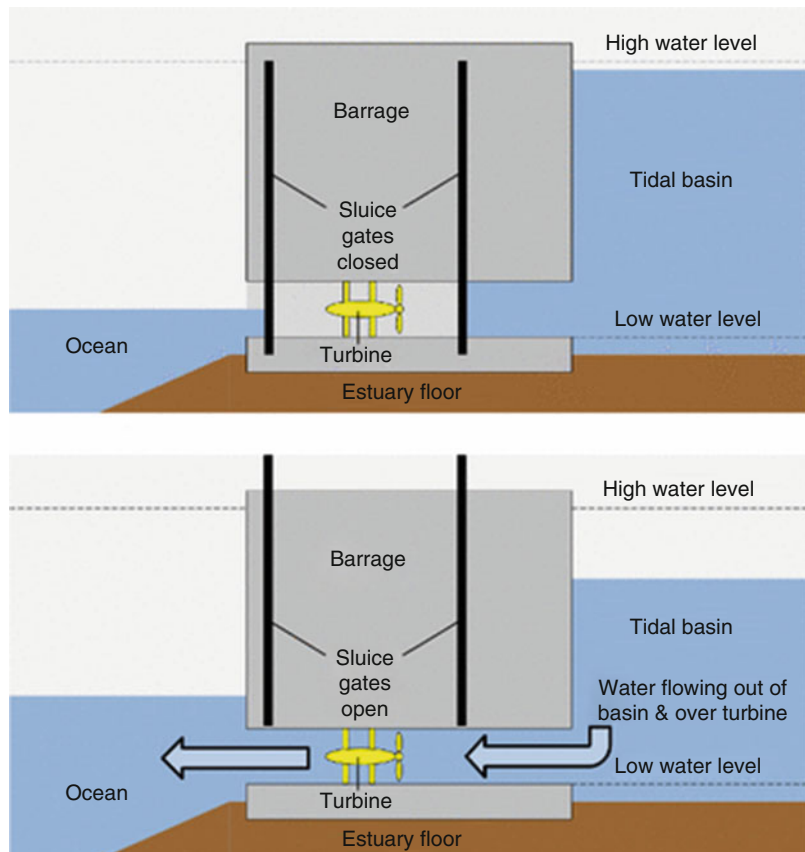
Tidal Barrage

Tidal barrage power systems take advantage of the differences between high tides and low tides by using a “barrage,” a type of dam, to block receding water during ebb periods. At low tide, water behind the barrage is released, and the water passes through a turbine that generates electricity. Figure 3 shows the schematic diagram of a tidal barrage.

One disadvantage of tidal barrage generation is that it can only generate electricity when the tide is

Tidal and Ocean Current Turbines,

Fig. 3 Schematic diagram of a tidal barrage (Roberts et al. 2016)



flowing either “in” or “out,” as during high and low tide times the tidal water is stationary. However, because tides are totally predictable, other power stations can compensate for this stationary period when there is no tidal energy being produced. Another disadvantage of a tidal barrage system is the environmental and ecological effects that a long concrete dam may have on the estuaries they span.

Horizontal Axis Tidal Turbine (HATT)

Tidal stream generation is very similar in principal to wind power generation, except this time water currents flow across turbine rotor blades which rotate the turbine, much like how wind turn the blades for wind power turbines. In fact, tidal stream generation areas on the seabed can

look just like underwater wind turbines (Fig. 4).

Tidal generators are designed to produce power when the rotor blades are spinning in both directions. Tidal stream generators use kinetic energy of moving water to power turbines; in tidal stream power generation, some turbines are submerged under the surface of water with sufficient clearance to avoid hazards to navigation and shipping, and some of them are visible on the ocean surface.

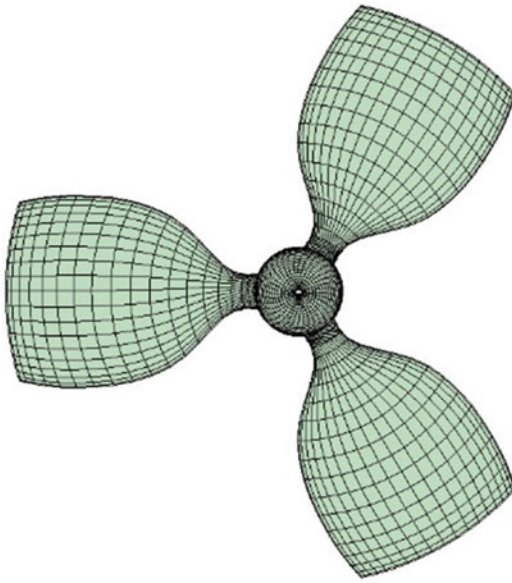
To generate power under both ebb and flood tides, tidal currents are designed to work with bi-directional current. Horizontal axis tidal turbines are designed to be either unidirectional or bi-directional. There exist a few descriptions of bi-directional tidal turbines in the open

literature, including the Tocado (1994) and Swan turbines (Orme et al. 2005). The Tocado turbines are designed to have controllable pitch blades to adapt to the inflow direction; their blades' sectional profile is unidirectional. The Swan turbine rotors have bi-directional blade sectional profile. These are all bare rotor blade turbines with no nozzle or duct. Liu and Bose (2012) prototyped a series of bi-directional bladed turbines for the Bay of Fundy. As the rotor blades are fully symmetrical, this series of turbines have the identical performance in either direction, when the inflow conditions are the same. A new optimization theory was established by setting the total energy production as the key criteria rather than the single power coefficient (Liu and Bose 2012). In the study for a 20-m HATT, pitch, pitch distribution, solidity, number of blades at a fixed solidity, etc. were all optimized for the optimum annual energy production under the given probability distribution of the tidal flow speed and corresponding time, in the Minas Passage, the Bay of Fundy, Canada. For a three-bladed 20-m rotor, the optimum p/D and shaft speed n are 0.27 and 0.17 revolution per second, respectively, producing annual renewable energy of 3141 MWh. The averaged constant power production, under the conditions including ebb and flood tides and all absolute speeds lower than the kick-in speed (about 1.3 m/s), is 385 kW. The solidity, or the expanded area ratio (EAR) for optimum power coefficient ($C_{POW} = 0.32$) at an optimum inflow speed for the 20-m bi-directional rotor in Liu and Bose (2012), was validated by a later

Tidal and Ocean Current Turbines,

Fig. 4 Hammerfest Strom HS1000 Turbine in Islay. (Courtesy of the Tethys), Scotland (Annex IV 2018)

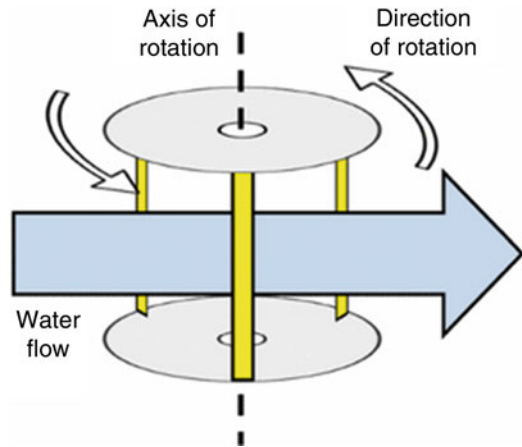




Tidal and Ocean Current Turbines, Fig. 5 An optimized bi-directional rotor with optimum solidity of about 0.4. Solidity is the ratio of the projected area of the blade to the area of the circle with the same diameter of the rotor

experimental work (Liu et al. 2014) to be about 0.45. This EAR value is much larger than most of the HATT turbines currently installed, except for the ducted OpenHydro turbines. The rotor with the optimum EAR for the power coefficient is shown in Fig. 5.

Blade failures have dogged the fledgling tidal current turbine industry and have been seriously affecting its apparent viability. Retrieval, salvage, and recovery operations take substantial effort and time, due to poor weather and harsh underwater conditions. Some failures include, for example, blade failure on the OpenHydro 16-m turbine installed in the Bay of Fundy, NS, Canada, on June 11, 2010 (CBC News 2010); blade failure of the two Verdant Power tidal turbines installed in the East River of New York City on December 11, 2006 (Shiekh et al. 2018); and the blade fracture of the Atlantis AR1000 turbine soon after its installation and connection to the grid of the European Marine Energy Centre (EMEC) in August 2010 (BBC News 2010). Failure of rotor blades is due to two main causes: (Allen 2007) the loads are underestimated or unknown, resulting in inadequate design strength, and (Alternative



Tidal and Ocean Current Turbines, Fig. 6 Schematic diagram of a vertical axis water turbine (Roberts et al. 2016)

Energy Tutorials 2018) manufacturing defects result in a much lower strength structure than designed. For these issues, Liu and Veitch (2012) conducted research work concerning blade sectional loading prediction and structural strength design and optimization to meet the strength requirements and save material without sacrificing hydrodynamic performance. For the same required safety factor of 6.51 at 3.1 m/s for the 20-m rotor with the same nickel-aluminum-bronze material, their method saved substantial material, comparing a refined initial design of a blade weight of 229 t with the optimized weight of 143 t, a total of 86 t (37.6%) of saving in material consumption, about \$17M saving at a cost of \$200/kg.

Vertical Axis Tidal Turbine (VATT)

Other forms of tidal energy include tidal fences which use individual vertical-axis turbines that are mounted within a fence structure, known as the caisson, which completely blocks a channel and forces water through them. Figure 6 shows the operation schematic diagram of a vertical axis turbine.

Oscillating Tidal Hydrofoil (OTH)

Another alternative way of harnessing tidal power is to use oscillating foil. The foil pitch, either passively or actively, creates a lift force

and induces the heave motion. The heave motion with force and displacement will create energy. This energy can be transferred to electricity. A schematic diagram for the early prototype of the oscillating foil turbine is shown in Fig. 7.

There are some innovative ideas on oscillating foil turbine. Dumas (Kinsey and Dumas 2012) developed an oscillating foil hydrokinetic turbine (HAO).

Utilizing the wing-in-ground (WIG) effect, Liu (2013) developed a dual-foil counterphase motion WIG turbine (WIGTT). This technology uses an active pitching control to produce arbitrary pitching motion for the highest possible

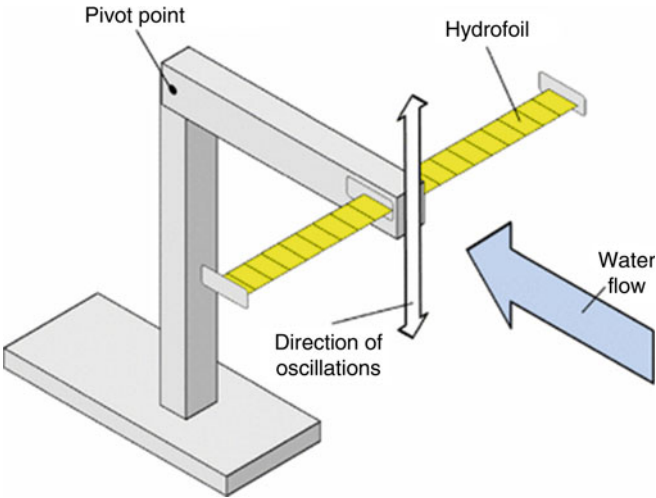
power production. A two-stage pitching mechanism is shown in Fig. 8.

For a two-stage WIGTT, there are two pairs of foils, and they can be arranged in many different configurations. One of the configurations is for the two pairs in tandem, as shown in Fig. 9.

An extensive numerical study was conducted for the design and optimization of the WIG turbine (Liu 2015). The results showed that the WIG turbine using trapezoidal pitching control motion can produce about 40% more power than sinusoidal pitching control. The counterphase pitching and heaving mechanism-produced WIG effect could further improve more than 30% of power production than two single oscillating foils.

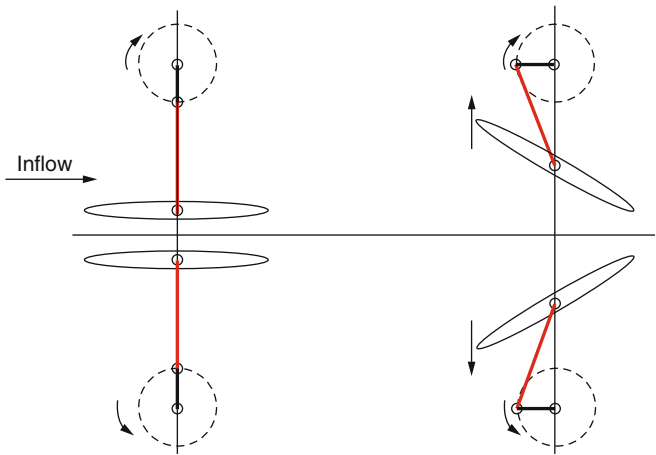
Tidal and Ocean Current Turbines,

Fig. 7 Oscillating hydrofoil tidal turbine (Roberts et al. 2016)

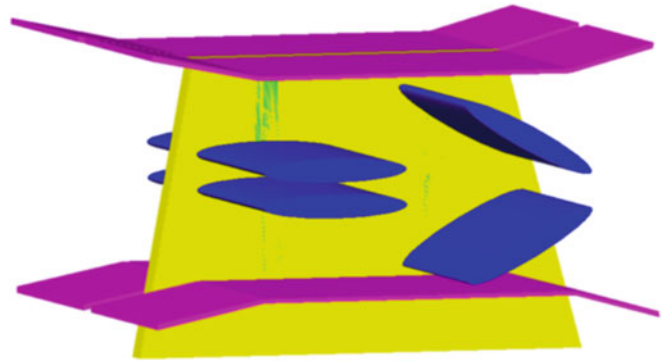


Tidal and Ocean Current Turbines, Fig. 8

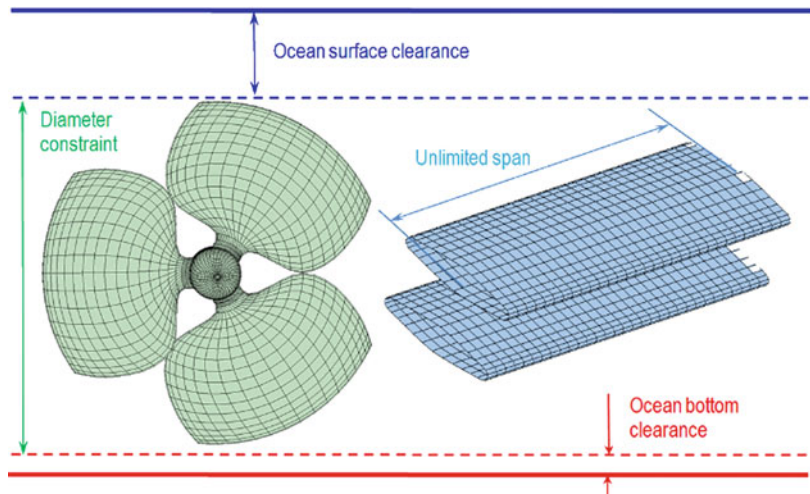
A two-stage active arbitrary pitching control mechanism (Liu 2013)



Tidal and Ocean Current Turbines, Fig. 9 WIG oscillating foil turbine in tandem configuration (Liu 2013)



Tidal and Ocean Current Turbines, Fig. 10 Water depth limitation for HATT diameter versus unlimited span of WIG foils



The power generated by a HAT is $P = 0.5 \cdot C_{\text{pow}} \rho V^3 A$, where C_{pow} , ρ , V , and A are power coefficient, fluid density, tidal inflow velocity, and swept area. By letting the swept areas of HATT and WIGTT be equal, i.e., $A_{\text{HATT}} = \pi D^2 / 4 = A_{\text{WIGTT}} = S_{\text{span}} 2h_a$, where S_{span} and h_a are foil span and the heave amplitude of each foil, time-domain unsteady panel methods for the WIGTT and HATT were applied to predict the power generation performance of a 20-m HATT and WIGTT foil turbine, respectively. The WIGTT produced over 70% more power than the HATT. One of the foils of the dual-foil WIG effect turbine also produced over 1.85 times of a single foil turbine for the same pitching motion without WIG effect.

Why WIG Effect Turbine: The Pros and Cons Based on the results, the advantages

and disadvantages of the WIG effect tidal turbine were listed as the following (Liu 2015):

The pros of the WIG effect turbine:

- High energy production performance, about 1.7 times of HAT with the same swept area
- High power capacity with confined water depth (see Fig. 10)

When the depth, and hence the HATT diameter, is a constraint, to increase energy production by ten times with the same water depth, 10 HATTs will be required to form a farm configuration, but for WIGTT with the same depth, it would need only to extend the span by ten times.

- Less impact on ecology. The maximum velocity at the 20-m HATT rotor tip (Liu and Bose 2012) is $V_{\text{tip}} = \pi D n = 20 \pi 0.17 = 10.68 \text{ m/s}$,

but for the foil of the WIGTT, it is $V_{\text{foil}} = h_{\text{amp}} \omega = 4 * 0.5 = 2.00$ m/s. WIGTT therefore has the following environmental advantages:

- (I) With 1/5 of the motion speed of HATTs, they have much less flow disturbance on surroundings.
- (II) With less than 2 m/s foil motion speed, they are unlikely to threaten/damage most marine creatures in ocean.
- (III) WIGTT foils have wavy flow downstream but no rotation induced by HAT rotors.
- (IV) Having a slower foil motion, WIGTTs operate more quietly.
- Safer operation and better hydrodynamic characteristics. WIGTT has the following benefits for design, maintenance, installation, and operation:
 - (I) Much simpler blade/foil structure – less chances of break-off and fracture of foils/blades than HATTs
 - (II) Less prone to tangling with fishing nets, etc., due to much slower and linear foil motion
 - (III) Less prone to collision damage with debris, ice blocks, wood logs, etc.
 - (IV) Lower requirement on structure strength
 - (V) More even velocity and hence pressure and load distribution on blade/foil, hence less prone to cavitation and its related damage in shallow waters and fast flow sites
 - (VI) Low kick-in speed: 0.5 m/s found in this study versus 1.3 m/s for HATT, applicable for much wide range of flow speeds

The cons of the WIG effect turbine:

- Power is required for pitch control. As mentioned earlier the pitch control consumes power. However, power consumption is only a small portion of the total energy generation. For the optimum ω of about 0.5 rad/sec, pitch control power consumption is about 10% and 2% for trapezoidal and sinusoidal pitch motions, respectively.
- Crankshaft or linear generator. For traditional generator/alternator, WIGTT requires motion

translation from heave to rotation, unless linear generator/alternator is used. Linear generator would eliminate efficiency loss in motion translation.

Vortex-Induced Vibrating Tidal Cylinders (VIVTT)

The technology idea of a bluff body energy converter was originated by Allen (2007). In the technology, a power generating system comprises a bluff (non-streamlined) body placed in a fluid stream at 90° to the flow direction for creating a von Karman vortex street behind the body. The bluff body oscillatory movements are coupled to and energize a power generator. An impedance matching system is employed for electrically varying the natural frequency characteristics of the bluff body such that, over a relatively wide range of fluid flow velocities, the bluff body oscillates at a resonant frequency of the body for maximum energy transfer from the fluid stream to the bluff body. An enhancement to bluff body energy converter technology was patented by Bernitsas et al. (2008), the vortex-induced vibration aquatic clean energy (VIVACE) converter. This study showed that the VIVACE is suitable for slow inflow current, as slow as 0.2 m/sec.

Comparison of Tidal Energy Between Ocean Tidal and Wave Technologies

Tidal energy generation technologies have many advantages compared to other forms of renewable energy. The main advantage is that it is predictable. The following table shows the comparison among the three kinds of renewable technologies.

Table 1 shows that the total explorable ocean wave energy is huge, but because of its low energy density, high cost, and no predictability, there is little completed for the development in wave energy capacity. The total explorable tidal energy is high, though much lower than the wave. The exploration rate of tidal energy is still very low, and most of the installed capacity is contributed by the tidal barrage, which has little room for more. This means that the tidal turbine technology development should be the strategic direction of the ocean tidal energy development.

According to World Energy Council (2016), today about 0.6 GW of commercial ocean energy

Tidal and Ocean Current Turbines, Table 1 Comparison of tidal and wave energy technologies

Recourses	Explorable globally (GW)	Capacity globally (GW)	Energy density	Predictability and stability	Visual impact	Building cost	Operation cost
Wave	3,368,000	0.56	Low	No	Bad	Very high	Very high
Tidal	137000	22	High	Very good	Good	High	High

generation capacity is in operation and another 1.7 GW under construction, with 99% of this accounted for by tidal energy. While relatively few commercial-scale wave, tidal stream projects are operational, there are three tidal stream commercial projects accounting for 17 MW of capacity (two in Scotland and one in France).

Tidal Energy Summary

Because of the predictability and hence the advantage of the reduction of fluctuation of the power grid and the density of the tidal energy, along with several other advantages, tidal energy technologies have experienced the fastest development in the ocean renewable energy industries. Currently the tidal barrage is still the biggest contributor to ocean tidal energy, but the tidal turbine development becomes more and more important strategic development.

The two main issues that hinder the development of tidal turbines are (1) the investment/cost of the manufacture and installation and (2) the utilization of the lower tidal flow speed. With the technology development in tidal turbines, these two issues are being gradually resolved. With the recent development of some novel technologies, for example, the WIGTTs, have much lower environmental impact to both flora and fauna, much lower kick-in speed (0.5 m/s vs. 1.3 m/s for HATTs) of which avails the possibility for rivers of low speed current, much higher power production efficiency, and shallow water applicability, etc. These novel technologies will eventually be at least an alternative or substitutes to traditional HATTs. There have had great establishment and progress in ocean tidal and current turbines, and these technologies proved their promising roles in the ocean renewable energy

stage, though there are still some challenges and issues to be met and addressed.

Acknowledgments The author would like to thank Professor Jonathan Whiting, Pacific Northwest National Laboratory, on behalf of Annex IV, for granting the copyright of Fig. 4 and Springer for the free copyright of Figs. 1, 6, and 7.

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Time of Flight (TOF)

- [Integrated Navigation](#)

TLP (Tension Leg Platform)

- [Tension-Leg platform](#)

Top Tensioned Risers (TTRs)

- [Steel Pipelines and Risers](#)

Touchdown Point

- [Pipeline Soil Interactions](#)

Touchdown Zone (TDZ)

- [Installation of Offshore Pipelines](#)

Tow System

- [Deep Tow System](#)

Tow-Body

- [Deep Tow System](#)

Towed Vehicle

- [Deep Tow System](#)

Tow-Fish

- [Deep Tow System](#)

Towing Experiment

- [Aquaculture Structures: Experimental Techniques](#)

Towing Tank Test

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Shanghai Rainbowfish Deepsea Equipment & Technology Co. Ltd, Shanghai, China

Synonyms

[Maneuvering tests](#); [Propeller open water tests](#); [Resistance test](#); [Seakeeping tests](#); [Self-propulsion tests](#)

Definition

Towing tank tests are tests performed in a towing tank with ship and offshore structure models or submarine and submersible models which include powering performance tests, maneuvering test, seakeeping test, water flow field measurement, and so on, which are the most important and basic parts of ship model tests. Surface ship models, underwater vehicles such as submarine, submersible, torpedo, Autonomous Underwater Vehicle (AUV), Remotely Operated Vehicle (ROV), and so on, amphibious vehicles such as military amphibious tanks for marine troops or even aircrafts used such as wing-in-ground-effect ships, or amphibious aircraft could carry out towing tank model tests. The model tests can be carried out under different environments, such as calm water, various kinds of waves with different height and frequency, or wind. The model, either floating on the water surface or submerged in the water, is usually towed by a device connected with the carriage during the test. The carriage runs on two rails for different speed. Forces or motions are measured. Towing tank tests were the earliest model tests that people carried out, and nowadays they have taken the largest share among all kinds of ship model tests. So when people talk about ship model tests or tank tests, they are talking about towing tank tests in fact (https://en.wikipedia.org/wiki/towing_tank_tests).

Scientific Fundamentals

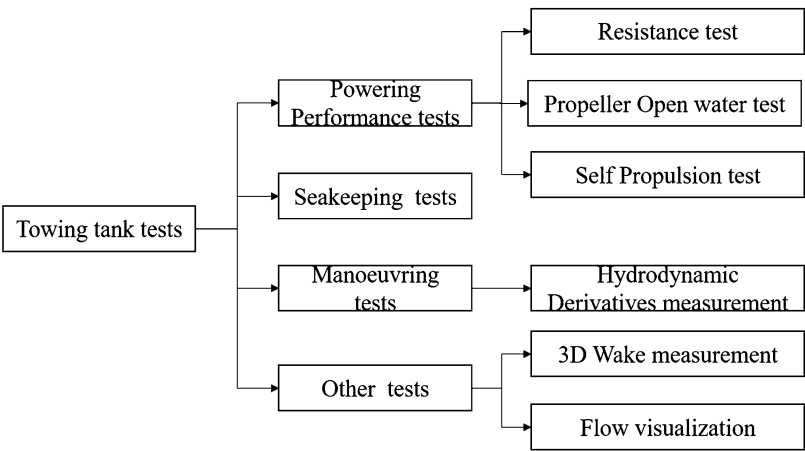
Towing Test Types

A division of various towing tank tests is given in Fig. 1. These include powering performance test, seakeeping test, maneuvering test, and other tests. Powering performance tests include resistance test, propeller open water test, and self-propulsion tests. Other tests include 3D wake measurement and flow visualization tests.

1. Resistance Tests

It is the hull resistance when the ship or water vehicle moves in the water at a speed or if there is water flow around (ITTC 1990a). The force measurement for the water flow that loaded on the vessel is carried out in the same way with the resistance tests. Usually, hull resistance is measured at increasing speeds in order to develop a curve for the vessel at specific condition, one or several displacements, or trim angles. For surface ships, the model tests are carried out mainly in moving-forward direction. However, for underwater vehicles, there will be resistance for several directions, which could be moving forward, backward in the horizontal plane, moving upward and downward in the vertical plane, and at last, moving transverse. Usually the trim angle and the sinkage of ships will be measured during the resistance tests.

Towing Tank Test,
Fig. 1 A division of
various towing tank tests



2. Propeller Open Water Tests

Propeller open water test is the propeller performance test without anything in front; this is why it is called open water. By carrying out this kind of tests, people will know the performance of the propeller, which is the most popular thruster for ships and underwater vehicles. The test data will be recorded by thrust coefficient, torque coefficient, and efficiency.

3. Self-Propulsion Tests

The ship or vessel, the propellers, and motor, three parts of a full-scale ship that can run on the sea, are put together in self-propulsion test. The thrust and torque of the propellers will be measured. The rotation of the propeller is usually settled according to the test plan. Several speeds will be carefully chosen, which are close to the design speed (ITTC 1990b). The speed of the design ship and the rotation of propeller will be predicted with the test data including the hull resistance tests and propeller open water tests all together. These kind of tests are very important for most ships, merchant ships, or navy ships, both surface warships and submarine and torpedoes, which are sensitive to fuel consumption or top speed. However, there are few cases for underwater vehicles models self-propulsion tests. Manned submersibles or Remotely Operated Vehicles (ROV) are not very sensitive to speed or energy consumption. For other underwater vehicles, they are not very big so it will be easier to carry out full-scale tests, not the scaled model tests.

4. Seakeeping Tests

Seakeeping tests are done to evaluate the behaviors of the ships or vessel in waves. In towing tank tests, the waves are ahead at sea or following. Usually the model can be self-propelled or towed with tethered spring. The heave, rolling, pitch, and accelerations will be measured during the tests compared with the wave data. When underwater vehicles are during the launch or recovery process, the waves will have great effect. However, the seakeeping ability is not as important as other performance, in fact, and the designers will not refine

or change the design too much to make the sub better seakeeping.

5. 3D Wake Measurement Tests

The water flow around the propeller is called the wake of the ship. Usually the water data is measured in a round circle with a diameter 20% larger than that of the propeller. Five holes Pitot tubes are tools, and nowadays, 3D Particle Image Velocimetry (PIV) technology is developed and applied more and more.

6. Flow Visualization Tests

For determining the flow pattern along the hull, various flow visualization techniques are available. Paint method or tuft with underwater videography is mainly used, and PIV can be useful and helpful by recording several sections along the ship. In this way, the flow field around the ship can be recorded.

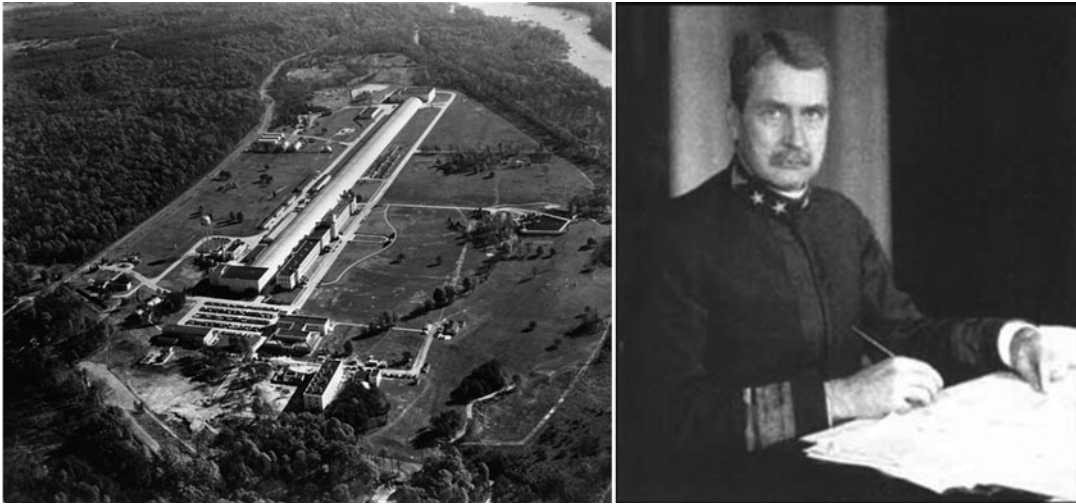
7. Maneuvering Tests

There are usually two kinds of maneuvering tests, free running model tests and captive model tests. As the towing tank is narrow compared to its length, captive model tests are more often carried out. During captive model tests, the model will be fixed in several conditions, different trim angles or rudder angles, for example. The force and moment of the model will be measured. Planar Motion Mechanism (PMM) is an important device for maneuvering tests. Hydrodynamic derivatives in the maneuvering mathematic models can be calculated by using the test data.

Historical Development

It is believed that W. Froude built the first towing tank in the world in Torquay for the Britain Navy Admiralty in 1871. Since then, towing tank tests have been carried out for more than 100 years. More than 150 towing tanks have been built in the world and tens of thousands towing tank tests are carried out every year.

David Taylor Towing Tank was built in Maryland in America. It is still the largest towing tank in the world. See Fig. 2, the photo of David Taylor Towing Tank, which was taken in 1940 after the tank was built. David Taylor himself is seen in Fig. 2. The tank is divided into three parts, a shallow towing tank, a deepwater towing tank,



Towing Tank Test, Fig. 2 David Taylor Towing Tank (on the left side) and David Taylor (on the right side)

and a high-speed towing tank. Research studies on large merchant ships, war ships, and submarines have been carried out ever since 1940, the year it was finished. Powering performance and seakeeping abilities model tests support the navy, coast guard, and offshore projects. Advance carriage, wave makers, and measurement devices make it possible to carry out a model as large as 40 feet long and bias less than 2%.

In China, the first towing tank was built in Shanghai in 1854. It was 70 m long. Then in 1965, a larger towing tank was built in Wuxi by China Ship Scientific Research Centre (CSSRC), 474 m long, 14 m wide, and 7 m deep, shown in Fig. 3. It is one of top three largest three towing tanks in the world (the other in USA and in Russia), and in 2022, it will be enlarged to 1108 m long. CSSRC built a depressurized towing tank in 1985, which is 150 m long, 7 m wide, and 5 m deep, as shown in Fig. 4. The air pressure inside can be decreased to as low as 5% of atmospheric pressure.

The Maritime Research Institute Netherlands was founded in 1929 as the Netherlands Ship Model Basin (NSMB) by the Dutch government and industry. Work was started in 1932, following completion of the deepwater towing tank. To cope with the ever-increasing demands of the industry for research in the fields of powering

performance, seakeeping, and maneuvering, including shallow water effects, cavitation, vibration, noise, etc., a whole series of special test laboratories were successively built (Deep Water Towing Tank 1951, Shallow Water Basin 1958, High Speed Basin 1965, Depressurized Towing Tank 1972, and Cavitation tunnel 1979). A new seakeeping and maneuvering basin became operational in the course of 1999. The upgrading of the depressurized towing tank into an depressurized wave basin was completed in 2012. In 2015, the high speed basin was rebuilt into a concept basin.

MARIN owns four towing tanks, including the deepwater towing tank, the depressurized wave basin, the shallow water basin, and the concept basin. Here the word Basin is equal to the word Tank.

The deepwater towing tank ($252 \times 10.5 \times 5.5$ m) is used to optimize resistance and propulsion characteristics of ship designs. To provide insight in the possible improvements in performance, the tank has the features to measure various wave and flow patterns. In addition to the standard resistance and propulsion tests, the rudder or pod angle, pod position, and propeller rotation direction can be optimized.

The depressurized wave basin ($240 \times 18 \times 8$ m) is a unique research facility for testing ships and offshore structures in the most realistic

Towing Tank Test,**Fig. 3** Deepwater towing tank in CSSRC**Towing Tank Test,****Fig. 4** Depressurized towing tank in CSSRC

operational conditions. The basin is fitted with wave generators, and the air pressure in the basin can be decreased to as low as 2.5% of atmospheric pressure. This way many important research aspects concerning cavitation, wave forces, flooding, and seakeeping can be studied using a properly-scaled condition for both water and air.

The depth of the shallow water basin (220×15.75 m) is adjustable from 0 to 1.15 m. It is used to optimize the propulsion characteristics of ships as well as the (low speed) maneuvering behavior in shallow water. This includes

factors such as proximity of quays and bank suction. The test can be used as input for simulations which help to optimize nautical strategies. The facility is also used for concept development and design support of new offshore designs in shallow water.

The concept basin has a length of 220 m, a width of 4 m, and a depth of 3.6 m. The basin is filled with fresh water. It is mainly designed to perform calm water and seakeeping model tests of ships and structures in the concept phase. Furthermore, the basin can be used for research purposes.

Key Technology in Carrying Out a Towing Tank Test

There are six elements for successful towing tank tests. They are the tank, the model, model speed, turbulence, measurement, and prediction.

1. The Tank

The tank should be built in a good environment where there is no heavy traffic, far from smoke and air pollution. The foundation should be solid and firm, for example, bedrock. At last, there should be fresh water supply nearby.

The tank should be large enough for the model. The cross-section area of ship models should be less than 0.5% of the tank. It is suggested to build the tank as long as possible, for more stable water flow around the ship model and safer for the carriage when speed decreases.

2. The Model

The model ship and the full-scale ship are geometrically similar. The model should be manufactured with high precision, and there are rudders or fins, which should be manufactured and installed to the ship model (Townsin 1990). Sometimes they are too small to be well manufactured. In other cases,

manipulators, for example, are usually too complex to be manufactured. The model is simplified in most cases. How to simplify the model should be recorded.

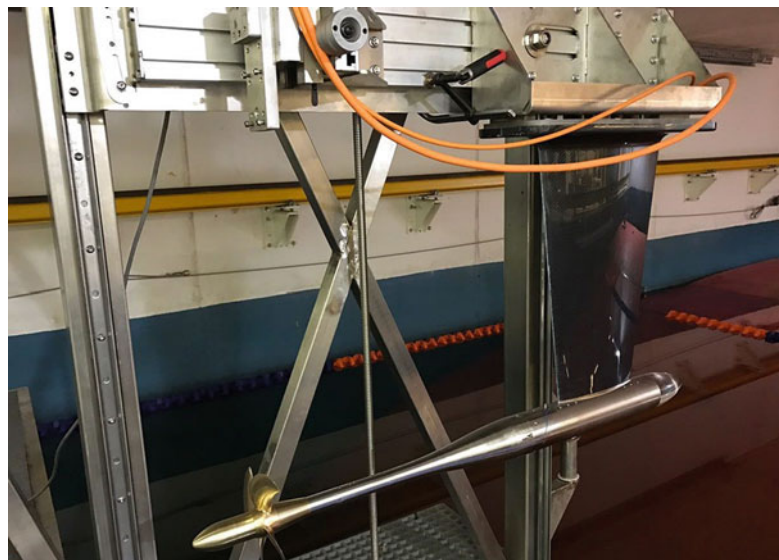
Careful attention should be paid to the roughness of the model skin. Usually the models are painted with some standard methods (Candries 2001).

3. Model Speed

During the tests, the model will be towed by the carriage. So the model speed is the carriage speed unless the centerline of the model is not parallel to the centerline of the running carriage. There is a towing system on the carriage including a towing pole and two towing guide poles, one is close to the stern and the other one is near the bow. The centerline of the towing system should be in the ship model centerline, which should be parallel to the centerline of the running carriage. The carriage runs on two rails, which should be parallel to each other, and the levelness and straightness of each rail should accord with the design and specifications.

The rotation control of the carriage motors and the diameters of each wheel should be checked. There is also a high standard for the speed control system.

Towing Tank Test,
Fig. 5 Open water
dynamometer



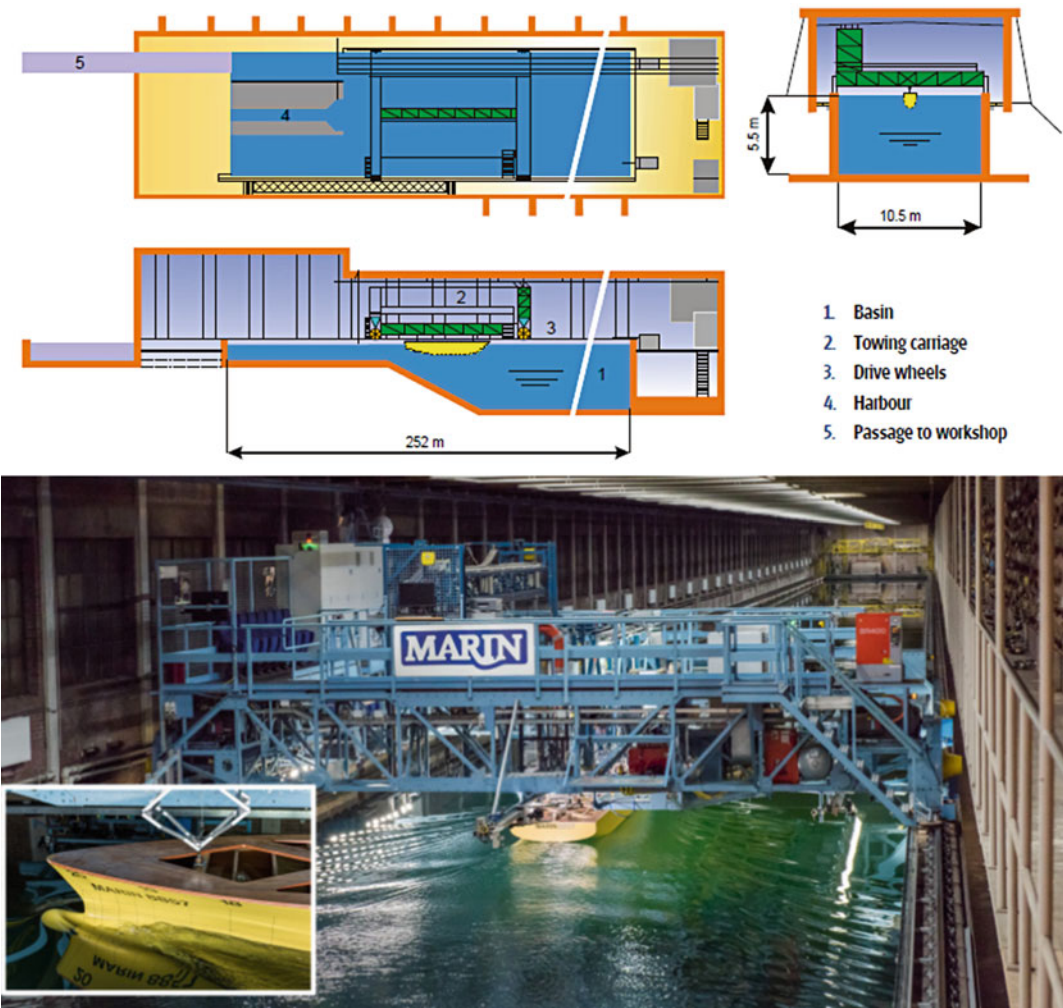
During the tests, the model speed is obviously not that of the full-scale ship. There is relation between them. For surface ships, the Froude Law is applied. In this case, the model speed $V_M = V_S/\sqrt{\lambda}$. Here, V_M is the model speed, V_S is the full-scale ship speed, and λ is the scale factor between the model and the ship.

4. Turbulence

The model is much smaller than the full-scale ship. In the hydrodynamic view, the model has a much smaller Reynolds number than the full-scale ship. Turbulence will be quite different near the model and in the

boundary layer, where it is more important. For example, there will be more laminar flow covering the model much larger than that of the full-scale ship. Then, the friction resistance could not be calculated in a scientific way. The laminar flow and turbulence flow perform quite differently from each other. Take the flow around a cylinder as an example, and there will be a lot of separation vortex for turbulence and much less for laminar flow.

Small wires or nails have been used to be turbulence stimulators on the model, which will stimulate the laminar to be turbulent. It's important where to fix them and how much.



Towing Tank Test, Fig. 6 Deep water towing tank in MARIN

5. Measurement

There are hull resistance devices to measure hull resistance, open water dynamometers for propeller open water tests, and self-propulsion dynamometers for self-propulsion tests. The devices are specially designed and should be regularly checked. See Fig. 5, a propeller model made of copper alloy was fixed in the front of a shaft, which is one part of the open water dynamometer. When the carriage started, the moving water would hit the propeller model without hitting other device or installation. This is why these tests are called open water.

6. Prediction

There is always bias between the model data and the full-scale ship performance. Scale effects play the most important role. The water flow between the model and the full-scale ship is always different even we have

already manufactured similar model and fixed turbulence stimulators (ITTC 2002). Hydrodynamic perform different for different scale and dimensions, which is called scale effect. There is no scientific method to solve the problem. However, many engineered methods have been developed. Friction allowance coefficient is widely applied today, based on thousands of samples between towing tank tests and full-scale ship sea trails (Townsin et al. 1984).

Key Applications

1. Deep Water Towing Tank in MARIN

The deep water towing tank in MARIN is not the largest in the world, but it is a representative tank.

Towing Tank Test,
Fig. 7 The model of
7000m Jialong Sub



Towing Tank Test,
Fig. 8 Towing tank tests
(forward resistance tests)
($\alpha = 0^\circ$)



The tank is 250 m long, as shown in Fig. 6. There were wave makers on the end of the tank, which were removed.

Resistance and self-propulsion tests, open water tests of various types of propulsor (open or ducted propeller), 3-D wake surveys, measurement of hydrodynamic forces and moments on submerged bodies, foils, etc., flow observation tests by 3D Particle Image Velocimetry (3D-PIV), paint or tufts, vertical/horizontal planar motion measurements, and longitudinal wave cut experiments.

2. 7000m Jiaolong Submodel Towing Tank Tests

The 7000m Jiaolong Sub has carried a lot of model tests during different design periods, shown in Fig. 7. Towing tank tests played an important role, as shown in Fig. 8.

Cross-References

► [Wind Tunnel Test](#)

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Traditional Aquaculture Structures

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Synonyms

[Fish cage](#); [Long-line aquaculture](#); [Mussel farm](#); [Offshore fish farm](#)

Definition

To satisfy the increasing demand for fishery products, net cage and raft facilities have been developed to raise fish and shellfish. A net cage is a kind of structure that mainly consists of a floating collar, a net, sinkers, and mooring lines. According to its operation performance, net cages can be divided into four types: gravity net cages, floating rope cages, spar sea stations, and tension-leg cages. Raft aquaculture systems are easy to operate and can be used to realize the culture of scallops, oysters, mussels, algae, and other species; consequently, the use of such systems has rapidly expanded in numerous countries.

Development History and Classification

The global demand for fishery products is becoming increasingly severe as the world's population expands and wild fish stocks decline. Marine aquaculture is expanding all over the world, and net cages and raft aquaculture systems are becoming prevalent in the aquaculture industry to satisfy the increasing need for sea food.

Traditional Net Cage Development History

According to records, cage culture began in Cambodia, with a history of more than 100 years. Japan initially began to examine wave-resistant net cages, and by the end of the 1980s, Japan



Traditional Aquaculture Structures, Fig. 1 Picture of a gravity net cage (Bi et al. 2020a)

had developed a large offshore net cage system. At present, Norway is at the forefront of offshore net cage development.

Traditional Net Cage Classification

Gravity Net Cage

Norway is a major producer of gravity cages. Gravity cages occupy a large share of the net cage market due to their superior anti-wave performance and convenient daily management and maintenance. China introduced gravity cages and started independent research and development in 1998.

The gravity cage is mainly composed of a floating system, net, counterweight system, and mooring system (Fig. 1). Floating systems are generally composed of two or three rows of high-density polyethylene floating pipes bent into circular or square frames. The floating pipe diameter is generally 200–300 mm. The diameter of the floating system depends on the size of the net cage. Floating systems appear in many different sizes, although they usually tend to be extremely large. The floating system provides buoyancy for the cage and supports the net system, which is mainly used to maintain the aquaculture water body, prevent fish from escaping, and prevent enemy invasion. Therefore, single- or double-layer net systems are adopted in practical applications. The net system is generally made of

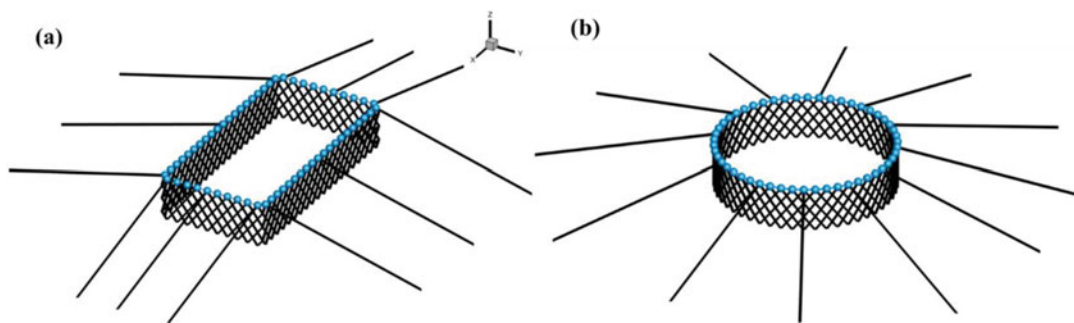
polyethylene or nylon material. To reduce friction damage to fish, the mesh is often woven into a non-nodular configuration. The size of the mesh depends on the size and varieties of the fish. A net system that is directly linked to a float collar is generally designed to be cylindrical. The counterweight system considerably influences the hydrodynamic characteristics of the gravity cage and deformation of the net. The mooring system fixes the net cage in the designated area and limits the drift of the net cage.

Floating Rope Cage

Floating rope cages first appeared in Japan and consisted of ropes, boxes, floaters, etc. (Fig. 2). Because of the flexible floating system, the entire net cage can move with the waves, thereby enhancing the anti-wave performance. In addition, the cage is a kind of six-sided closed box body, which prevents it from being damaged by winds and waves and prevents the fish from escaping.

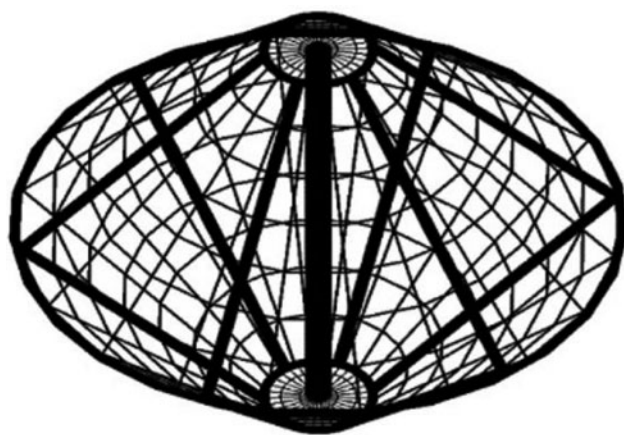
Spar Sea Station

A spar sea station is mainly composed of the frame body and mooring system (Fig. 3). The frame mainly includes the working platform, center column, pendant, net bag, wire rope, and inflation system. In China, spar sea stations are enhanced in terms of two aspects, specifically, the center column and working platform. The



Traditional Aquaculture Structures, Fig. 2 Picture of a floating rope cage

Traditional Aquaculture Structures, Fig. 3 Picture of a spar sea station (Xu and Qin 2020)



greatest innovation is that the working platform can slide along the column, and the floating state of the cage can be adjusted. Owing to these enhancements, spar sea stations are easy to clean; moreover, the net bag can be conveniently checked, and it is easy to catch and classify the fish.

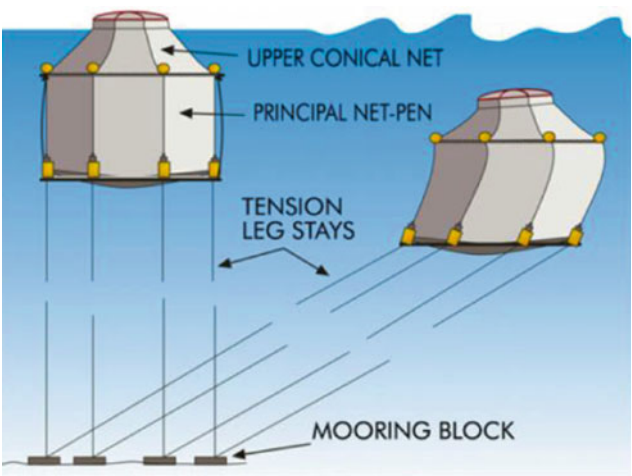
The particularity of the spar sea station determines its specific characteristics. Spar sea stations can resist the effects of 5–10 m high waves. Moreover, such stations have a small volume loss rate, long service life, large net change period, and low cost. Furthermore, spar sea stations can be dragged as one unit, thereby effectively avoiding the effects of red tides. Due to their excellent anti-wind and anti-current performance, spar sea stations have emerged as the primary choice for aquaculture in sea regions characterized by large water depths and wave heights. However, such cages also suffer from a fatal weakness.

Compared with gravity cages, spar sea stations are extremely large, and as a result, their effective aquaculture volume is only one-third of that of a gravity cage having the same height and diameter. This disadvantage limits the operation management; moreover, additional ancillary facilities and professional management personnel are needed, which results in a cost increase.

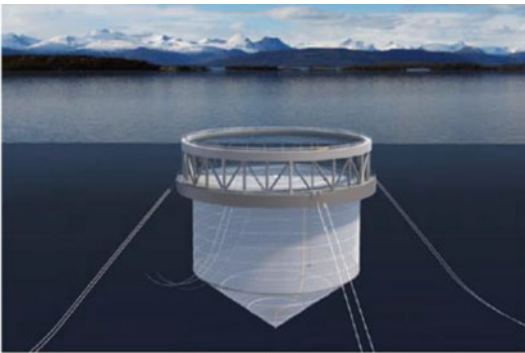
Tension-Leg Cage

Tension-leg cages (Fig. 4), which exhibit high wind and wave resistance, represent one of Norway's main cage products. A tension-leg cage is composed of jar-shaped cages, tension legs, and mooring blocks in three parts. The jar-shaped cages are the main body for fish breeding and adult fishing. The tension legs are usually extensible ropes that tie the net to the mooring blocks. The mooring blocks are fixed on the bottom of the sea to set the shape of the net.

Traditional Aquaculture Structures, Fig. 4 Picture of a tension-leg cage. (Picture from http://refamed.com/gabbie_mare/tlc_system.html)



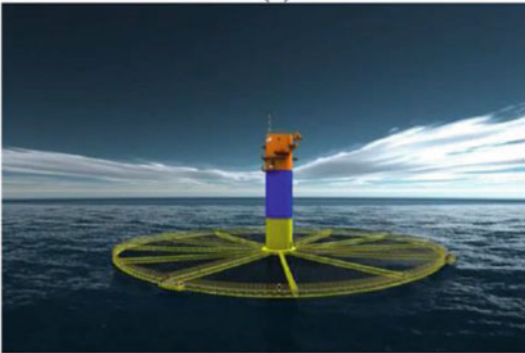
(a)



(b)



(c)



(d)

Traditional Aquaculture Structures, Fig. 5 Pictures of novel platforms in fish farm: (a) surface floating vessel-shaped salmon farm; (b) semi-submersible rigid fish farm;

(c) semi-submersible rigid fish farm, Ocean Farm 1; and (d) semi-submersible spar fish farm concept (Xu and Qin 2020)

Novel Platforms

In recent years, many large offshore fish farms such as vessel-shaped, semi-submersible rigid fish farm and semi-submersible spar fish farm

have achieved a satisfactory performance in terms of current and wave resistance. Figure 5 shows photos of several new fish farming facilities among many others.

Raft Culture Development History

Raft structures are mostly concentrated in coastal waters. Due to the excessively high breeding density, marine pollution, and deficiency of nutrition in coastal waters, which result in the high mortality of shellfish in shallow seas, it is imperative for long-line aquaculture to be developed toward the open sea to expand the breeding space. However, without the protection of the bay, intense sea conditions pose a considerable challenge to the safety of offshore aquaculture facilities. Scholars have adopted various methods to optimize the structural design to resist wind and waves. Chile and New Zealand are some of the main countries using raft facilities to breed mussels. In China, raft facilities are mainly used to cultivate kelp, wakame, stone cauliflower, mussels, and scallops.

Raft Culture Classification

The raft culture framework is mainly used to culture shellfish (scallops, oysters, abalone) and algae (kelp, nori). Raft facilities have two main forms: sleeves and lantern nets. Sleeves are mainly used to realize the aquaculture of algae

and mussels; lantern nets are used to culture several kinds of shellfish (Fig. 6).

According to their different buoyancy systems, raft facilities can be divided into long-line structures and HDPE tube structures.

Long-Line Aquaculture Facilities

Long-line aquaculture facilities represent a form of raft farming that is generally adopted in the industrial aquaculture of shellfish and algae. Each long line consists of two parallel backbone ropes supported by buoys spaced a few meters apart (Fig. 7).

HDPE Tube Aquaculture Facility

This kind of long-line aquaculture facility uses high-density polyethylene (HDPE) tubes as the buoyancy system. Several studies have demonstrated that long-line HDPE tubes are highly stable under wave action, which facilitates the growth of products.

According to the position of the buoyancy system, it can be divided into two types: a floating type and submerged type (Fig. 8).

Traditional Aquaculture Structures,

Fig. 6 Pictures of raft facilities with sleeves and lantern nets. (Picture from <http://www.xn121.com/pictures/gqt/06/1718449.shtml>)



Sleeves for Mussel



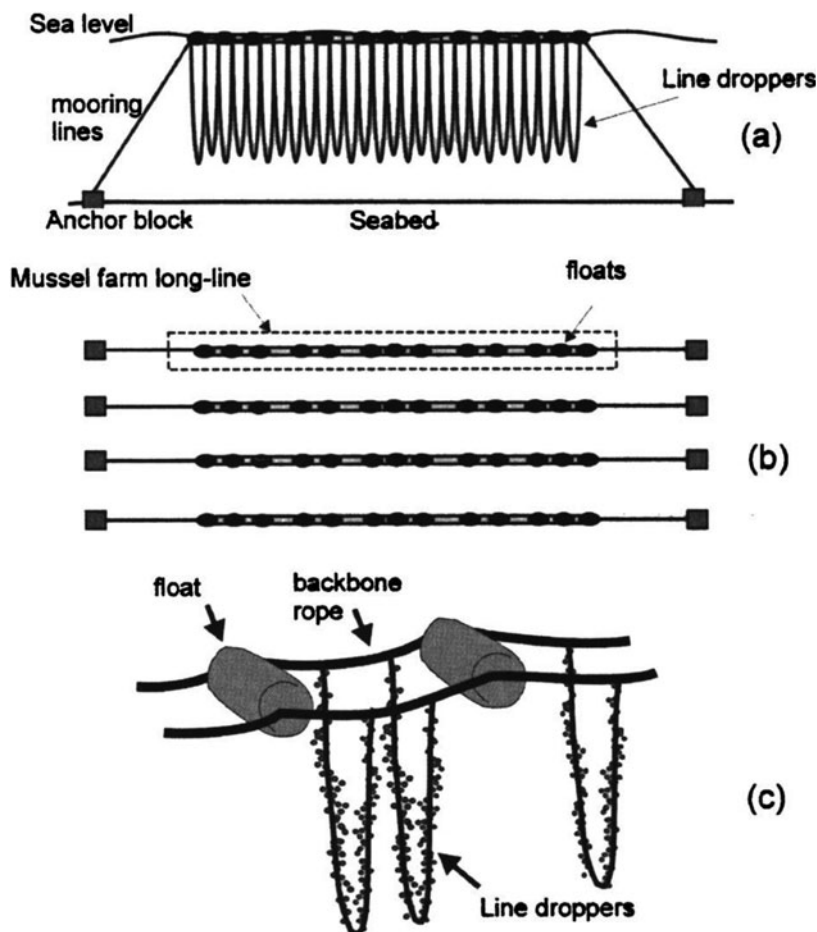
Sleeves for Alga



Sleeves for Shellfish

Traditional Aquaculture Structures,

Fig. 7 Sketches of a New Zealand-style long-line mussel farm: (a) elevation mooring lines are usually aligned approximately 25° from the horizontal; (b) plan view of a block of long lines that may contain as many as 15 individual long lines; and (c) small section of a long-line system showing floats, backbone ropes, and stocked line droppers (Stevens et al. 2008)



Floating Type

The buoyancy system floats on water and is thus suitable for calm sea areas. This structure is easy to manage and harvest.

Submerged Type

The submerged raft structure carries the buoyancy system to a certain depth below the surface. In this manner, the structure can avoid the damage caused by strong winds and waves and remain stable. Such structures are suitable for rough sea conditions, although management and harvesting are difficult.

Development Trend

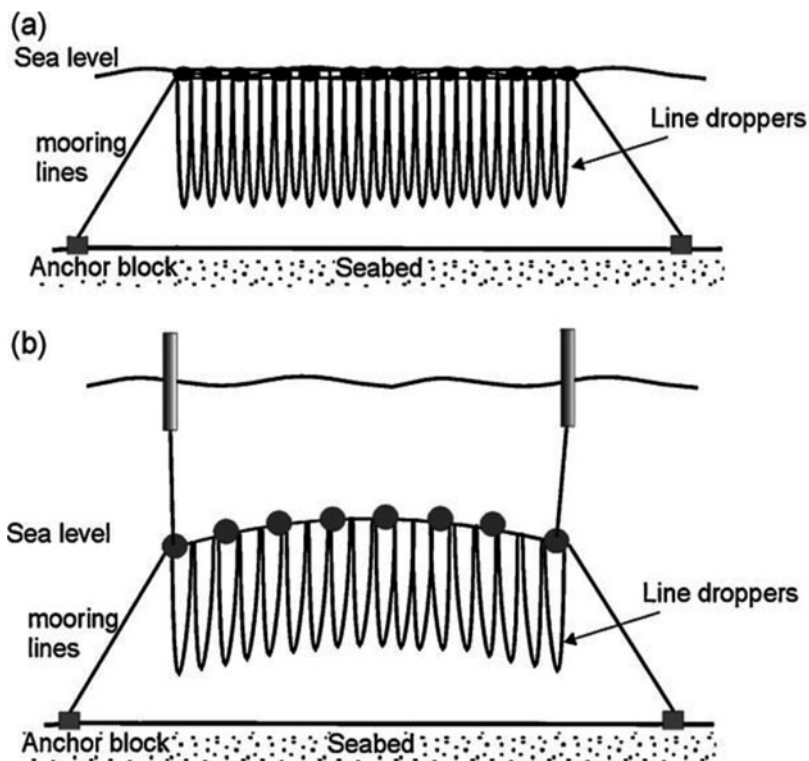
Traditional net cage culture and raft culture systems can only be arranged in shallow bays. These structures are vulnerable to terrestrial pollution

and exhibit inferior wind and wave resistance. Moreover, the organisms attached to the net affect the water exchange, and the residual bait cannot be discharged in time. All these issues lead to the frequent occurrence of diseases and deterioration of the product quality. In addition, the marine environment near the shore is polluted, which limits the supply of coastal waters for use. Consequently, the marine aquaculture industry has been forced to move farther into the sea.

Compared with traditional offshore aquaculture structures, offshore aquaculture facilities are characterized by a large capacity, high yield, and high efficiency and are ecologically friendly. Such structures represent an advanced mode of the rational development and utilization of marine fishery resources. As offshore aquaculture structures are located in deeper waters, they are

Traditional Aquaculture Structures,

Fig. 8 Sketches of (a) floating-type and (b) submerged-type systems (Stevens et al. 2008)



subjected to environmental loads caused by adverse sea conditions. The design of their net cages and raft structures is closely related to their hydrodynamic characteristics. The simulation of the hydrodynamic response is extremely complex and challenging, involving multiple disciplines such as hydrodynamics, structural mechanics, soil mechanics, and ethology. Researchers have conducted extensive research through theoretical analyses, physical model tests, numerical simulations, and even field prototype observations.

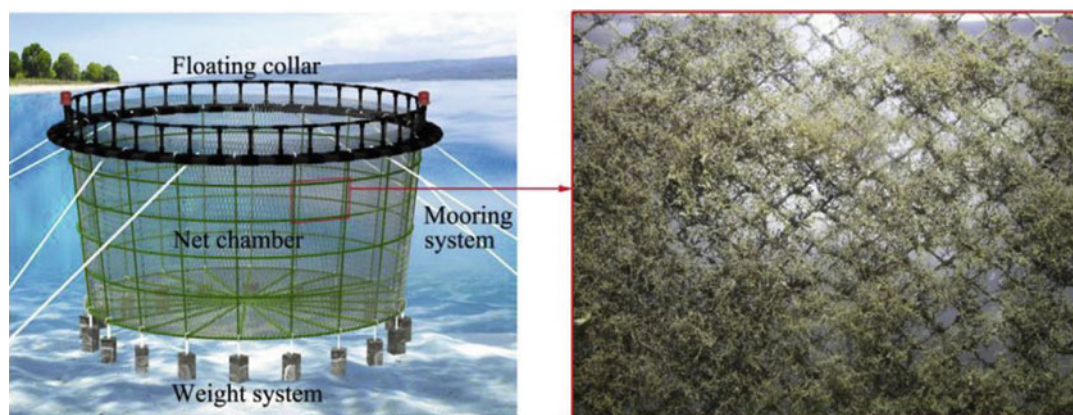
Research Focus

Net Cage

Biofouling

Biofouling is a phenomenon involving the accumulation of marine organisms (Fig. 9), which can lead to rapid occlusion of the netting mesh (Braithwaite et al. 2007). Biofouling communities

act as disease reservoirs and can increase the risk to farmed fish (Fitridge et al. 2012). Moreover, the hydrodynamic forces acting on the net cage are dramatically increased, thereby increasing the risk of structural failure and fatigue damage (Klebert et al. 2013). Biofouled netting may easily deform in currents, thereby detrimentally affecting the effective volume of farmed fish. In addition, the attenuation of currents through the net cage increases, which decreases the water exchange and oxygen supply inside the net cage (Bi et al. 2015). In addition, the motion responses, geometric deformation, dynamic forces, and flow fields regarding the net cage are important for both the design and the maintenance of offshore aquaculture facilities. Thus, investigating the hydrodynamic characteristics of biofouled nets is of significance for aquaculture engineering. In recent decades, a large number of studies have been carried out to investigate the effects of biofouling on the hydrodynamic characteristics of the net structure (Swift et al. 2006; Gansel et al. 2015; Bi et al. 2018).



Traditional Aquaculture Structures, Fig. 9 Sketch of the net cage (left) and photo of a biofouled net (right) (Bi et al. 2018)

Design of Mooring System

The design factors of mooring systems considerably influence the hydrodynamic characteristics of marine structures. Huang and Pan (2010) and DeCew et al. (2010) studied the hydrodynamic behaviors of net cages with different mooring systems for reference. Single-point mooring, grid array, and fixed multipoint mooring arrangements have been considered for net cages. The mooring scope should be considered in the design of a mooring system to avoid collision or entanglement with other components (DeCew et al. 2010).

Flow Reduction

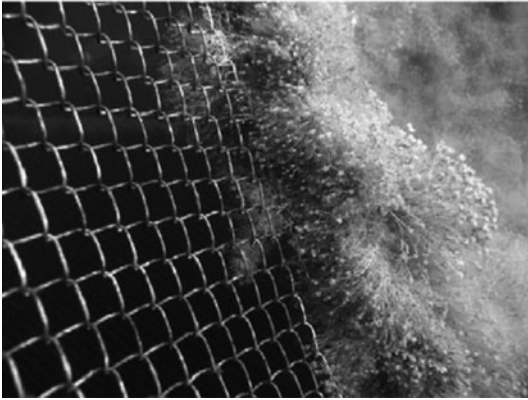
In the open ocean, fluid flows through the net cage and raft structure, and the motion of the flow is highly complex. A net is a kind of small-scale flexible structure. The interaction between the net and fluid is highly complicated. Although the hydrodynamics of the fishing net have been extensively examined, the behaviors of the fluid flowing around a fishing net and velocity reduction downstream of a fishing net remain problems to be solved. These mechanisms have been typically investigated using scaled physical and numerical models and, where possible, field measurements. From both engineering and ecological perspectives, the flow field of open-ocean aquaculture net cages under the effect of steady ocean currents must be considered. Aarsnes et al. (1990) carried out a series of tests to study the velocity

distribution within net cage systems and developed velocity reduction formulas for the net cages. Fredriksson (2001) studied the flow velocity in an open ocean cage through field measurements and observed that the velocity was reduced by 10%. Lader et al. (2003) conducted a series of experiments to investigate the forces and geometry of a net cage in uniform flow and measured an average velocity reduction of 20% inside the cage. Johansson et al. (2007) performed field measurements at four farms in Norway and observed a considerable reduction (33–64%) in the current passing through the cages. Gansel et al. (2012) conducted laboratory tests and field measurements to study the effects of biofouling and fish behavior on the flow field inside and around stocked salmon fish cages. More recently, Bi et al. (2014) developed a numerical strategy to study the flow inside and around flexible fish cages by combining the porous-media fluid model and the lumped-mass mechanical model based on the work of Zhao et al. (2013). Rasmussen et al. (2015) conducted full-scale measurements to visualize the flow field in the wake of a salmon farm.

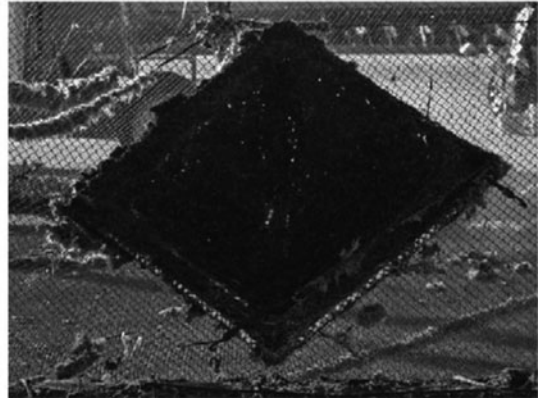
Net Material

Traditionally, net chambers are composed of nylon netting. However, potential improvements in fish farming operations and maintenance can be achieved by substituting polymer nets with copper

a



b



Traditional Aquaculture Structures, Fig. 10 Photographs of a fish cage after 7 months of deployment in the North Atlantic. (a) Image of copper netting in the ocean

with a section of biofouled nylon netting; (b) heavily fouled diver door made of nylon (Tsukrov et al. 2011)

alloy netting. In particular, copper nets demonstrate excellent resistance to corrosion and biofouling (Tsukrov et al. 2011), the potential to protect fish against predator attacks and increased structural integrity and stability of the net structure volume under extreme weather conditions. The effectiveness of the biofouling resistance of copper netting is illustrated in Fig. 10.

Numerical Method

As a kind of flexible structure, a net produces large deformations under the action of marine environmental loads. Therefore, research on the hydrodynamics of the whole structure is focused on net deformation.

One of the primary engineering problems to be solved is the design of aquaculture structures built to withstand the environmental loads of the open ocean. These loads are commonly due to water currents and surface waves. The design process often relies heavily upon the use of physical model tests in wave/tow basin facilities. Recently, numerical models have been developed and applied specifically for open ocean aquaculture applications. Most of these models adopt the finite element analysis (FEA) and lumped mass method with the Morison equation formulation to calculate the loads due to currents and waves (Zhao et al. 2010). Tsukrov et al.

(2003) proposed the concept of consistent finite elements and introduced nonlinear elements to optimize the numerical model to calculate the hydrodynamic response of the net panel. Some researchers also used ABAQUSTM with the ABAQUS/Aqua model. Kristiansen and Faltinsen (2012) applied a screen type of force model to investigate the current loads on the fish cage.

To calculate the current field, because a net can be approximated by many small cylinders connected with knots, Patursson et al. (2010) and Zhao et al. (2013) proposed an approach to simulate the net as a thin volume of porous media to analyze the flow field around the net cage structure. In this approach, the net cage is modeled as a porous membrane with a finite thickness over which the total pressure drop in the incoming current is defined by a combination of Darcy's law and an additional term accounting for the inertial effects. Although a series of studies were conducted on the flow field inside and outside the net structure, the details of the flow field around each net mesh have not been clarified. Notably, the numerical simulation of the flow field around a stocked net cage is complex due to the innumerable meshes and other properties, such as the boundary conditions of the fish skin and structural flexibility.

Fish Behavior

The presence of fish occupies up to 2.5% of the volume, which influences the flow through the internal structure as well as the hydrodynamic loads (He et al. 2018). Experimental and CFD methods have been used to predict the influence of fish behavior on aquaculture facilities (Johansson et al. 2014; He et al. 2018; Bi et al. 2020b).

Raft Aquaculture

Factors Influencing Product Growth

A greater oxygen exchange and higher stable water temperatures can accelerate the growth of scallops (Jensen et al. 2007). However, certain studies have highlighted the negative effect of the movement of suspended enclosures introduced by wave actions on the growth of scallops (Freites et al. 1999). Strong wave exposure and movement of the structure may lead to the increased allocation of energy to body maintenance and production of skeletal structures, leaving a reduced amount of energy for growth (Kingzett and Bourne 1991).

Design Factors

The design of the raft structure, such as the component materials and space of individual sleeves, also effect growth (Díaz et al. 2011). The difference in growth can be attributed to the amplitude of structural motion, as calm structures result in a high yield. Wood blocks, rubber tubing, and small-floater surface buoys can be used to minimize the effect of shaking caused by wave action.

Arrangement

Submerged raft structures exhibit a higher resistance to waves and current conditions, thereby rendering the structures less vulnerable to structural damage, deformation, and abrasion and achieving high survival rates by producing individuals of a higher quality (Lee et al. 2009). Furthermore, the harmful effects of toxic algal blooms can be avoided in this configuration. Increasing the depth of lantern nets, although not to the point at which growth is limited by the

lower temperatures and decreased food abundance, is another strategy to reduce fouling and limit competing organisms (Lodeiros and Himmelman 1996). The configuration and culture depth may lead to differences in food production (Emerson et al. 1994). Gagnon and Bergeron (2017) observed the forces and motion of a mussel long-line system in the open ocean to determine the drag coefficient of the droppers and shielding effect on the bulk drag force exerted by steady currents.

Numerical Method

Numerical models using the finite element method were developed or adapted to predict the loading and motion of raft facilities of various designs under currents and monochromatic wave forcing (Stevens et al. 2008; Tsukrov et al. 2003). A simulation model of the structure involving the lumped mass method was used to calculate the displacement and mooring line forces of an off-shore submerged buoy cable structure subjected to ocean waves (Matsubara et al. 1985). To numerically examine suspended long-line culture systems with lantern nets, López et al. (2017) analyzed the stress on a submersible long-line culture system with lantern nets subjected to parallel and perpendicular forces by using a dynamic simulation model based on the finite element method. Many studies have proposed that suspended kelp and scallop aquaculture systems affect the circulation and considerably reduce current speeds (Lin et al. 2016; Plew et al. 2015). In addition, severe fouling may be accompanied by suspended cultivation systems.

Net Cage Culture Distribution and Output in China

According to the statistics of the China Fishery Statistical Yearbook of 2020, the output of the traditional net cage culture in 2019 was 550,317 tons, 7.44% less than that in 2018, accounting for 2.66% of the total culture output in 2019. Moreover, the traditional cage culture area in 2019 was 22,926,367 m², 55.74% less than that in 2018, accounting for 0.12% of the total culture area in China in 2019 (Table 1).

Traditional Aquaculture Structures, Table 1 Output and area of traditional cage culture in all provinces in 2019

Region	Output (t)	Aquaculture area (m ²)
Total	550,317	22,926,367
Liaoning	1576	373,071
Zhejiang	23,317	619,192
Fujian	283,128	14,682,814
Shandong	71,950	2,179,784
Guangdong	119,662	3,446,924
Guangxi	33,224	612,493
Hainan	17,460	1,012,089

Traditional Aquaculture Structures, Table 2 Output and area of raft culture in all provinces in 2019

Region	Output (t)	Aquaculture area (m ²)
Total	6,174,565	325,314
Hebei	299,333	48,417
Liaoning	1,061,006	47,517
Jiangsu	104,648	42,974
Zhejiang	486,909	19,983
Fujian	1,524,792	44,062
Shandong	1,882,345	98,439
Guangdong	445,052	16,996
Guangxi	370,480	6926

Raft Culture Distribution and Output in China

According to the statistics of the China Fishery Statistical Yearbook of 2020, the raft culture output in 2019 was 6174,565 tons, corresponding to an increase of 0.79% compared to that in 2018, accounting for 29.9% of the total culture production in 2019. The raft aquaculture water area was 325,314 m², corresponding to a decrease of 4.07% compared with that in 2018, accounting for 16.3% of China's total culture area in 2019 (Table 2).

Cross-References

- [Modern Aquaculture Structures](#)
- [Net Structures: Biofouling and Antifouling](#)
- [Net Structures: Design](#)
- [Net Structures: Hydrodynamics](#)

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Transport

► Transport Ship

Transport Ship

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Synonyms

Transport; Reefer

Definition

Ships and other watercraft are used for maritime transport. Types can be distinguished by propulsion, size, or cargo type. Shipping may be for commerce, for recreation, or for military purposes.

Scientific Fundaments

Basic Type and Development History

Most modern transport ships can be placed in one of a few categories.

Bulk carrier, or, bulk freighter, is a merchant ship specially designed to transport unpackaged bulk cargo, such as grains, coal, ore, and cement, in its cargo holds. Since the first specialized bulk carrier was built in 1852, economic forces have fueled the development of these ships, causing them to grow in size and sophistication. Today's bulk carriers are specially designed to maximize capacity, safety, efficiency, and durability. As of 1999, the International Convention for the Safety of Life at Sea defines a bulk carrier as “a ship constructed with a single deck, top side tanks and hopper side tanks in cargo spaces and intended to primarily carry dry cargo in bulk; an ore carrier; or a combination carrier” (van Dokkum 2003) which can be seen in Fig. 1. However, most classification societies use a broader definition where

a bulk carrier is any ship that carries dry unpackaged goods. Multipurpose cargo ships can carry bulk cargo but can also carry other cargoes and are not specifically designed for bulk carriage. The term “dry bulk carrier” is used to distinguish bulk carriers from bulk liquid carriers such as oil, chemical, or liquefied petroleum gas carriers. Very small bulk carriers are almost indistinguishable from general cargo ships, and they are often classified based more on the ship's use than its design (Frankel 1985).

Today, bulk carriers make up 15–17% of the world's merchant fleets and range in size from single-hold mini-bulk carriers to mammoth ore ships able to carry 400,000 metric tons of deadweight (DWT) (Haylar 2003). A number of specialized designs exist: some can unload their own cargo, some depend on port facilities for unloading, and some even package the cargo as it is loaded. Over half of all bulk carriers have Greek, Japanese, or Chinese owners, and more than a quarter are registered in Panama. South Korea is the largest single builder of bulk carriers, and 82% of these ships were built in Asia.

Container ship is the cargo ship that carries entire load in truck-size containers, in a technique called containerization. They form a common means of commercial intermodal freight transport. Informally known as “box boats,” they carry the majority of the world's dry cargo. Most container ships are propelled by diesel engines and have

Transport Ship,

Fig. 1 Bulk carrier – Sabrina I. (<https://en.wikipedia.org/>)



about 10 to 30 crew members. They generally have a large accommodation block at the stern, directly above the engine room. Since the 1960s the transport of containers has continued to grow. The specific advantage of the use of containers is that the cargo can be transported directly from customer to customer, and not just from port to port. The transport by water is just a link in the chain of transport. Container vessels have grown from a capacity of 1500 TEU (1966) to approximately 8000 (2002) (van Dokkum 2003).

The sizes of containers vary. The ISO standards distinguish the TEU and the FEU, which may differ in height.

TEU = 20 ft equivalent unit. The nominal length of these containers is 6.10 m. The actual length is 38 mm shorter, leaving some space between the containers.

FEU = 40 ft equivalent unit. The nominal length of these containers is 12.20 m.

Today, about 90% of non-bulk cargo worldwide is transported by container ships, and the largest modern container ships can carry over 21,000 TEU. Container ships now rival crude oil tankers and bulk carriers as the largest commercial seaborne vessels.

Container ships are distinguished into seven major size categories: small feeder,

feeder, feedermax, Panamax, Post-Panamax, New Panamax, and ultra-large. As of December 2012, there were 161 container ships in the VLCS class (Very Large Container Ships, more than 10,000 TEU), and 51 ports in the world can accommodate them (Cudahy 2006).

The size of a Panamax vessel is limited by the original Panama Canal's lock chambers, which can accommodate ships with a beam of up to 32.31 m, a length overall of up to 294.13 m, and a draft of up to 12.04 m. The Post-Panamax category has historically been used to describe ships with a molded breadth over 32.31 m; however, the Panama Canal expansion project has caused some changes in terminology (Haylar 2003). The New Panamax category is based on the maximum vessel size that is able to transit a new third set of locks, which opened in June 2016; it can be found in Fig. 2. The third set of locks were built to accommodate a container ship with a length overall of 366 m (1,201 ft), a maximum width of 49 m (161 ft), and tropical freshwater draft of 15.2 m (50 ft). Such a vessel, called New Panamax class, is wide enough to carry 19 rows of containers, can have a total capacity of approximately 12,000 TEU, and is comparable in size to a capesize bulk carrier or a Suezmax tanker.

Transport Ship,

Fig. 2 MV Mærsk Mc-Kinney Møller. (<https://en.wikipedia.org/>)



Tankers are cargo ships for the transport of fluids, such as crude oil, petroleum products, liquefied petroleum gas (LPG), liquefied natural gas (LNG), and chemicals, also vegetable oils, wine, and other food – the tanker sector comprises one third of the world tonnage. Major types of tankship include the oil tanker, the chemical tanker, and gas carrier. Tankers also carry commodities such as vegetable oils, molasses, and wine. In the US Navy and Military Sealift Command, a tanker used to refuel other ships is called an oiler (or replenishment oiler if it can also supply dry stores), but many other navies use the terms tanker and replenishment tanker (Wiltshire 2008).

In 1954, Shell Oil developed the average freight rate assessment (AFRA) system, which classifies tankers of different sizes. To make it an independent instrument, Shell consulted the London Tanker Brokers' Panel (LTBP). At first, they divided the groups as General Purpose for tankers under 25,000 tons deadweight (DWT), Medium Range for ships between 25,000 and 45,000 DWT, and Large Range for the then-enormous ships that were larger than 45,000 DWT (Haylar 2003). The very large crude carrier MV Sirius Star is shown in Fig. 3. The ships became larger during the 1970s, and the list was extended, where the tons are long tons:

10,000–24,999 DWT: Small tanker
 25,000–34,999 DWT: Intermediate tanker
 35,000–44,999 DWT: Medium Range 1 (MR1)
 45,000–54,999 DWT: Medium Range 2 (MR2)
 55,000–79,999 DWT: Large Range 1 (LR1)
 80,000–159,999 DWT: Large Range 2 (LR2)
 160,000–319,999 DWT: Very Large Crude Carrier (VLCC)
 320,000–549,999 DWT: Ultra Large Crude Carrier (ULCC)

Refrigerated ships (usually called reefers) are cargo ships typically used to transport perishable commodities which require temperature-controlled transportation, mostly fruits, meat, fish, vegetables, dairy products, and other food-stuffs. The reefer ship Dole Honduras unloading bananas is a kind of typical ship, as can be found in Fig. 4.

Reefer ships may be categorized into three types:

1. Side-door vessels have watertight ports on the ship's hull, which open into a cargo hold. Elevators or ramps leading from the quay serve as loading and discharging access for the forklifts or conveyors. Inside these access ports or side doors, pallet lifts or another series of conveyors bring the cargo to the respective

Transport Ship,

Fig. 3 The very large crude carrier MV Sirius Star. (<https://en.wikipedia.org/>)



Transport Ship,

Fig. 4 The reefer ship Dole Honduras unloading bananas. (<https://en.wikipedia.org/>)



decks. This special design makes the vessels particularly well suited for inclement weather operations as the tops of the cargo holds are always closed against rain and sun.

2. Conventional vessels have a traditional cargo operation with top opening hatches and cranes/derricks. On such ships, when facing wet weather, the hatches need to be closed to prevent heavy rain from flooding the holds. Both above ship types are well suited for the handling of palletized and loose cargo.
3. Refrigerated container ships are specifically designed to carry containerized unit loads where each container has its individual refrigerated unit. These containers are nearly always 20 ft equivalent units (often called TEU) that are the size of “standard” cargo containers that are loaded and unloaded at container terminals and aboard container ships. These ships differ from conventional container ships in their design, power generation, and electrical distribution equipment. They need provisions made for powering each container’s cooling system. Because of their ease of loading and unloading cargo, many container ships are now being built or redesigned to carry refrigerated containers (Kohli 2000).

Roll-on/roll-off (RORO or ro-ro) ships are vessels designed to carry wheeled cargo, such as cars, trucks, semitrailer trucks, trailers, and railroad cars, that are driven on and off the ship on

their own wheels or using a platform vehicle, such as a self-propelled modular transporter (Emi 2006).

Types of RORO vessels include ferries, cruiseferries, cargo ships, barges, and RORO service for air deliveries. New automobiles that are transported by ship are often moved on a large type of RORO called a pure car carrier (PCC) or pure car/truck carrier (PCTC).

Elsewhere in the shipping industry, cargo is normally measured by the metric ton, but RORO cargo is typically measured in lanes in meters (LIMs). This is calculated by multiplying the cargo length in meters by the number of decks and by its width in lanes (lane width differs from vessel to vessel, and there are several industry standards). On PCCs, cargo capacity is often measured in RT or RT43 units (based on a 1966 Toyota Corona, the first mass-produced car to be shipped in specialized car carriers and used as the basis of RORO vessel size. 1 RT is approximately 4 m of lane space required to store a 1.5 m wide Toyota Corona) or in car-equivalent units (CEU). Figure 5 shows a pure car carrier ship’s starboard side.

The largest RORO passenger ferry is MS Color Magic, a 75,100 GT cruise ferry that entered service in September 2007 for Color Line. Built in Finland by Aker Finnyards, it is 223.70 m (733 ft 11 in) long and 35 m (114 ft 10 in) wide and can carry 550 cars or 1270 lane meters of cargo.

Transport Ship, Fig. 5 A pure car carrier ship's starboard side. (<https://en.wikipedia.org/>)



Transport Ship, Fig. 6 The largest car/truck carrier – MV Tønsberg. (<https://en.wikipedia.org/>)

The RORO passenger ferry with the greatest car-carrying capacity is Ulysses, owned by Irish Ferries. Ulysses entered service on 25 March 2001 and operates between Dublin and Holyhead. The 50,938 GT ship is 209.02 m (685 ft 9 in) long

and 31.84 m (104 ft 6 in) wide and can carry 1342 cars/4101 lane meters of cargo (Fig. 6).

The PCTC has liftable decks to increase vertical clearance, as well as heavier decks for “high-and-heavy” cargo. A 6,500-unit car ship,

with 12 decks, can have three decks which can take cargo up to 150 short tons (136 t; 134 long tons) with liftable panels to increase clearance from 1.7 to 6.7 m (5 ft 7 in to 22 ft 0 in) on some decks. Lifting decks to accommodate higher cargo reduces the total capacity.

These kinds of vessels perform a usual speed of 16 knots at eco-speed, while at full speed can achieve more than 19 knots.

With the building of Wallenius Wilhelmsen Logistics's 8,000-CEU car carrier *Faust* out of Stockholm in June 2007, car carriers entered a new era of the large car and truck carrier (LCTC) (Llovio 2007). Currently, the largest are Hoegh Autoliners Six Horizon class vessels with capacity of 8,500 CEU each.

The car carrier *Auriga Leader*, belonging to Nippon Yusen Kaisha, built in 2008 with a capacity of 6,200 cars, is the world's first partially solar-powered ship.

Ocean liner is the passenger ship primarily used as a form of transportation across seas or oceans. Liners may also carry cargo or mail and may sometimes be used for other purposes (e.g., for pleasure cruises or as hospital ships).

Ocean liners are usually strongly built with a high freeboard to withstand rough seas and adverse conditions encountered in the open ocean. Additionally, they are often designed

with thicker hull plating than is found on cruise ships and have large capacities for fuel, food, and other consumables on long voyages.

The first ocean liners were built in the mid-nineteenth century. Technological innovations such as the steam engine and steel hull allowed larger and faster liners to be built, giving rise to a competition between world powers of the time, especially between the United Kingdom and Germany. Once the dominant form of travel between continents, ocean liners were rendered largely obsolete by the emergence of long-distance aircraft after World War II. Advances in automobile and railway technology also played a role. By 2015, RMS *Queen Mary 2* is the only ship still in service as an ocean liner, the RMS *Queen Elizabeth 2* having retired a few years prior; it can be seen in Fig. 7.

Ferries are a form of transport, usually a boat or ship, but also other forms, carrying (or ferrying) passengers and sometimes their vehicles. Ferries are also used to transport freight (in lorries and sometimes unpowered freight containers) and even railroad cars. Most ferries operate on regular, frequent, return services. A foot-passenger ferry with many stops, such as in Venice, is sometimes called a water-bus or water taxi. Ferries form a part of the public transport systems of many



Transport Ship, Fig. 7 Queen Mary 2–05. (<https://en.wikipedia.org/>)

Transport Ship,

Fig. 8 The Staten Island Ferry – commuters between Manhattan and Staten Island in NYC.

(<https://en.wikipedia.org/>)



waterside cities and islands, allowing direct transit between points at a capital cost much lower than bridges or tunnels.

Ferry boats often dock at specialized facilities designed to position the boat for loading and unloading, called a ferry slip. If the ferry transports road vehicles or railway carriages, there will usually be an adjustable ramp called an apron that is part of the slip. In other cases, the apron ramp will be a part of the ferry itself, acting as a wave guard when elevated and lowered to meet a fixed ramp at the terminus – a road segment that extends partially underwater.

Some of world's busiest ferry routes include the Star Ferry in Hong Kong and the Staten Island Ferry in New York City. The Staten Island Ferry is shown in Fig. 8.

Design Consideration

Bulk carriers are usually discharged by grabs or by suction pipes. Pouring the cargo through a shooter or via a conveyor belt doses the loading. Bulk carriers have large upper and lower ballast tanks to give the empty vessel enough draft and a better behavior while in transit (van Dokkum 2003). Ship transporting ore have a special design. Ore is very heavy (stowage factor is approximately $0.5 \text{ m}^3/\text{t}$), and thus ships only need small holds to be loaded completely. To prevent a too large stability, the holds must not be situated too low or too

close to the sides of the ships. Some bulk carriers can also function as a tanker.

Tankers are a relatively new concept, dating from the later years of the nineteenth century. Before this, technology had simply not supported the idea of carrying bulk liquids. The market was also not geared toward transporting or selling cargo in bulk; therefore, most ships carried a wide range of different products in different holds and traded outside fixed routes. Liquids were usually loaded in casks – hence the term “tonnage” refers to the volume of the holds in terms of how many tuns or casks of wine could be carried. Even potable water, vital for the survival of the crew, was stowed in casks. Carrying bulk liquids in earlier ships posed several problems:

1. Loading and discharging: Bulk liquids must be pumped – The development of efficient pumps and piping systems was vital to the development of the tanker. Steam engines were developed as prime movers for early pumping systems. Dedicated cargo handling facilities were now required ashore too – As was a market for receiving a product in that quantity. Casks could be unloaded using ordinary cranes, and the awkward nature of the casks meant that the volume of liquid was always relatively small – Therefore keeping the market more stable.

2. Free surface effect: A large body of liquid carried aboard a ship will impact on the ship's stability, particularly when the liquid is flowing around the hold or tank in response to the ship's movements. The effect was negligible in casks but could cause capsizing if the tank extended the width of the ship, a problem solved by extensive subdivision of the tanks.

Key Applications

Maritime transport can be realized over any distance by boat, ship, sailboat, or barge, over oceans and lakes, through canals, or along rivers. While extensive inland shipping is less critical today, the major waterways of the world including many canals are still very important and are integral parts of worldwide economies. Virtually any material can be moved by water; however, water transport becomes impractical when material delivery is time-critical such as various types of perishable produce. However, water transport is highly cost-effective with regular schedulable cargoes, such as transoceanic shipping of consumer products – and especially for heavy loads or bulk cargos, such as coal, coke, ores, or grains. Arguably, the industrial revolution took place best where cheap water transport by canal, navigations, or shipping by all types of watercraft on natural waterways supported cost-effective bulk transport.

Recreational or educational craft still use wind power, while some smaller craft use internal combustion engines to drive one or more propellers or, in the case of jet boats, an inboard water jet. In shallow draft areas, such as the Everglades, some craft, such as the hovercraft, are propelled by large pusher-prop fans.

Cross-References

- [FLNG](#)
- [FPSO](#)
- [Luxury Cruises](#)
- [Ship Electrical Design](#)
- [Ship Machinery Design](#)

- [Ship Overall Design](#)
- [Ship Structural Design](#)
- [Ship-Fitting Design](#)
- [Surface Warships](#)

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Transverse Stability

- [AUV/ROV/HOV Stability](#)

Trellis-Coded Modulation (TCM)

- [Underwater Acoustic Communication](#)

Triaxial Compression

- [Shallow Foundations](#)

Triaxial Tension

- [Shallow Foundations](#)

Tributyltin (TBT)

► [Net Structures: Biofouling and Antifouling](#)

TRITON: Triangle Trans-Ocean Buoy Network

► [Tropical Atmosphere Ocean \(TAO\) Buoy](#)

Tropical Atmosphere Ocean (TAO) Buoy

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Synonyms

GTMBA: Global tropical moored buoy array;
TAO: Tropical atmosphere ocean buoy; TRITON:
Triangle trans-ocean buoy network

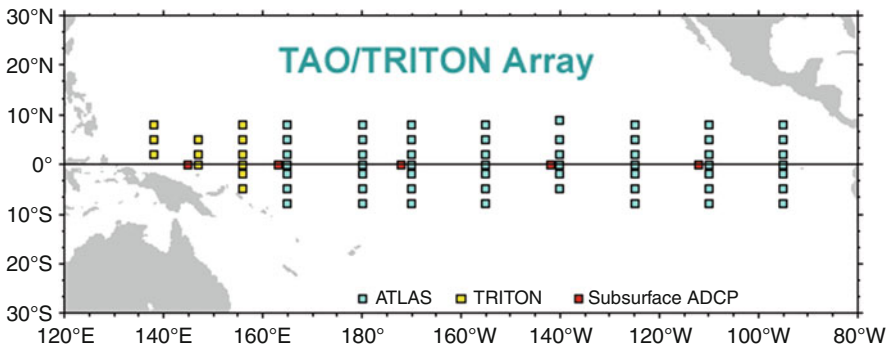
Definition

The TAO buoys are the buoys especially designed for studying the macro phenomenon called El Niño, and it was motivated by the 1982–1983 El Niño events. The Tropical Atmosphere Ocean (TAO) project is a major international effort that instrumented the entire tropical Pacific Ocean with approximately 70 deep ocean moorings/buoys. The development of the TAO array started in 1985 and led by the TAO Project Office of the Pacific Marine Environmental Laboratory (PMEL), the full array of 70 moorings was completed in 1994 (see Fig. 1) (tao.ndbc.noaa.gov/proj_overview/proj_overview_ndbc 2018). The TAO array was renamed as the TAO/TRITON array on 1 January 2000.

The array is a major component of the El Niño/Southern Oscillation (ENSO) Observing System, the Global Climate Observing System (GCOS), and the Global Ocean Observing System (GOOS).

Scientific Fundamentals

The TAO array consists of several types of moorings among which the Next Generation ATLAS (Autonomous Temperature Line Acquisition System) moorings are the dominant ones at present. Prior to the Next Generation ATLAS moorings, the standard ATLAS moorings were used as the



Tropical Atmosphere Ocean (TAO) Buoy, Fig. 1 TAO/TRITON array (tao.ndbc.noaa.gov/proj_overview/proj_overview_ndbc 2018)

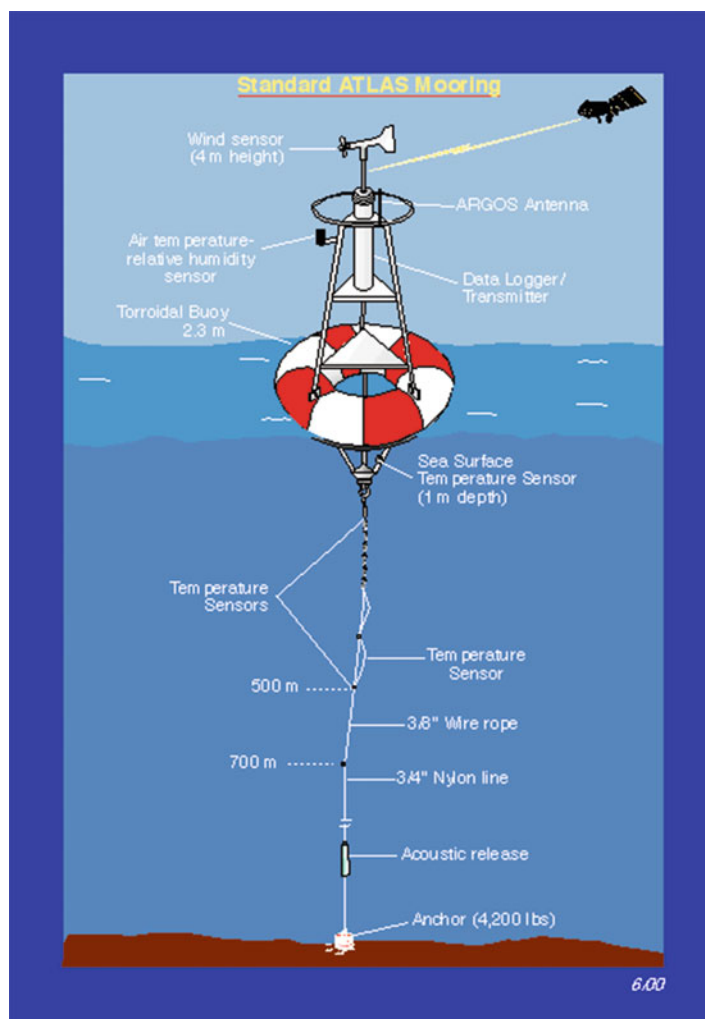
dominant ones for the array. A notable improvement to the Next Generation ATLAS moorings were the inductively coupled sensors for subsurface data which simplified fabrication, and therefore, eliminated the thermistor cable and its labor-intensive assembly and deployment procedures (Milburn et al. 1996).

In addition to the ATLAS moorings, other moorings such as T-FLEX moorings, TAO Refresh moorings, Acoustic Doppler Current Profiler moorings, TRITON moorings, and BAILONG moorings are also used in TAO array. In this section, various TAO/TRITON Mooring types will be introduced.

Historical Standard ATLAS Moorings

After testing and deployment of prototype ATLAS moorings, the first elements of the large-scale monitoring TAO array were deployed in the eastern Pacific in November 1984. The full TAO array was eventually completed in December 1994. The standard ATLAS mooring (shown in Fig. 2) had a design lifetime of one year, and the system proved to be robust and reliable. Over 500 Standard ATLAS moorings were deployed between 1984 and 2001. The final standard ATLAS was recovered in November 2001 and Next Generation ATLAS moorings are now used exclusively in the TAO array.

Tropical Atmosphere Ocean (TAO) Buoy,
Fig. 2 Standard ATLAS mooring (pmel.noaa.gov/gtmb 2018)



Standard ATLAS moorings measured surface winds, air temperature, relative humidity, sea surface temperature, and ten subsurface temperatures from a 500 m long thermistor cable. Daily-mean data were telemetered to shore in near real-time via NOAA's polar-orbiting satellites and Service Argos. A small subset of hourly values (2–3 per day) coinciding with satellite passes were also transmitted in real time. Hourly values of surface data were internally recorded and available after mooring recovery (pmel.noaa.gov/gtmb 2018).

The Standard ATLAS buoy is a 2.3 m diameter toroid fabricated with fiberglass over a foam core with a simple aluminum tower and a rigid stainless-steel bridle. Nonrotating 0.96 cm diameter 3×19 wire rope is used in the upper 500 m of the mooring. Eight-strand plaited nylon line is used for the compliant member of the mooring with an acoustic release above a 2000 kg railroad wheel anchor and a typical mooring scope of 0.98. An electrical cable with breakouts is clamped parallel to the wire rope for telemetering data from subsurface sensors to the buoy. Active sensors transmit data in a simple time-multiplexed scheme using frequency encoding on a common data/power line. Electronics and batteries are housed in an aluminum tube on the tower of the buoy with meteorological sensors, sea surface temperature, and the subsurface cable connected to the tube through a cast aluminum top cap with marine penetrators. This “low-tech” approach has kept the cost of an individual system down, enabling the array to be completed and maintained on a relatively small budget (tao.ndbc.noaa.gov/proj_overview/pubs/mil96paper 2018).

The sensor systems and sampling schemes on the Standard ATLAS system have been standardized as described in Mangum et al. (1995).

The system measures vector-averaged winds, air temperature, relative humidity, sea surface temperature (SST), 10 subsurface temperatures to a depth of 500 m, and pressure of the lower two temperature sensors. The moorings are designed to be recovered and redeployed after a nominal 1-year period. Data is transmitted daily via Service Argos in three 32-byte buffers and is available to the community in near real time via

the GTS, the World Wide Web, and anonymous FTP.

Next Generation ATLAS Moorings

The first Next Generation systems were deployed in the array in May 1996. The transition to Next Generation systems throughout the array was completed in November 2001. Compared to the Standard ATLAS moorings, the Next Generation ATLAS moorings have the following advantages (tao.ndbc.noaa.gov/proj_overview/pubs/mil96paper 2018):

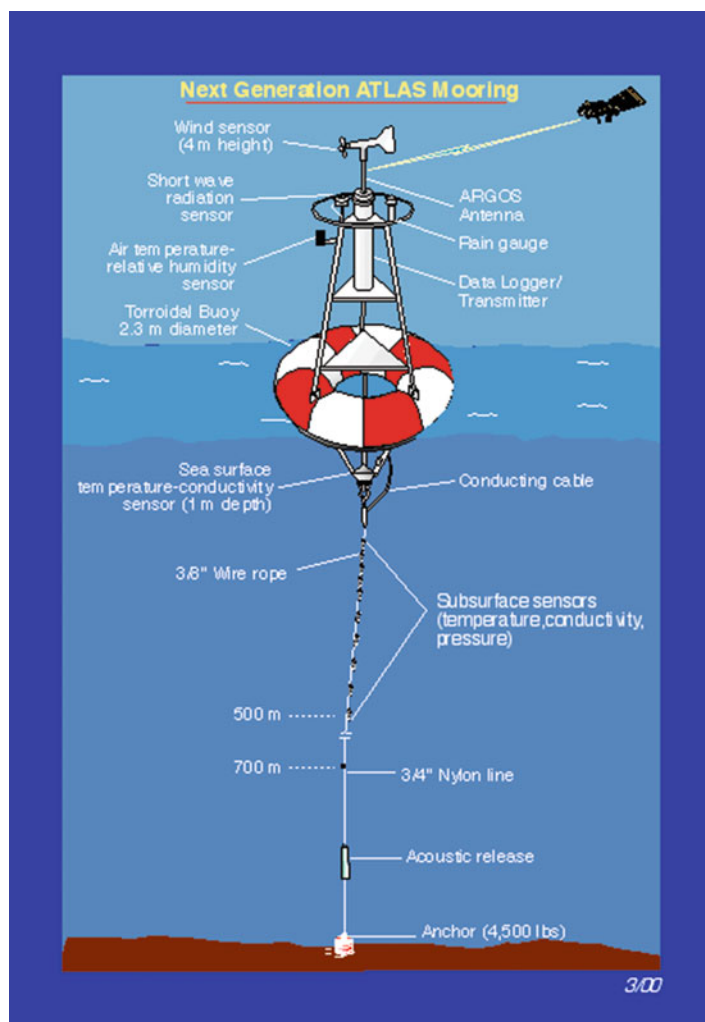
- Improved data quality
- Added new sensors
- Increased temporal resolution of internally recorded data
- Improved reliability to extend system life
- Simplified fabrication procedures
- Reduced costs

The Next Generation ATLAS system shown in Fig. 3 uses the same basic hardware system as described above, but incorporates new electronics, added sensors, and can be deployed with a slack mooring without an acoustic release. Taut moorings maintain the sensor locations on the wire rope at or near the desired depths, but a slack mooring will cause large vertical excursions of the lower sensors. Accurate pressure measurements in the lower modules will be used to map the positions of the temperature measurements using a simple cable model. The longer scope in the moorings will lower the tension in the line and ease the requirements for accurate depth and line-length determinations necessary for taut moorings. Also, the lower part of the moorings will be left in place when the wire rope and sensors are recovered and replaced on service visits, saving the expense of anchors, hardware, and acoustic releases.

The most obvious and significant improvement in the ATLAS system is the incorporation of inductively coupled sensors for subsurface data. The sensors merely clamp onto the wire rope strength member that serves as one of the inductive elements. This simplifies fabrication, since it will no longer be necessary to manufacture the

Tropical Atmosphere Ocean (TAO) Buoy,

Fig. 3 Next Generation ATLAS mooring (pmel.noaa.gov/gtmmba 2018)



sensor cable with its labor-intensive assembly and deployment procedures. Addressable modules on the cable allow the system to be expanded for new sensors by adding the appropriate hardware and software interfaces.

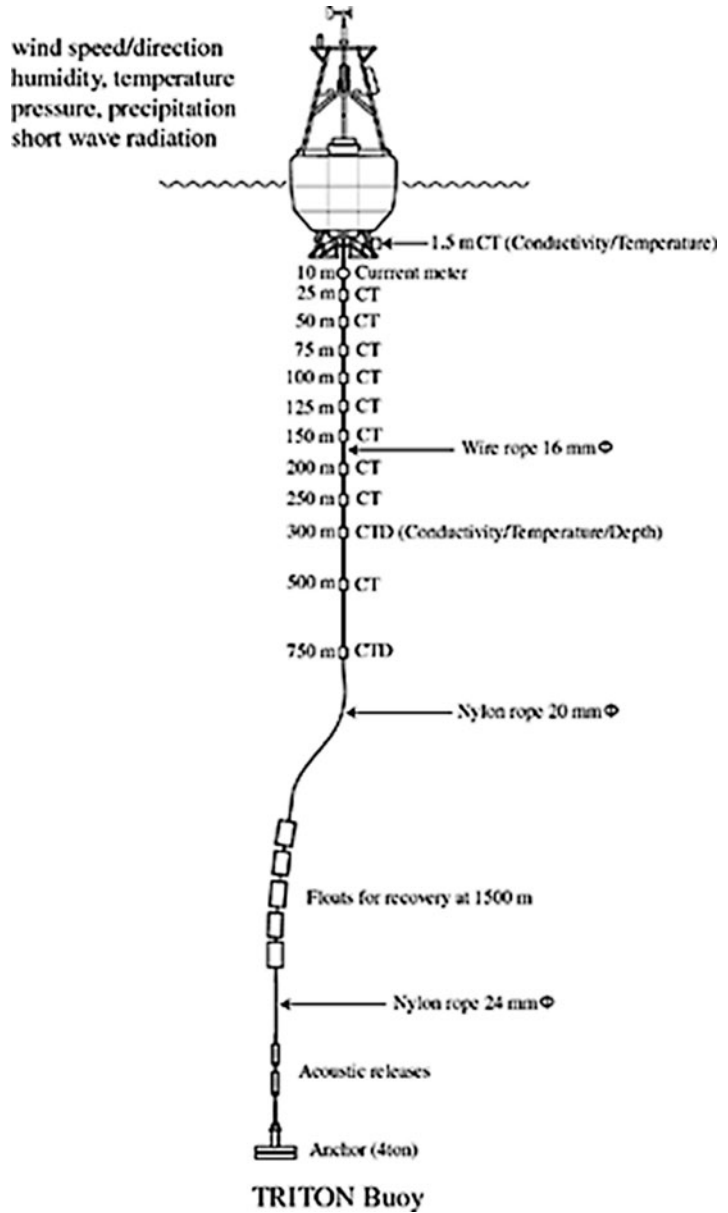
TRITON Moorings

In October 1999, responsibility for TAO moorings along and to the west of 156E was assumed by the Japan Agency for Marine-Earth Science and Technology (JAMSTEC) which has deployed Triangle Trans-Ocean Buoy Network (TRITON) buoys at those locations. Data from both ATLAS and TRITON moorings are merged in a common data base and available from both PMEL and

JAMSTEC. M-TRITON moorings, a smaller TRITON mooring also developed by JAMSTEC are deployed in RAMA mooring (pmel.noaa.gov/gtmmba 2018).

The TRITON moorings consist of 12 conductivity and temperature sensors that are installed at depths of 1.5 m, 25 m, 50 m, 75 m, 100 m, 125 m, 150 m, 200 m, 250 m, 300 m, 500 m, and 750 m (Ando et al. 2005). A single current meter is at 10 m, and surface meteorological sensors are mounted on the tower of the mooring. Other sensors on TRITON moorings measure temperature, salinity, wind speed and direction, air temperature and relative humidity, short-wave radiation, and rainfall. Figure 4 shows the TRITON Buoy mooring configuration.

Tropical Atmosphere Ocean (TAO) Buoy,
Fig. 4 TRITON Buoy
 (jamstec.go.jp 2018)



T-FLEX Moorings

“Tropical Flex (T-FLEX)” moorings were developed by the Engineering Division at PMEL to replace the Next Generation ATLAS moorings in RAMA and PIRATA as components become obsolete mooring (pmel.noaa.gov/gtmba 2018). T-Flex moorings allow greater flexibility with respect to number and types of instruments that may be deployed, and utilize more commercial off-the-shelf instrumentation.

T-Flex electronics are housed in drop-in canisters for deployment in a modified ATLAS toroid (filled toroid with 2 wells for canisters; 1 for electronics, the second for an external battery). An iridium modem and antenna are integrated into the T-flex canister. There is also an RF modem on the buoy to enable ship to buoy communications. T-Flex tower modifications include a T-cup handle, cable runs around the lower bars, and an additional mounting pad on top for optional

sensors. Eight (8) T-Flex moorings were deployed alongside ATLAS moorings in RAMA and PIRATA between March 2011 and December 2015. A NOAA Technical Memorandum comparing the system performance and data equivalency is in preparation. The first stand-alone T-Flex was deployed in August, 2015 at 4S 81E.

Hourly values of all measurements are returned to shore via Iridium satellite, formatted into WMO BUFR code, and placed on the GTS by PMEL. Higher resolution data are logged internally, and downloaded once the mooring is recovered.

TAO Refresh Moorings

The TAO Array has transitioned from the National Oceanic and Atmospheric Administration's (NOAA) research arm (Pacific Marine Environmental Laboratory, PMEL) to its operational arm within the National Weather Service at the National Data Buoy Center (NDBC). NDBC's transition strategy had two motives: First, to maintain continuity of the existing observations for the climate record and, second (but equally as important) to ensure its future contributions and survival which will be accomplished by a comprehensive equipment upgrade, known as the TAO Technology Refresh (TAO Refresh). TAO Refresh was always an important component of NDBC's transition of the Array. Presently, the Array uses the Next Generation ATLAS mooring system, designed in the 1990s and the deployments completed in 2001, now termed TAO Legacy (tao.ndbc.noaa.gov/proj_overview/mooring_ndbc 2018).

As part of a comprehensive test plan to ensure continuity of the climate records, NDBC conducted laboratory tests and several prototype TAO Refresh systems deployed in the Gulf of Mexico and the Pacific TAO array. Currently eight (8) TAO Refreshed buoys are deployed in the Pacific for in-situ testing with TAO Legacy. The intention is to reduce to 5 TAO Refreshed buoys for extended testing. Initial evaluations indicate good agreement between TAO Legacy and TAO Refresh. These refresh buoys are deployed adjacent to legacy buoys, which are considered a co-located buoy. The refresh buoy data will be available via the Global Telecommunication System (GTS) under the SXPA01

KWNB header in World Meteorological Organization (WMO) FM-18 BUOY alphanumeric formats. The Refreshed buoy is designed to meet requirements for National Weather Service, climate research, and international Global Ocean Observing System (GOOS) and Global Climate Observing System (GCOS) purposes while increasing the real-time data available for climate analysis, forecasts, and numerical model predictions.

Acoustic Doppler Current Profiler Moorings

Subsurface Acoustic Doppler Current Profiler (ADCP) moorings are deployed at a few locations to provide velocity profiles the upper 200–300 m of the water column. ADCP moorings are mostly deployed at equatorial locations where geostrophy breaks down and direct velocity measurements are required. ADCPs are usually deployed in tandem with nearby surface moorings, which may also measure velocity as a backup to the ADCP and/or to provide velocity data near the surface where acoustic back scatter from the sea surface interferes with the ADCP signal. The vertical resolution of the ADCP is about 8 m. ADCP data are available only after the subsurface moorings are recovered mooring (pmel.noaa.gov/gtmmba 2018).

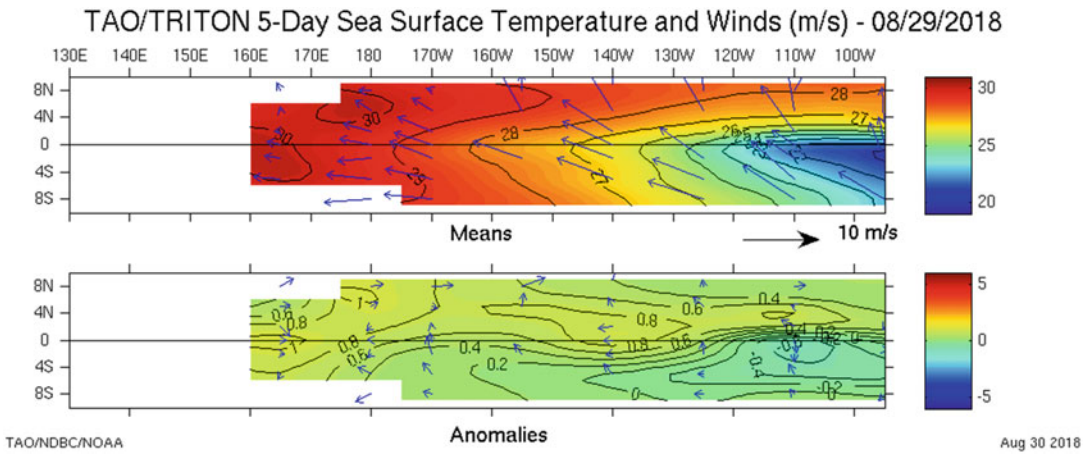
BaiLong Moorings

China's First Institute of Oceanography (FIO) has introduced ATLAS-like moorings, known as BaiLong, one of which occupies a RAMA site. Data from this mooring is telemetered to and quality controlled by FIO and made available on the GTMBA data web page.

With the above moorings, now NDBC provides the measured data and analysis on daily basis from the TAO buoys. Figure 5 shows the plots of the sea surface temperature and wind measurements from the TAO/TRITON buoys mooring (pmel.noaa.gov/gtmmba 2018).

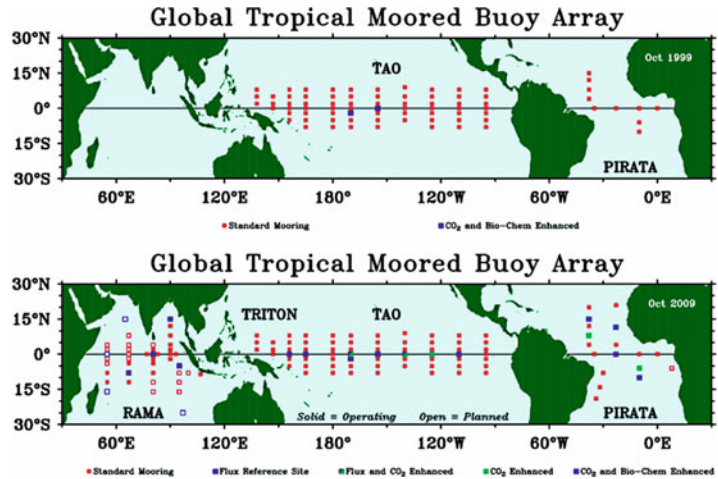
Key Applications

The TAO/TRITON array was extended to the Global Tropical Moored Buoy Array (GTMBA,



Tropical Atmosphere Ocean (TAO) Buoy, Fig. 5 Sea surface temperature and wind measurements (tao.ndbc.noaa.gov/refreshed/index 2018)

Tropical Atmosphere Ocean (TAO) Buoy, Fig. 6 Full View of the Global Tropical Moored Buoy Array (tao.ndbc.noaa.gov/refreshed/index 2018)



see Fig. 6) after mid-1990, with the additions of the Prediction and Research Moored Array in the Tropical Atlantic (PIRATA), and the Research Moored Array for African-Asian-Australian Monsoon Analysis and Prediction (RAMA) in the Indian Ocean. Now the GTMBA is a global system with its interest spanning intraseasonal-to-decadal and longer timescales on global basis, including:

- El Niño/Southern Oscillation and its decadal modulation in the Pacific
- The meridional gradient mode and equatorial warm events in the Atlantic

- The Indian Ocean Dipole
- The mean seasonal cycle, including the Asian, African, Australian, and American monsoons
- The intraseasonal Madden-Julian Oscillation, which originates in the Indian Ocean but affects all three ocean basins
- Trends that may be related to global warming

The Prediction and Research Moored Array in the Tropical Atlantic (PIRATA) was designed to study ocean-atmosphere interactions in the tropical Atlantic that affect regional weather and climate variability on seasonal, interannual, and longer time scales. Ocean-atmosphere

interactions in this region influence the development of droughts, floods, severe tropical storms, and hurricanes, with impacts felt by millions of people in the Americas and Africa. PIRATA was first established in the mid-1990s and has undergone expansions and enhancements since 2005 to improve its utility for ocean and climate research and forecasting. PIRATA is supported by France (IRD, Météo-France, CNRS and IFREMER), Brazil (INPE and DHN), and the USA (NOAA).

The Research Moored Array for African-Asian-Australian Monsoon Analysis and Prediction (RAMA) was designed to study the Indian Ocean's role in the monsoons. One third of the world population depends on monsoon driven rainfall for agricultural production, so improving our understanding and ability to predict the monsoons has been a long-standing objective of the international scientific community. RAMA, the newest tropical buoy array, was initiated in the traditionally data sparse Indian Ocean in 2004 following deployment of successful pilot scale arrays by Japan and India. RAMA has since grown through the formation of new partnerships that at present include Indonesia, China, the USA, and the Bay of Bengal Large Marine Ecosystem (BOBLME) program.

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TRYGGHET TILLIT SERVICE (TTS)

- [Block Assembly Line](#)

TSDM – Tow-Sled Type Driving Mechanism

- [Underwater Driving Subsystem](#)

Tsunami Warning Buoy

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Definition

Tsunami Warning Buoy is a floating device designed as part of the Tsunami Warning System. Paired with a sensor on the ocean floor that watches for seismic activity, the buoy is a communications hub that transmits tsunami data to an international network and helps give affected populations a few precious minutes to prepare.

Synonyms

[DART: Deep-Ocean Assessment and Reporting of Tsunamis](#); [TDB: Tsunami Detection Buoy](#); [TWS Tsunami Warning System](#)

Scientific Fundamentals

Introduction

Tsunamis are huge waves of water caused by displacement during the natural disaster such as earthquakes, landslides, volcano eruptions, or meteorite impacts, and their force can cause massive damage, about 86% of tsunamis are caused

by earthquakes. Tsunamis travel at very fast speeds, and they often exceed 160 km per hour and can reach up to 800 km per hour (in the deep ocean). As the waves approach the coast, their speed decreases and their amplitude increases. A tsunami that is just a meter in height in the deep ocean can grow to tens of meters waves at the shoreline, which can devastate the coastlines of entire ocean basins, as happened with the 2004 tsunami which affected areas from Somalia to Indonesia. About 99% of tsunami fatalities occur within 250 km of the tsunami's origin.

From an initial tsunami generating source area, waves travel outward in all directions much like the ripples caused by throwing a rock into a pond. As these waves approach coastal areas, the time between successive wave crests varies from 5 to 90 min. The first wave is usually not the largest in the series of waves, nor is it the most significant. Furthermore, one coastal community may experience no damaging waves while another, not that far away, may experience destructive deadly waves. Depending on several factors, some low-lying areas could experience severe inland inundation of water and debris of more than 300 meters.

The first early tsunami warning systems developed in Hawaii, but it was not until the late twentieth century that today's tsunami warning system was developed and launched off the northwest coast of the USA. The National Oceanic and Atmospheric Administration (NOAA) Tsunami Warning Centre (website see (The National Oceanic 2018)) is a scientific agency that tries to limit the damage that tsunamis can do by strategically placing buoys along the areas where there is a history of tsunamis. These booths provide early warnings of possible tsunamis. Buoys such as DART (Deep-ocean Assessment Reading Tsunamis) were originally developed by NOAA to detect tsunamis. These stations are placed in regions with a history of destructive tsunamis.

NOAA operates these tsunami warning centers, namely National Tsunami Warning Center (NTWC, website see (National Tsunami Warning Center 2018)), NTWC responsible for coastal waters from Alaska to California, including Washington), Pacific Tsunami Warning Center

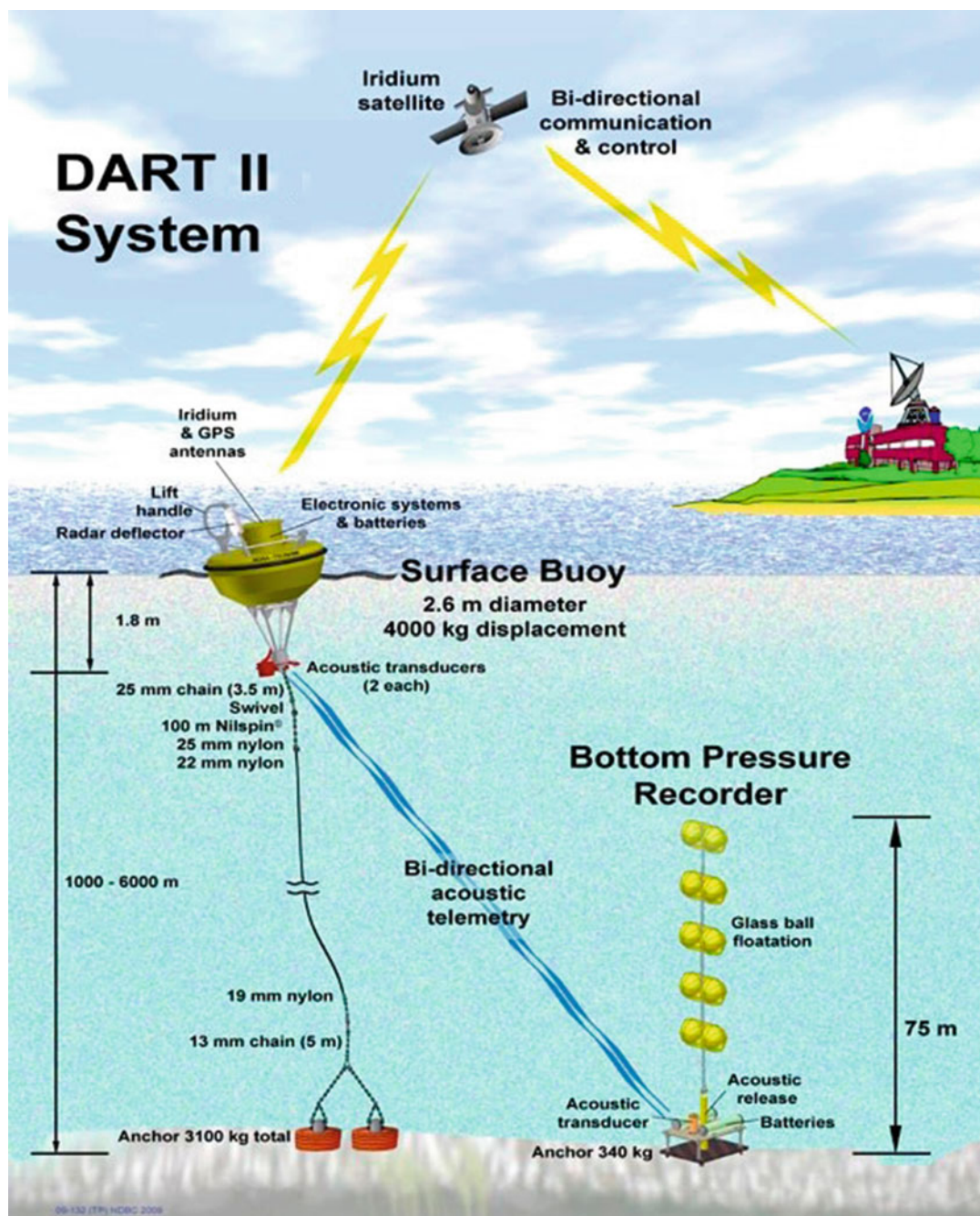
(PTWC, website see (Pacific Tsunami Warning Center 2018)), PTWC responsible for all American interests in the Pacific, including Hawaii), and International Tsunami Warning Center (ITWC, website see (International Tsunami Warning Center 2018)), ITWC responsible for the other 25 member nations in the Pacific Ocean Basin).

International Tsunami Information Center (ITIC) has published a 12-page brochure, *Tsunami The Great Waves* (Tsunami 2018), which provides information on what tsunamis are, how fast and how big they can be, and what causes them, and describes programs undertaken to mitigate this hazard, including the development of tsunami warning centers, research programs, and safety rules describing what to do when a tsunami attacks your coastline. The English version was last updated and revised in December, 2014. It is also available in Spanish, French, and Chinese.

Scientific Fundamentals of Tsunami Warning Buoy

Usually the tsunami buoys are located near some trench or borders between plates, where most of tsunami events in the world were generated. The determination of location follows some important rules. One of the most is that the tsunami buoys can record real time typical tsunami wave in no more than half an hour after earthquake-generated tsunami occurred. There are two main functions for tsunami detection buoy, one is to detect whether the tsunami is generated and record typical tsunami wave in 15–30 min and show after earthquake occurred; the other is that the real-time tsunami wave data can be assimilated into forecasting system to improve the accuracy in tsunami forecasts.

A tsunami warning buoy has two components: the surface buoy and the underwater sensor (see Fig. 1). The underwater sensor has two modes, the standard mode and the event mode. The underwater sensor is usually at standard mode most of the time. When on-board software identifies a possible tsunami, the station leaves standard mode and begins transmitting in event mode. Here, we will take the commercial product DART II (Deep-ocean Assessment and Reporting of Tsunamis) systems to explain the scientific fundamental of



Tsunami Warning Buoy, Fig. 1 DART system configuration: a diagram of the Dart II system. (Figure from Deep-ocean Assessment and Reporting of Tsunamis (DART) 2018)

the tsunami warning system; more details about the systems can be found in website (Deep-ocean Assessment and Reporting of Tsunamis (DART) 2018).

The systems (as shown in Fig. 1) consist of an anchored seafloor bottom pressure recorder (BPR) and a companion-moored surface buoy for real-time communications (Gonzalez et al. 1998). An

acoustic link transmits data from the BPR on the seafloor to the surface buoy.

The BPR collects temperature and pressure at 15-s intervals. The pressure values are corrected for temperature effects and the pressure converted to an estimated sea-surface height (height of the ocean surface above the seafloor) by using a constant 670 mm/psia. The system has two data reporting modes, standard and event. The system operates routinely in standard mode, in which four-spot values (of the 15-s data) at 15-min intervals of the estimated sea surface height are reported at scheduled transmission times. When the internal detection software (Deep-ocean Assessment and Reporting of Tsunamis (DART) Tsunami 2018) identifies an event, the system ceases standard mode reporting and begins event mode transmissions. In event mode, 15-s values are transmitted during the initial few minutes, followed by 1-min averages. Event mode messages also contain the time of the initial occurrence of the event. The system returns to standard transmission after 4 h of 1-min real-time transmissions if no further events are detected. The link (Deep-ocean Assessment and Reporting of Tsunamis (DART II) 2018) shows an animation for us to understand how the system works.

Tsunami Detection Algorithm

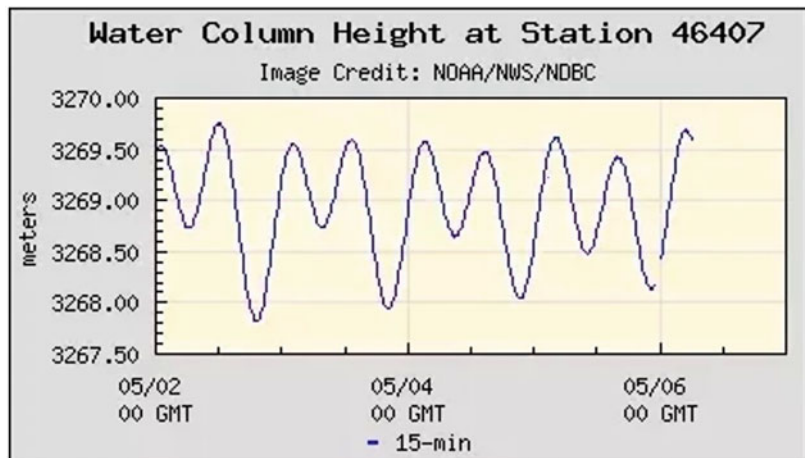
Tsunamis differ from wind-generated waves in that they are characterized as a shallow-water wave even in the deep ocean, and their pressure

wave extends all the way to the sea floor, whereas standard surface wave pressures can only be measured at about half their wavelength below the surface. The deep-ocean provides the perfect filter for detecting Tsunamis, eliminating all the noise from the surface waves. As a result, the BPR registers only the tidal fluctuations and tsunamis. The BPR is extremely sensitive and can detect water level changes of one millimeter or less, which is critical since tsunamis in deep water may only be a centimeter tall. Figure 2 shows the data acquired from NOAA buoy 46,407 off the coast of Oregon; as we can see, it follows the tidal cycle. Figure 3 shows a tsunami detected by NOAA buoy 32,412 on 27th February 2010 off the coast of Chile; at time “1,” the buoy went into event mode due to a magnitude 8.8 earthquake in Chile; 3 h later at time “2,” a tsunami was measured at the buoy site with a height of 24 cm. The pressure data is transmitted through the water column using an acoustic model and is then relayed off the buoy through the Iridium satellite network to the warning center. There that data is fed into models to improve the predictions and forecasting for the size and areas of impact of the tsunami.

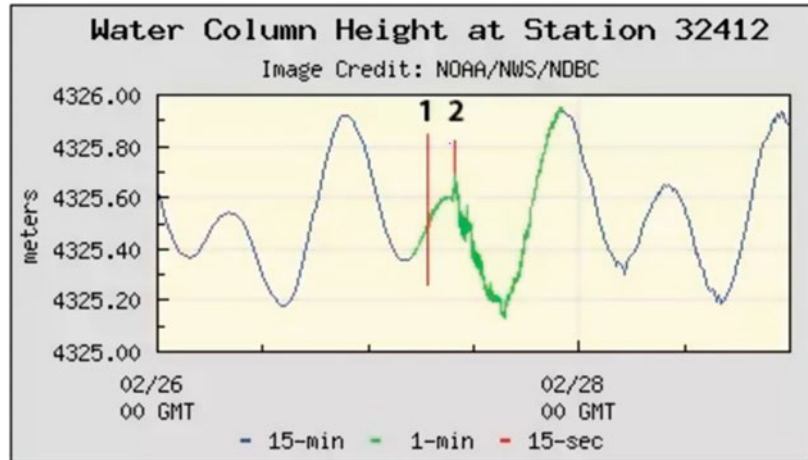
The tsunami detection algorithm must exclude the effects of white noise and tidal. As a result of this, the algorithm set a threshold value for the estimated amplitude of the pressure fluctuations within the tsunami frequency band, i.e., 30 mm in the North Pacific Ocean, which is based on the

Tsunami Warning Buoy,

Fig. 2 The pressure data of buoy 46,407 off the coast of Oregon. We can view all of NOAA’s oceanographic data buoys and their associated data on the National Data Buoy Center (NDBC) Website (National Data Buoy Center (NDBC) 2018)



Tsunami Warning Buoy,
Fig. 3 A tsunami detected
 by buoy 32,412 on 27th
 February 2010 off the
 coast of Chile



observation of the background oceanic noise in the past years. The predictions will update every 15 s (i.e., DART II). Once the estimated amplitudes exceed the threshold, the gauge goes into a rapid reporting mode to provide detailed information about the tsunami. It will remain in this mode for at least 4 h.

Figure 4 shows the application of the algorithm to a theoretical pressure series in DART (Deep-ocean Assessment and Reporting of Tsunamis (DART) Tsunami 2018). The series consists of an M2 tide with an amplitude of 1 meter and a short pulse that has an amplitude of 5 cm and a duration of 15 min. The pulse affects the difference both directly and through its indirect effect on the prediction. Following the first and largest differences, pulses continue to occur each hour with diminishing amplitude until the pulse no longer influences the predictions. As shown in Fig. 4, a software flag is set to -1 each time the difference exceeds the threshold. In turn, this exceedance flag controls a reporting flag that puts the DART gauge into its rapid reporting mode. The reporting flag is set to -1 as soon as the threshold is exceeded and remains equal to -1 until 4 h has passed since the last time the threshold was exceeded. The gauge then returns to its monitoring mode.

Figure 5 shows small-amplitude tsunamis observed by internally recording gauges in the North Pacific (Deep-ocean Assessment and Reporting of Tsunamis (DART) Tsunami 2018).

Relative to the high-passed data, the background noise in the difference series is either equal in amplitude or slightly enhanced. Neither tsunami is high enough in amplitude to set a DART gauge into rapid reporting mode.

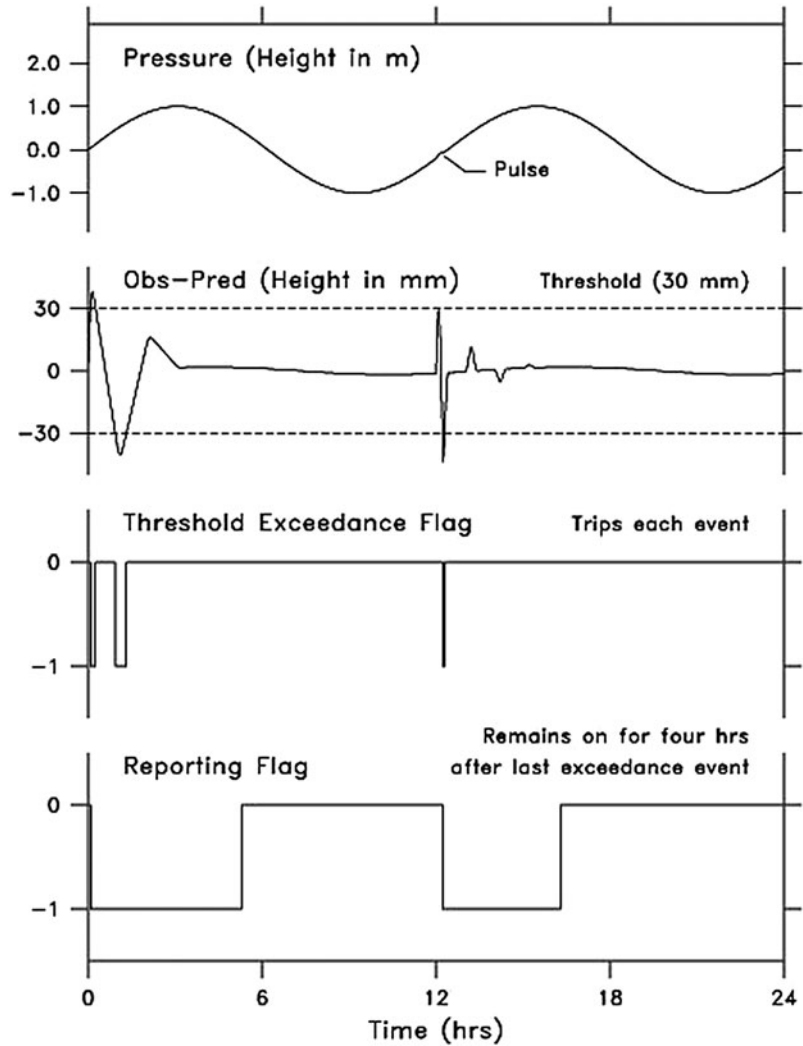
Applications

Dart II

The first-generation DART I systems had one-way communications from the BPR to the Tsunami Warning Centers (TWC) and National Data Buoy Center (NDBC) via the western Geostationary Operational Environmental Satellite (GOES West) (Milburn et al. 1996). DART I became operational in 2003. NDBC replaced all DART I systems with the second-generation DART II systems in early 2008. DART I transmits standard mode data once an hour (four estimated sea-level height observations at 15-min intervals). For information about the development history of this technology, see (NOAA Center for Tsunami Research 2018).

DART II became operational in 2005 (Green 2006). A significant capability of DART II is the two-way communications between the BPR and the TWCs/NDBC using the Iridium commercial satellite communications system (Meinig et al. 2005). The two-way communications allow the TWCs to set stations in event mode in anticipation

Tsunami Warning Buoy,
Fig. 4 The application of
the tsunami detection
algorithm to a theoretical
pressure series (Deep-ocean
Assessment and Reporting
of Tsunamis (DART)
Tsunami 2018)



Courtesy of PMEL

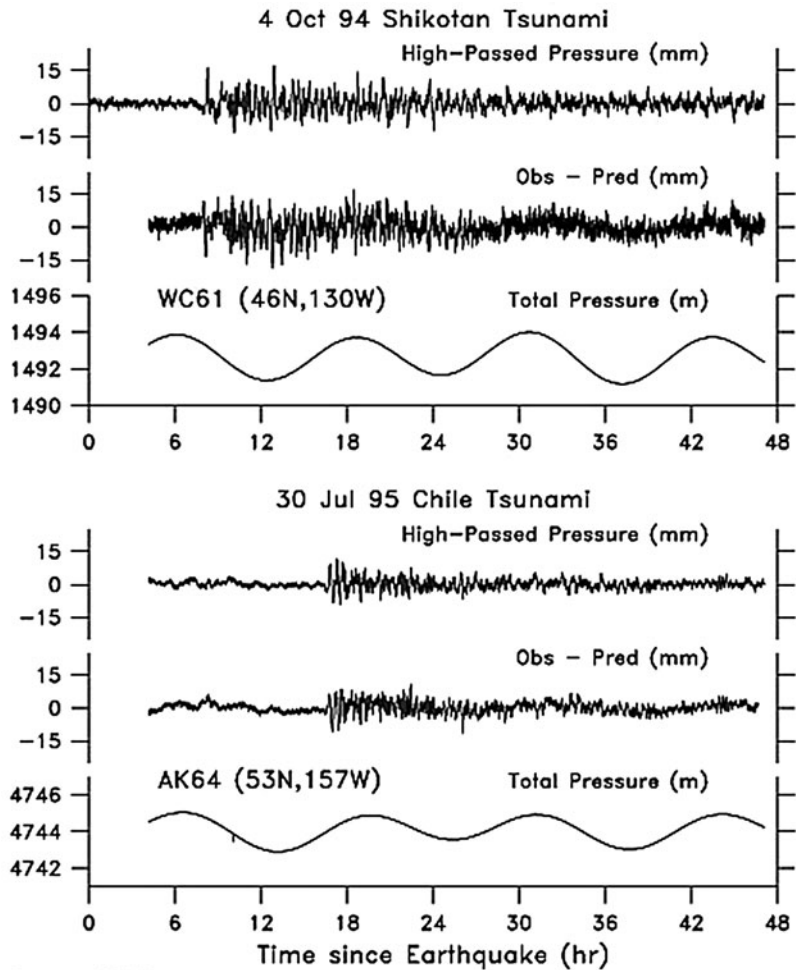
of possible tsunamis or retrieve the high-resolution (15-s intervals) data in 1-h blocks for detailed analysis. DART II systems transmit standard mode data, containing 24 estimated sea-level height observations at 15-min intervals, once every 6 h. The two-way communications allow for real-time troubleshooting and diagnostics of the systems. These buoys have two independent and redundant communications systems. NDBC distributes the data from both transmitters under separate transmitter identifiers. NDBC receives the data from the DART II systems, formats the data into bulletins grouped by ocean basin (see

(The NDBC 2018), for a listing of the bulletin headers used for each transmitted identifier), and then delivers them to the National Weather Service Telecommunications Gateway (NWSTG) that then distributes the data in real-time to the TWCs via NWS communications and nationally and internationally via the Global Telecommunications System.

Table 1 gives the performance characteristics and specifications of DART II in summary (Summary of DART II 2018). The DART II Tsunami warning buoys are placed all over the world to protect citizens of countries from dying or

Tsunami Warning Buoy,

Fig. 5 A small-amplitude tsunami observed by internally recording gauges in the North Pacific (Deep-ocean Assessment and Reporting of Tsunamis (DART) Tsunami 2018)



Courtesy of PMEL

getting injured; Fig. 6 shows the current deployed DART II location (The current deployed DART II 2018).

SAIC Tsunami Buoy

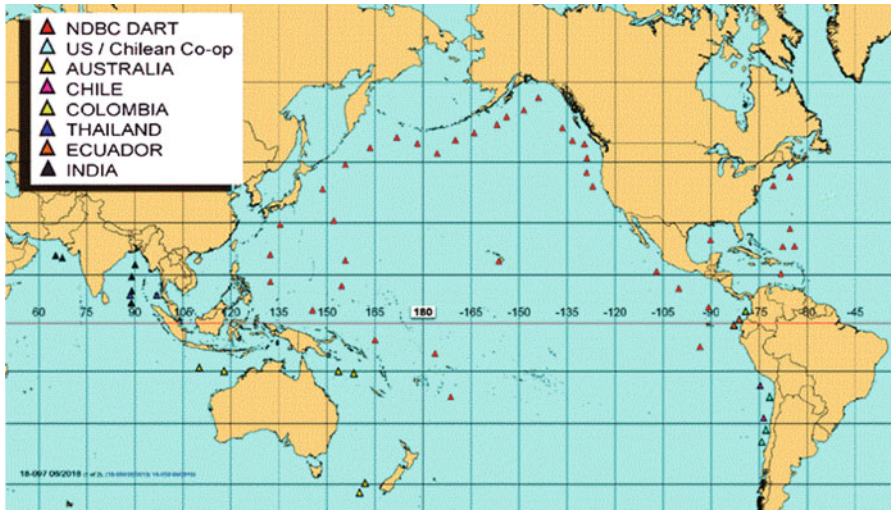
The SAIC Tsunami Buoy (STB) is an enhanced version of commercial tsunami buoy system based on the NOAA DART system in technology. SAIC (Science Applications International Corporation) has patent license agreement from NOAA and remains the only company currently licensed by NOAA to build commercial DART-based tsunami buoys (SAIC 2018). Under the license agreement, SAIC has produced more than 35 s-generation buoy systems based on the NOAA DART II system, and two types of third-

generation systems based on the Easy-to-Deploy (ETD) DART technology. As SAIC enters another decade of work with NOAA, the company is manufacturing more than 85% of the world's commercially deployed tsunami buoys, helping to make the concept of a globally interconnected tsunami buoy network a reality.

In 2006, the SAIC Tsunami Buoy (STB) was born, and the company began manufacturing its first commercial prototype tsunami assessment system. Like the DART II system, the STB has three subsystems: a bottom pressure recorder (BPR), a surface communications buoy, and a mooring subsystem. The third-generation (3G) commercial ETD DART systems are smaller in size and weight compared to the STB but still

Tsunami Warning Buoy, Table 1 Summary of DART II performance characteristics and specifications

Mandatory characteristic	Specification
Measurement sensitivity	Less than 1 millimeter in 6000 meters; 2×10^{-7}
Sampling interval, internal record	15 s
Sampling interval, event reports	15 and 60 s
Sampling interval, tidal reports	15 min
Two-way end-to-end communications	On demand, tsunami warning center (TWC) trigger
Tsunami data report trigger	Automatically by tsunami detection algorithm
Data flow, BPR to TWC	Less than 3 min after triggered event
Desired characteristic	Specification
Reliability and data return ratio	Greater than 80%
Maximum deployment depth	6000 meters
Minimum deployment duration	1 year
Operating conditions	Beaufort 9
Theoretical battery life, buoy	Greater than 2 years
Theoretical battery life, tsunameter	Greater than 4 years
Maximum status report interval	Less than 6 h



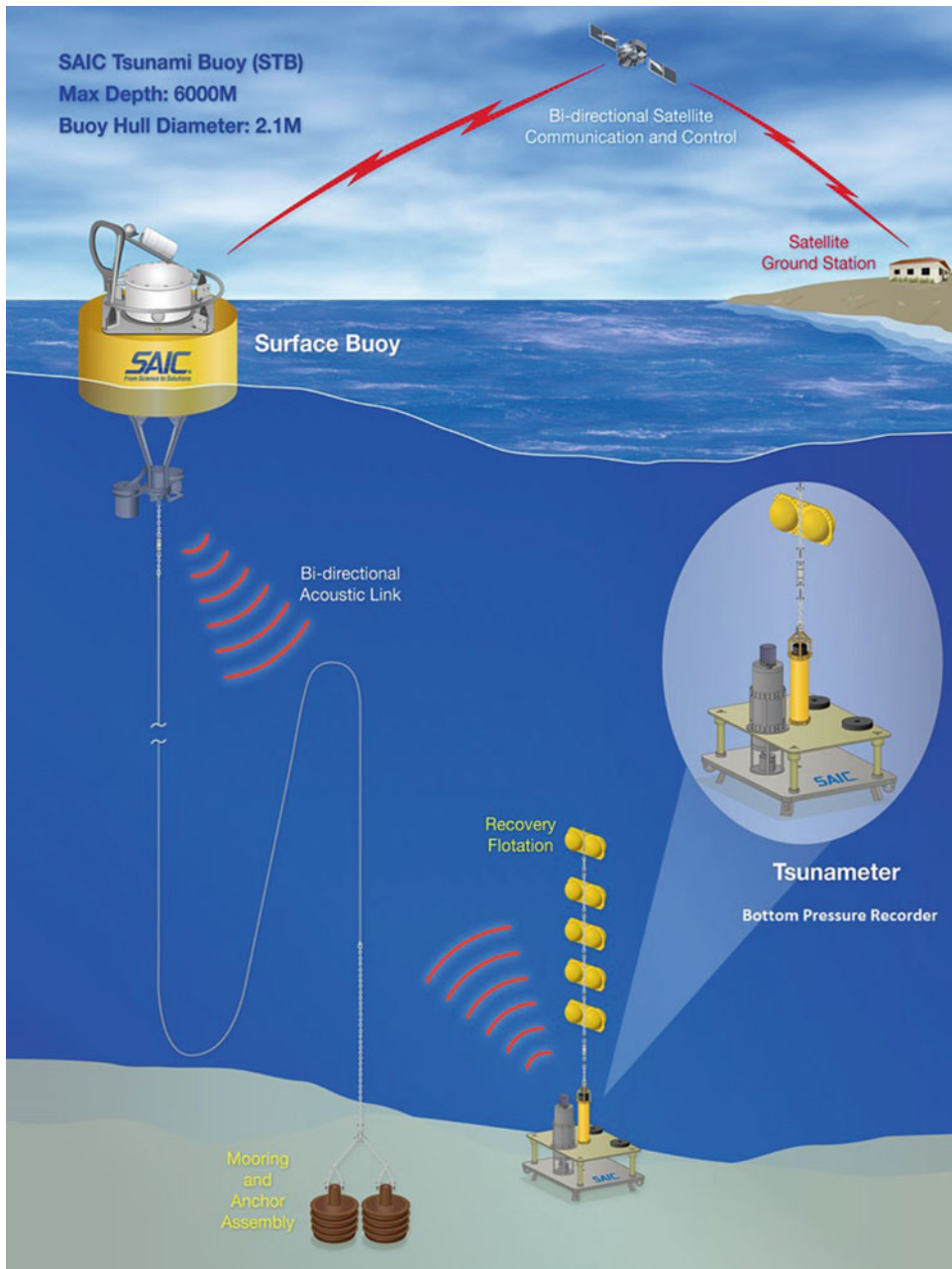
Tsunami Warning Buoy, Fig. 6 The current deployed DART II location shown on a map (The current deployed DART II 2018)

maintain DART performance standards. Despite its small size, the ETD DART's performance has been exceptional in challenging locations. SAIC's second- and third-generation buoy systems are currently operational in maritime countries worldwide, including Australia, Chile, China, India, Japan, Russia, and Thailand, and are gathering actionable data for their users and NOAA.

After producing the third generation ETD DART, SAIC developed the hybrid third-generation STB-MF tsunami buoy system

(as shown in Fig. 7). These upgrades included (SAIC 2018):

- Replacement of the standard low-frequency (LF) BPR with a single pressure vessel MF BPR.
- Increased reliability of the BPR by eliminating wet-mate cables and connectors, and reducing from five (5) pressure vessels to one (1).
- Utilization of the updated ETD DART core circuit board package.

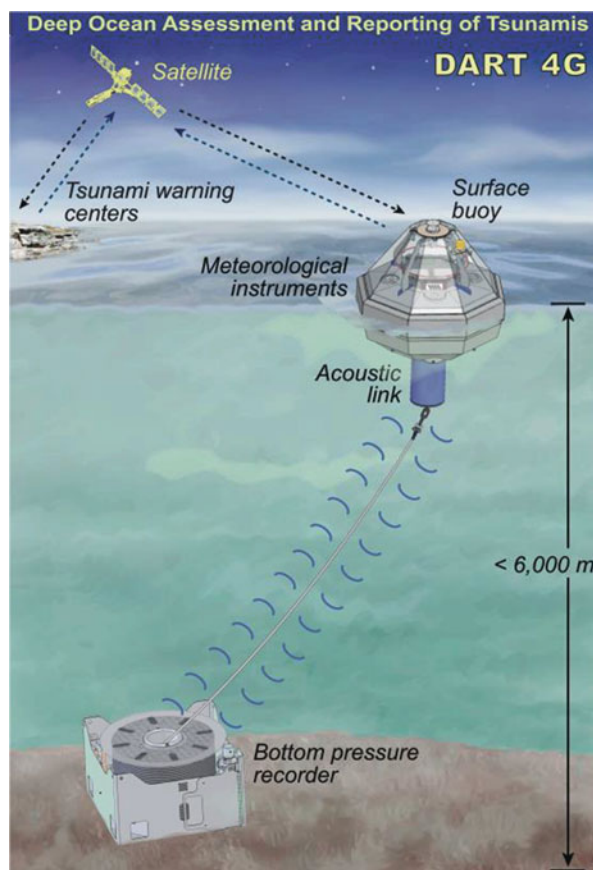


Tsunami Warning Buoy, Fig. 7 The hybrid third-generation STB-MF system with single pressure vessel bottom pressure recorder (BPR). (Figure from SAIC 2018)

- Replacement of the bulky, older technology LF transducers with more compact and updated MF transducers, to provide a 60–70° communication cone as opposed to the 40° cone in the LF system.
- More flexible buoy and BPR depth placement due to larger communication cone and watch circle.
- Increased use of inverse catenary moorings that are more robust in higher ocean currents.

Tsunami Warning Buoy,

Fig. 8 The DART 4G system consists of a surface buoy tethered to an anchored seafloor bottom pressure recorder (BPR). (Figure from DART 4G 2018)



To solve the ultimate challenge of near-field tsunamis warning, NOAA Pacific Marine Environmental Laboratory (NOAA PMEL) has developed a fourth-generation (4G) DART system (see Fig. 8), and the developmental 4G DART systems are currently being evaluated off the Chilean and Oregon coasts with good results. The 4G DART includes a new and improved pressure sensor, increased sampling rate, advanced internal processing software, and upgraded power management. A commercial 4G system is expected to be available within the next 12 months (reported in Sep.2016, see (SAIC 2018)).

A New Tsunami Warning Buoy in Japan

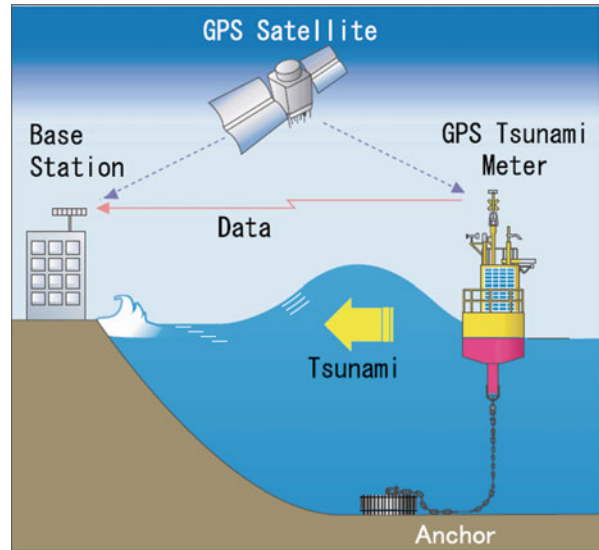
Kato et al. (2005) reported a new tsunami warning system, which uses real-time kinematic (RTK) GPS technology to position a GPS receiver mounted on top of a buoy floating at the sea surface, relative to a land-based GPS receiver.

The data are transmitted from buoy to shore using radio transmission. The GPS phase data are processed on a PC placed at the land-based station (see Figs. 9 and 10). The newly designed system is considered a more practical application to tsunami disaster mitigation than to survive for harsher oceanic environments far offshore.

A tsunami due to the 2004 M7.4 September fifth earthquake off the Kii peninsula, Japan, was recorded at the newly designed GPS buoy, about 13 km off the Muroto Promontory, southwestern Japan. The tsunami arrived at the buoy about 8 min before its arrival to the nearest tide gauge station at the tip of the Muroto Promontory. The predicted tsunami record based on preliminary source model shows excellent agreement (rms 2.7 cm) with the observed GPS record. This demonstrates that GPS buoy observations can be used not only for a tsunami warning system, but also for precise determination of the tsunami source.

Tsunami Warning Buoy,

Fig. 9 The monitoring system of the new GPS tsunami warning buoy. (Figure from Kato et al. 2005)



The German-Indonesian Tsunami Early Warning System

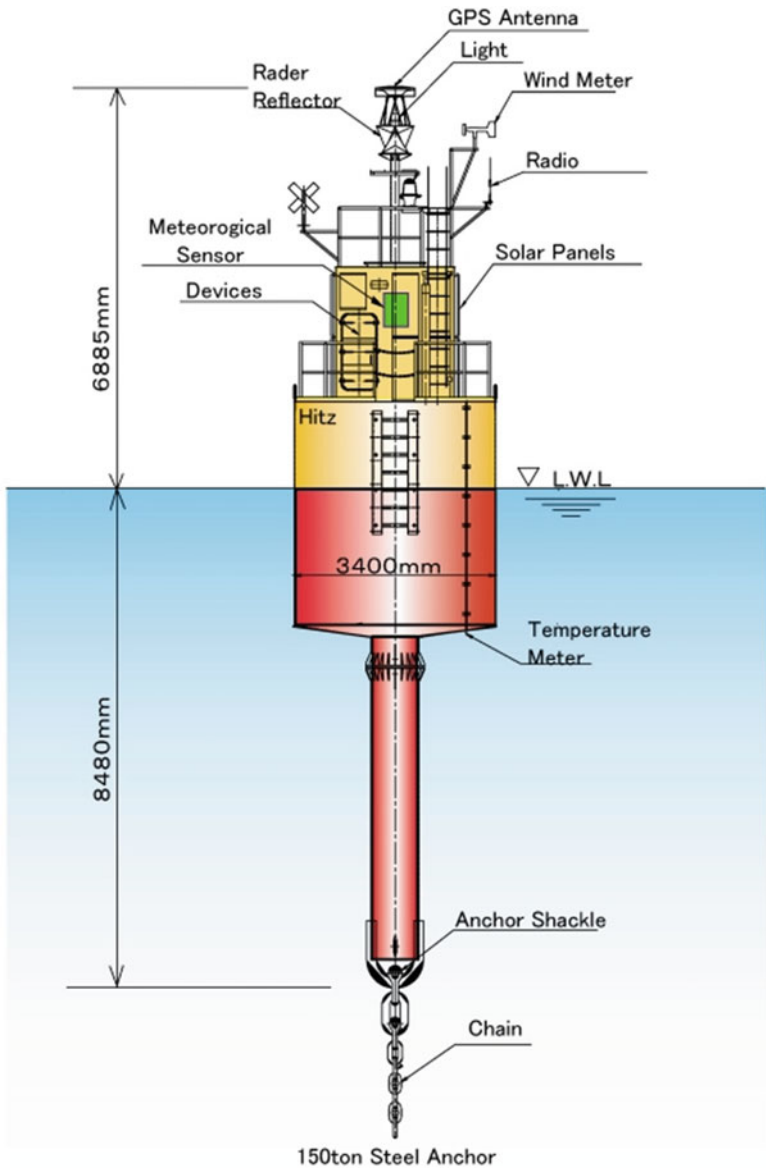
Due to the specific geological situation in Indonesia, the previously used, established tsunami warning systems are not optimal for Indonesia. The earthquakes in the Indian Ocean at Indonesia originate along the Sunda Trench, a subduction zone which extends in an arch from the northwest tip of Sumatra to Flores in eastern Indonesia. If a tsunami originates here, in an extreme case, the waves reach the coast within 20 min, so that only very little time remains for an early warning. Therefore, the concept of the German-Indonesian Tsunami Early Warning System (GITEWS) was based on this prevailing condition; the concept is shown in Fig. 11. The buoy that was once used and is now no longer in operation is shown in the Fig. 12.

The GITEWS is primarily based on model-based coupling of seismological data with GPS measurements and level measurements. There are more than 300 sensors distributed across all of Indonesia which supply their data to the warning center in real time. From these dates, a newly developed automatic decision support system (DSS) will compile a picture of the disaster situation, based on which the decision will be made whether to issue an alarm. The following steps will be taken in the process (GITEWS 2018):

- Quake location and strength: Ascertainment with seismological data; all earthquakes (worldwide) are recorded. All earthquakes $M \geq 2$ are evaluated in the national warning center. Earthquake information is basically provided. Tsunami warning alerts are only issued if a tsunami is expected (earthquake magnitude > 7).
- Fracture mechanism: Only strong suboceanic earthquakes with a distinct vertical component can cause tsunamis. An initial assessment as to whether the ocean floor has moved vertically can be determined using land fixed points.
- Ascertainment of a tsunami: On the coasts and on the offshore islands of Indonesia, tide gauges were installed with GPS components, which monitor the sea level. The data are integrated into the warning process.
- Decision-making: Within less than 5 min, the decision can be made as to whether a warning needs to be issued and – if so – to which coastal sections.

In the GITEWS, buoys are measurement instruments for the detection and verification of a tsunami in the early developing stage. The most important information, namely, the fast earthquake location and magnitude, without which either a simulation or a warning can be generated, cannot be supplied by buoy systems. Buoy

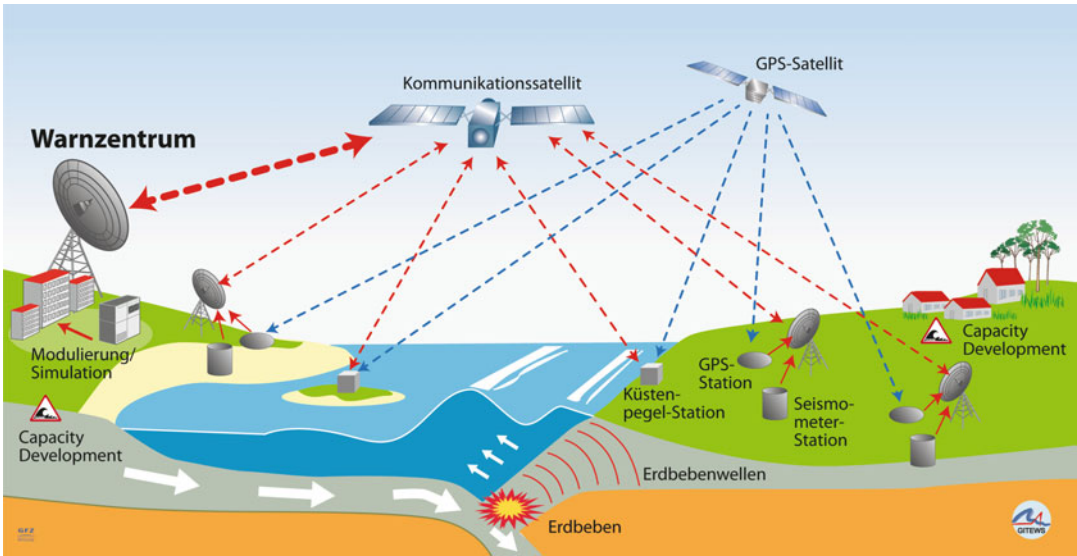
Tsunami Warning Buoy,
Fig. 10 A new GPS design
of tsunami warning buoy.
(Figure from Kato et al.
2005)



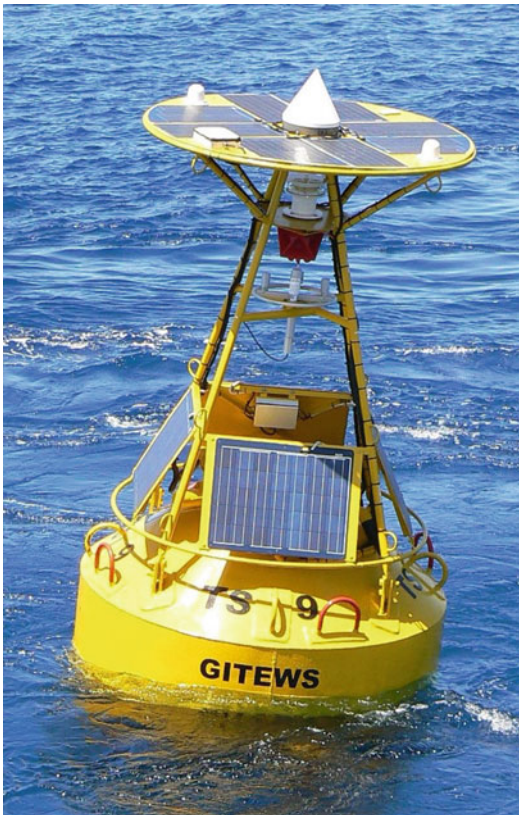
systems for the direct measurement of a tsunami were initially part of the GITEWS concept. The further development of the GPS Shield made it possible to discontinue pursuing the buoy concept. Therefore, buoys have no longer been part of the operational warning system since 2010, so that the high maintenance cost of buoy installations near coastlines can also be omitted (GITEWS 2018).

After that, GITEWS has developed model calculation methods to determine the possible extent

of a tsunami across the ocean. The measurement data, such as seismological parameters, GPS measurements of land displacements, and coastal tide gauge measurements of the sea level are being included in the model calculations. This results in a dynamic-updated model that constantly reflects the processes in the ocean and can forecast their future progress. The Tsunami Service Bus (TSB) – the integration platform in GITEWS – realizes the service-oriented architecture (SOA) approach by implementing the Sensor Web



Tsunami Warning Buoy, Fig. 11 Technical concept of GITEWS. (From website GITEWS 2018)



Tsunami Warning Buoy, Fig. 12 GITEWS tsunami buoy (The buoys are no longer in operation as part of the GITEWS since 2010)

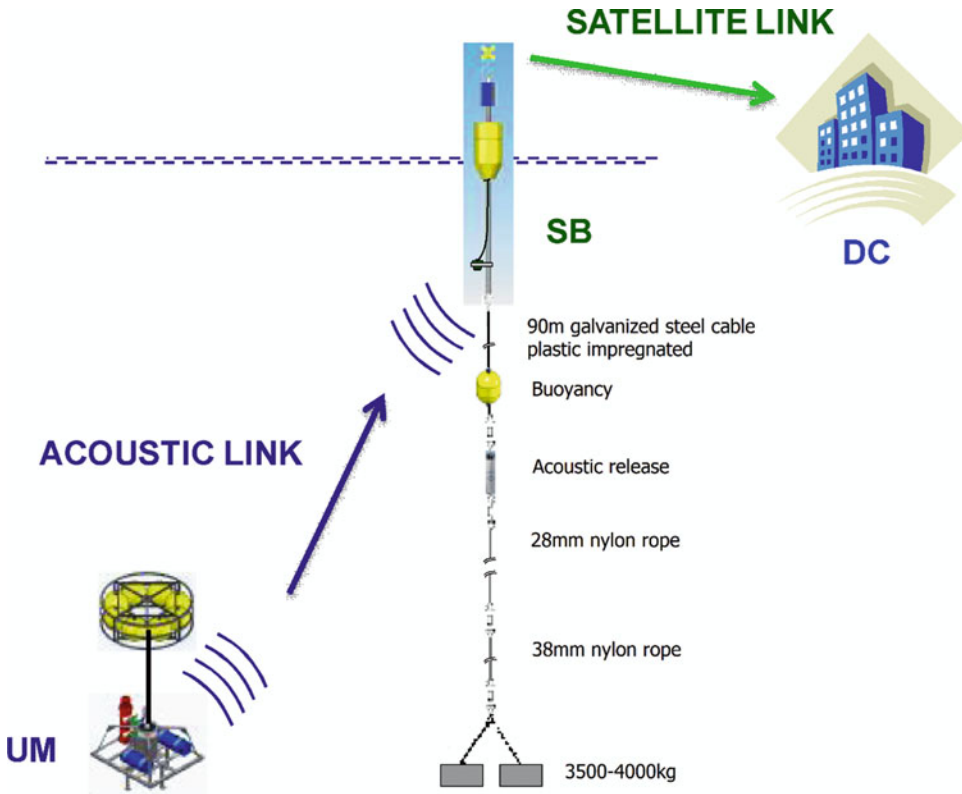
Enablement (SWE) standards and services (Fleischer et al. 2010).

Envirtech Tsunami Detection Buoys

The Tsunami Warning System (TWS) designed by Envirtech (website see (Envirtech Tsunami Buoys 2018)) is composed of the following main subsystems (as shown in Fig. 13):

- an underwater module (UM) to be installed at the sea bottom in open sea location for the accurate measurement of the tide and the identification of anomalous conditions,
- a surface buoy (SB) moored close to the UM to receive via acoustic link the measurement of UM to be transferred via satellite link to a data center (DC) for visualization and analysis,
- a data center (DC) on land composed of hardware and software infrastructures for the reception via satellite link of the data of SB and UM to be displayed, stored, and analyzed.

The tsunami buoy system comprises two components: the underwater module (UM), anchored to the sea floor, and the surface buoy (see Fig. 14). The surface buoy is composed of three main parts: a subsurface pole, a float (cone + cylinder) in the



Tsunami Warning Buoy, Fig. 13 Tsunami Warning System (TWS) designed by Envirtech. (Figure from Envirtech Tsunami Warning Buoy 2018)

middle, and a turret on the top. The sensors on the sea floor measure the change in height of the water column above by measuring associated changes in the water pressure. The water column height is communicated to the surface buoy by acoustic telemetry and then relayed via satellite to the tsunami warning center.

The underwater module detects by itself the presence of a “special” event and starts to operate in event mode. The sea level is acquired every 15 s, and collected data are transmitted to the surface each minute. In few seconds, data arrives to the control center and an alarm procedure is raised. Figure 15 shows the Sumatra event on October 2010, as collected by an Envirtech Tsunami buoy deployed in South China Sea, 2,600 km far the earthquake epicenter on a seabed 4,400 meters deep. The seismic waves reached the underwater module few minutes after the earthquake, setting it in alarm mode (Courtesy SOA –

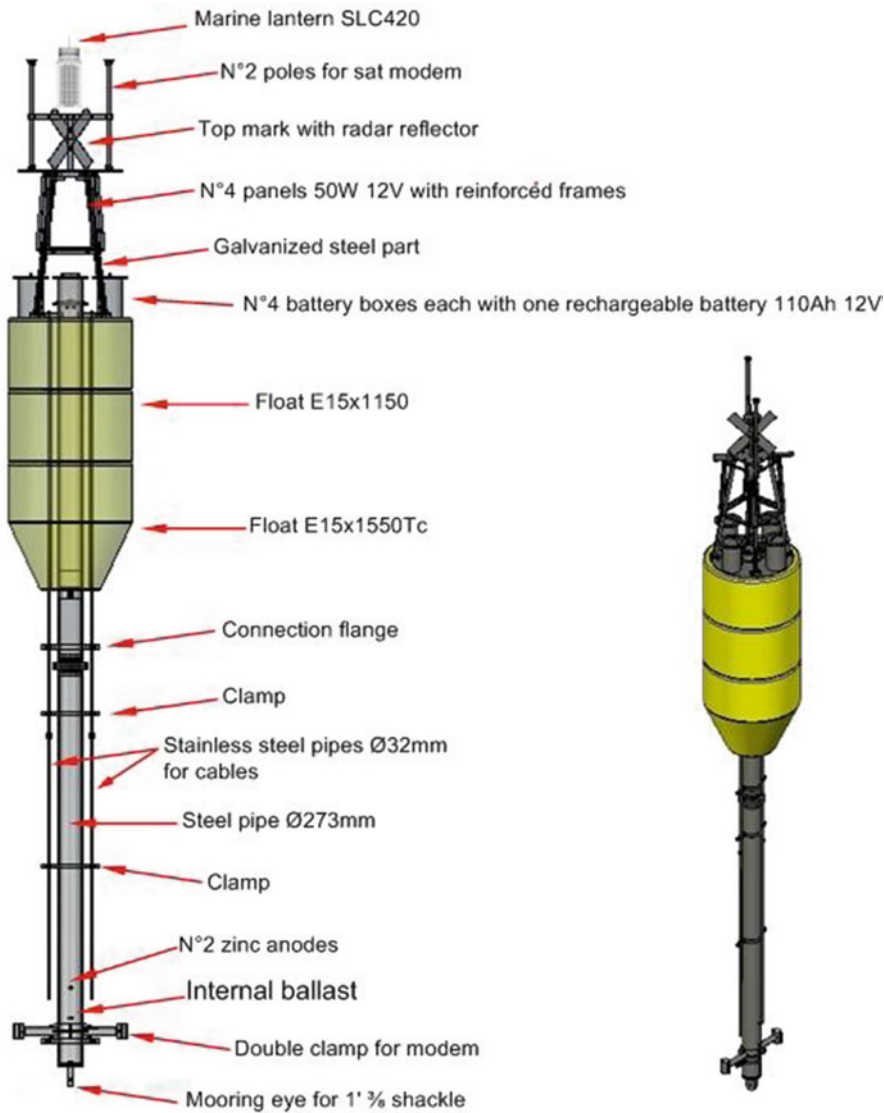
China). No tsunami followed the event in South China Sea.

The buoy has multiple composite functions. In addition to tide and tsunami detection, the buoy can be optionally supplied with further payloads such as a complete meteorological station (single or double), an acoustic doppler current profilers (ADCP) for multicell current data collection and water quality multiparameter probe.

The MKIII has been built to remain operational during super typhoons, with sea conditions up to Beaufort 14. In 2010, the buoy proved its great strength when the super typhoon Megi crossed its mooring position in the South China Sea.

Summary

When the tsunami devastated wide areas around the Indian Ocean in December 2004, the effects and the dangers of a tsunami came into global public consciousness with extensive media



Tsunami Warning Buoy, Fig. 14 The Envirtech MKIII-002 tsunami warning buoys. (Figure from Envirtech Tsunami Buoys 2018)

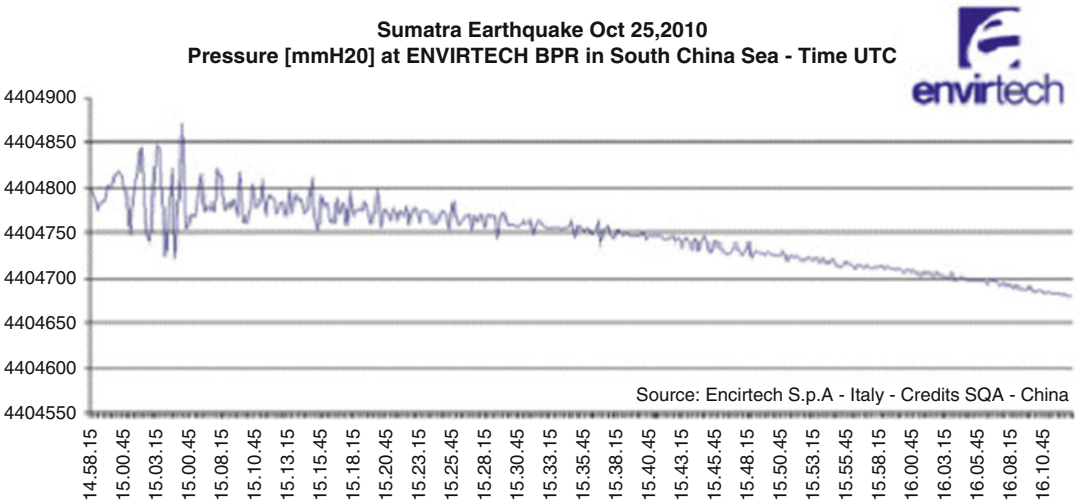
T

coverage. In the memory and perception of tourists and holiday makers, seashore sites may forever bear tsunami-related dangers, resulting in the desire for effective, reliable, and easy-to-use tsunami alarm systems.

In the existing modern tsunami warning systems, Global Navigation Satellite System (GNSS) plays a central role, whether in DART system with buoys or in GITEWS without buoys. The key role in these systems is played by GPS. Thanks to the

developments made in GNSS data processing, especially in the framework of the International GNSS Service, a relatively cheap infrastructure of high-tech high-sea buoys can provide a very accurate and efficient tsunami early warning system.

One of the challenges of tsunami warning is the application of rapid forecasting, for example, the application in Indonesia, where a tsunami caused by an earthquake in Sulawesi in 2018 killed hundreds of people. A long-term goal of tsunami



Tsunami Warning Buoy, Fig. 15 Sumatra event collected by an Envirtech tsunami buoy deployed in South China Sea on October 2010. (Figure from website Envirtech Tsunami Buoys [2018](https://www.envirtech.com/Envirtech_Tsunami_Buoys.html))

assessment is to develop sensors that can detect and measure near-field tsunamis as close to the power generation area as possible. If this is achieved, countries will be able to obtain data more quickly, which could reduce evacuation times.

Cross-References

- [Deep-Ocean Assessment and Reporting of Tsunamis \(DART\) BUOY](#)
- [Meteorological Monitoring and Measurement Buoy](#)
- [Wave Measurement Buoy](#)
- [WaveRider Buoy](#)
- [WaveScan Buoy](#)

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TTR (Top Tensioned Riser)

- SPAR Platform

Tungsten Inert Gas Welding (TIG)

- Welding Technology

Tungsten Insert Gas (TIG)

- Piping Technology

Turbine Installation Vessel (TIV)

- Jack-Up Platforms

Turret Mooring

- Single-Point Mooring

2D, Two Dimensional

- Numerical Simulation of Ice-Going Ships

TWS Tsunami Warning System

- Tsunami Warning Buoy

U

UFR, Umbilical, Flow Line, and Riser

- [Moored Ship in Ice](#)

ULS – Ultimate Limit States

- [Design of Renewable Energy Devices](#)

Ultimate Strength

- [Offshore Structure Design Under Ice Loads](#)

Ultrahigh Molecular Weight Polyethylene (UHMWPE) Fiber Net

- [Net Structures: Design](#)

Ultrashort Baseline (USBL)

- [Autonomous Underwater Vehicle \(AUV\)](#)
- [Doppler Velocity Log for Navigation System in Underwater Vehicle](#)

Ultra-short Baseline (USBL)

- [Integrated Navigation](#)
- [Long Baseline Underwater Acoustic Location Technology](#)
- [Ultra-short Baseline Underwater Acoustic Location Technology](#)

Ultra-short Baseline Underwater Acoustic Location Technology

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Synonyms

BATS (Broadband Acoustic Tracking System); Heading sensor (HS); Inertial navigation system (INS); Long baseline (LBL); Motion Reference Unit (MRU); Short baseline (SBL); Super short baseline (SSBL); Ultra-short baseline (USBL)

Definition

The underwater acoustic positioning system can be divided into long baseline (LBL), short baseline (SBL), and ultra-short baseline (USBL) or

super short baseline (SSBL) systems according to the spacing of the elements. **Ultra-short baseline** positioning system array length is generally on the order of several centimeters to tens of centimeters, usually less than or equal to half a wavelength, and the phase difference between the received signals of each array element is used to solve the spatial position of the target. All array elements form a sound array, which can be mounted at the bottom of the carrier or suspended from the side of the carrier or installed where fluid noise and structural noise are relatively weak. The ultra-short baseline system has high requirements for the installation of the acoustic array. The installation position of each array element before installation needs to be determined in advance, that is, the position deviation of the array element relative to the carrier coordinate system and the installation deviation angle of the acoustic array need to be known (Philips 2003; Vickery 1998).

The ultra-short baseline positioning system has the advantages of convenient installation, flexible use, and low system cost, but the positioning accuracy is weaker than the long baseline. Ultra-short baseline positioning systems usually choose the low-frequency or medium-frequency band. If the ultra-short baseline positioning system needs to cover the whole sea depth range, the low-frequency (8–16 kHz) or lower-frequency band is generally selected. The relationship between the operating frequency and the maximum working distance is shown in the following Table. 1.

The ultra-short baseline positioning system has two working modes, one is an acoustic response mode, and the detected target sends an inquiry

signal to each transponder, and the interrogated transponder sends back a response signal according to the received command and passes the inquiry signal, and the time difference between the response signals is used to calculate the distance. The other is the synchronous clock mode. The clock system at both ends of the inquiry and response needs to perform high-precision clock synchronization, and the distance is calculated by the time difference from the timing of the synchronization pulse to the receipt of the response signal.

Positioning Principle

Ultra-short baseline arrays have different positioning principles for narrowband and wideband received signals. When the signal transmitted by the target is a narrowband signal, the direction finding is performed by measuring the phase difference between the received signal and the different array elements; when the signal transmitted by the target is a wideband signal, the delay difference between the received signals and the different array elements is measured. Perform direction finding positioning (Han et al. 2015).

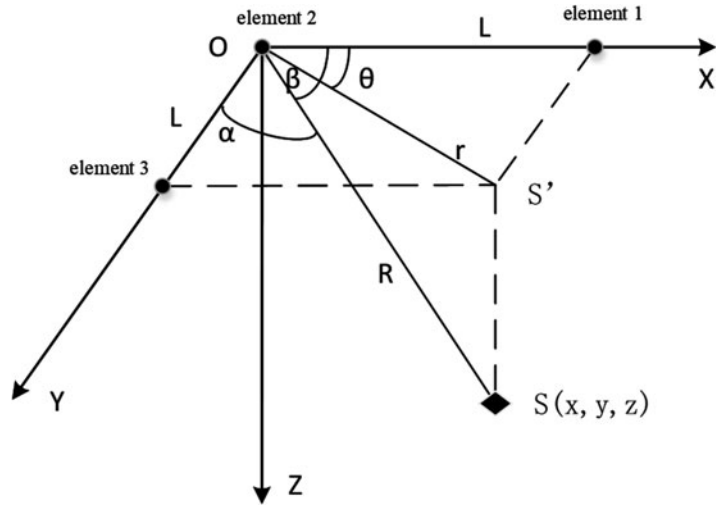
As shown in Fig. 1, the origin of the coordinate is O; the center of the array is the element 2 at the origin; the element 1 and the element 3 are located on the X and Y axes, respectively; and the distance from the center of the array is L, Z-axis. The bottom is perpendicular to the XOY plane. Let the coordinates of the target be S(x, y, z), then the following relationship:

Ultra-short Baseline Underwater Acoustic Location Technology, Table 1 Relationship between working frequency band and maximum working distance

Serial number	Band	Operating frequency (kHz)	Maximum working distance (m)
1	Low frequency (LF)	8~16	10,000
2	Medium frequency(MF)	18~36	3500
3	High frequency(HF)	30~60	1500
4	Extremely high frequency(EHF)	50~110	1000
5	Very high frequency(VHF)	200~300	100

Note: It is assumed that the noise level of the shipborne acoustic transducer array is less than 95 dB in the operating band and the underwater sound source level of the underwater transponder is greater than 195 dB re1μPa@1m.

Ultra-short Baseline Underwater Acoustic Location Technology, Fig. 1 Acoustic array positioning principle



$$\cos \alpha = \frac{x}{R}$$

$$\cos \beta = \frac{y}{R}$$

$$R = \sqrt{x^2 + y^2 + z^2}$$

where α is the angle between the vector and the X-axis, β is the angle between the vector and the Y-axis, and R is the target and the center of the array.

For the projection of S on the XOY plane, its angle θ with the X-axis is the target horizontal azimuth

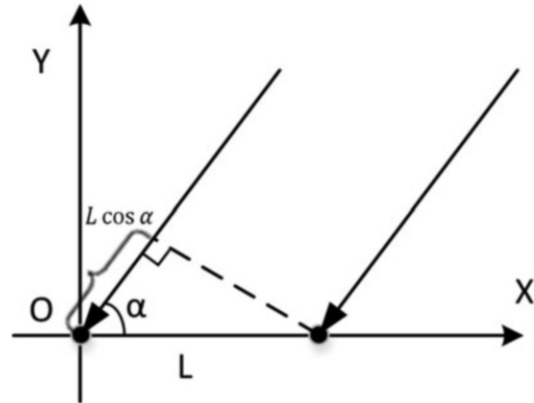
$$\theta = \tan^{-1} \frac{y}{x} = \tan^{-1} \frac{\cos \beta}{\cos \alpha}$$

$$R = \sqrt{x^2 + y^2}$$

$$z = \sqrt{R^2 - r^2}$$

where r is target horizontal slant distance and z is target depth.

It is assumed that the size of the array is much smaller than the distance between the target and the array, that is, $R \gg L$. At this time, it can be approximated that the target is incident from the far field, and the acoustic wave reaching each element is approximately plane-wave incidence, as shown in Fig. 2.



Ultra-short Baseline Underwater Acoustic Location Technology, Fig. 2 Coaxial two-element incident signal positioning principle

The delay difference between the array elements is satisfied.

$$\tau = \frac{L \cos \alpha}{c}$$

where c is the measured sound velocity in the sea area of the operation.

When the system uses the signal as a narrow-band signal and depends on the phase difference between the measured signal and the receiving array element, the target position is calculated (Sun et al. 2014).

$$x = \frac{\lambda \varphi_x ct}{2\pi L}$$

$$y = \frac{\lambda \varphi_y ct}{2\pi L}$$

where φ_x is the phase difference between two elements on the X-axis; φ_y is two-element phase difference on the Y-axis; and t is the one-way propagation delay of the acoustic signal from the target to the array.

When the system uses the signal as a wideband signal, the target position is calculated by using the delay difference. The actual measured value is the sum and R , which can be obtained by the response time measurement.

$$x = \frac{\lambda \tau_x R}{L}$$

$$y = \frac{\lambda \tau_y R}{L}$$

where τ_x is delay difference of signals received by two elements of X-axis $\tau_x = \frac{\cos \alpha}{L}$ and τ_y is delay difference of two signals received by Y-axis $\tau_y = \frac{\cos \beta}{L}$.

Application and Development Trends

The ultra-short baseline positioning system appeared relatively late, and reports on ultra-short baseline positioning systems were first seen in the early 1980s. After more than 40 years of continuous development, the products of a few influential companies occupy most of the world's markets. Among the more representative companies are iXblue in France, Kongsberg in Norway, Sonardyne and Nautronix in the United Kingdom, LinkQuest and EdgeTech in the United States.

France iXblue company's advantage field for marine science research, the new generation of ultra-deep long-range ultra-short baseline positioning system POSIDONIA2 maximum positioning depth of up to 7000 m, the maximum positioning slant distance of 8000~10,000 m, and positioning accuracy of 0.2% slant distance,

can be applied to deep-sea target tracking and positioning, such as ROV and AUV, and submarine tubing and cable-laying construction. The medium-range ultra-short baseline positioning system GAPS has a 3D geometric acoustic array that combines positioning with an optical fiber inertial navigation system (INS) to provide no less than 500 acoustic channels. Recently, the underwater acoustic communication function has been added to the system. In the depth roadmap mode, it can form an LBL/USBL/INS integrated navigation system together with a long baseline.

The Norwegian Kongsberg company has been engaged in the research and development of acoustic positioning systems. Its products have been applied in the field of dynamic positioning and submersible docking. The HiPAP series of ultra-short baseline positioning products have been developed. From 1996 to 2015, three generations of products were launched. HiPAP502/452/352/102 is a third-generation product that uses the same hardware and software platform and has a long baseline position function. The new communication and positioning acoustic protocol Cymbal is used in the product, and the positioning accuracy is further improved. The μ PAP series is a compact ultra-short baseline positioning system that can be operated on the surface carrier, with a built-in motion sensor and inertial navigation system. It is used for positioning requirements of ROV, tow body, diver, and other targets within 1000~4000 m underwater. The accuracy is 0.45% slant distance.

The UK's Sonardyne company's strengths are offshore oil and gas development. In 2005, it successfully launched WideBand's full range of ultra-short baseline products based on broadband digital technology. In 2010, it launched the second generation of ultra-short baseline products of 6G broadband digital technology. The main products are Mini-Ranger2, Ranger2, and Scout. The Mini-Ranger2 is a compact and portable product that can be combined with WSM6+, WRT6, and WM6 transponders. With a working range of up to 10,000 m and an accuracy of 0.1% slant distance, Ranger2 uses the unique multi-pulse stack technology to achieve data update rates of up to 1 s in any water depth

environment, supporting long baseline/ultra-short baseline combined positioning and inertial navigation. The Scout series includes Scout, ScoutPlus, and ScoutPro, which can be used for small target positioning such as ROV and tow body in the range of 500–1000 m. Nautronix products dominate the offshore drilling and mining industry. Its NASDrill-USBL is developed for use in near-shore drilling and other environments that require high noise and harsh environments. The system has a range of 4,500 m and an accuracy of 0.5% slant distance.

In 2002, LinkQuest of the United States launched the first diving ultra-short baseline positioning system TrackLink 1500, which quickly became a bestseller. The TrackLink 10,000 series developed in recent years belongs to the low-frequency remote ultra-short baseline positioning system. Based on its unique acoustic broadband spread spectrum technology, it can provide users with ultra-long-range deep-water communication and tracking and positioning solutions. EdgeTech, formerly known as ORE Marine, and ORE Marine products are TrackPoint II, TrackPoint 3, and TrackPoint 3P. EdgeTech has introduced the long-range ultra-short baseline positioning system BATS. The system is available in three configurations, portable BATS (Broadband Acoustic Tracking System), desktop BTAS, and installation. BATS can be combined with Motion Reference Unit (MRU) and heading sensor (HS) (Jin et al. 2018).

With the continuous improvement of marine survey and marine engineering requirements, ultra-short baseline positioning technology has gradually shifted from single positioning mode to integrated positioning mode, such as LBL/USBL, SBL/SUBL, and LBL/SBL/USBL systems; from acoustic positioning to acoustics positioning integrated inertial navigation positioning, Motion Reference Unit; signal system from narrowband to wideband; positioning for a small number of users to provide multi-target targets in the work area; and from single positioning function to multifunctional integration, such as integrated underwater acoustic communication; positioning accuracy is developed from the initial tens of meters to the meter.

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Ultra-violet (UV)

► Fiber Optic Hydrophone

Umbilical Cable

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Synonyms

Armor layer; Electric cable; Filler and wrap; Hydraulic pipe; Metallic tubes; Optical fiber; Sheath; Subsea production systems; Thermoplastic hoses

Definition

In general, umbilical cable is a kind of integrated functional structure, whose structure is similar to a baby's helically wound umbilical cord which is widely used in aerospace field (such as between the launch pad and the vehicle or between the space suit and the vehicle), diving (such as the umbilical cable that supplies oxygen for diving suits), and ocean engineering (such as applications to ROV underwater robot and subsea production systems). The purpose of this entry focuses on the umbilical cable which is used in the subsea production systems (Drumond et al. 2018).

Subsea production system is widely used in the development of subsea petroleum, petrochemical, and natural gas industries. In general, subsea wellhead, subsea production facilities, and subsea pipeline partly or fully transmit the extracted petroleum, petrochemical, natural gas, and water to the marine-dependent facilities or end-termination interfaces, which realized the development of offshore petroleum, petrochemical, and natural gas industries. The typical subsea production system can be divided as follows: subsea well system (including wellhead and Christmas tree), production system (including manifolds, template, and protective structure), and subsea pipe system (including chemical injection pipeline, subsea umbilical cable, and riser). Compared to other ways, the subsea production system is more suitable for developing deep water, marginal petroleum, and natural gas fields for the reason that it has a huge advantage. Since subsea umbilical cable is one of the main equipments of the subsea production systems, it plays a very important role to keep the operation safe. In another word, it is an indispensable component in subsea production systems, as a lifeline between surface facilities and subsea production systems (Yue et al. 2013).

The function of umbilical cables connecting upper floater and subsea production systems is to transmit power, hydraulic power, control and feedback signals, and chemical between them. Power, hydraulic power, and chemical are usually transmitted from top to bottom in one

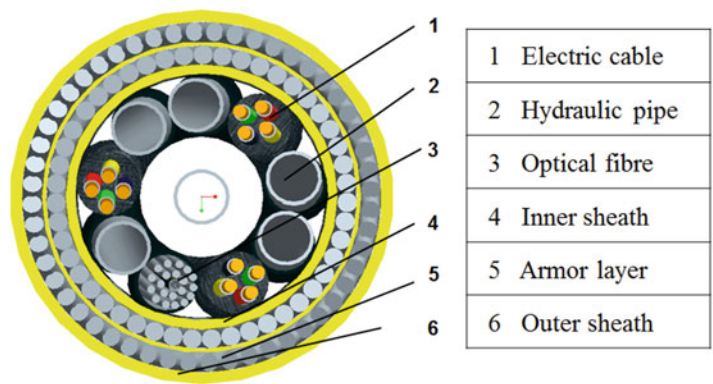
direction. The control and feedback signals are two-way transmissions (Kliewer 2013). The control signals are usually transmitted from top to bottom, while the feedback signals are usually transmitted from bottom to top, including all kinds of sensor signals and action status. In addition, storage and operation can be performed at specific temperatures within the service life range. Normally, the operation of the umbilical cable is resulted from the joint integration of the above functions and the operation of each component contributes to.

Scientific Fundamentals

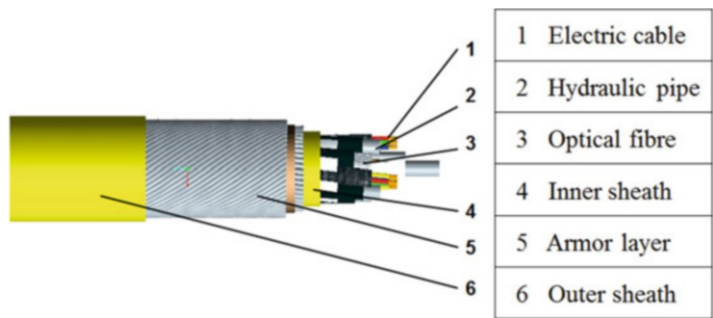
Structure Composition

The structures of subsea umbilical cables are various, which are determined by different environments and functions together. In most cases, there is no existence of unified standard or style. In general, the components of umbilical cables can be classified into two categories: One is the functional components, including the electrical cables, optical fibers, thermoplastic hoses, metallic tubes, and polymeric sheath (Hardee 2010). Another one is the strengthening components, including the carbon fiber rod, armor steel wires, and fillers. Thereinto, electric cable component is mainly used for the transmission of power and control. In addition to electrical communication, it can also be used for signal transmission. Optical fiber component is mainly used to transmit all kinds of signals, which is also used to monitor the status information of the umbilical cable under special circumstances. Pipe component contains different kinds of pipelines, such as steel pipes and hoses, which is used to transmit hydraulic fluid, chemical, petroleum, and natural gas. Filler is used to fill the void and fix the relative position of each component of the umbilical cable (Tang et al. 2014). Armor layer is usually placed outside the umbilical cable, whose function is to improve the axial stiffness and strength of the umbilical and enhance the stability of the subsea umbilical cable with more weight per length. Polymeric sheath is usually wrapped outside electric cables, optical fibers, and other types of pipelines.

Umbilical Cable,
Fig. 1 The structure of
umbilical cable



Umbilical Cable,
Fig. 2 The three-
dimensional structure of
umbilical cable



The function of sheath is to insulate and protect the umbilical cable (Lu et al. 2017) (Figs. 1 and 2).

Electric Cable

Electric cable is made up of conductor, insulation, and sheath, and sometimes there is also armor and shield included. Electric cable includes power cable and signal cable. Insulation layer provides dielectric strength and electrical insulation (Beaudonnet et al. 2013). Electric cable component is covered with sheath, which protects the cable component, makes the cable round, and adjusts the size and weight. Shield layer mainly has the function of shielding the magnetic field, while the armor mainly plays the role of enhancement (Yang et al. 2015).

Hydraulic Pipe

As a result of different functions of the umbilical cables, hydraulic pipe has different structures, sizes, numbers, and positions. The subsea umbilical cables with steel pipe components are usually used in deepwater ocean due to the much stronger resistance to internal and external pressure

compared with the thermoplastic hoses (Chitwood 2006). Most steel pipes are covered with plastic layers, which can prevent cushion and wear between steel pipes and other components or steel pipes.

Optical Fiber

Optical fiber is the most important carrier of communication transmission. There are one or more optical fiber components in the umbilical cable. In general, optical fiber cores are no more than 12 cores. The fiber is suspended in a thixotropic ointment, which can prevent water, keep the fiber position stable, and protect the fiber to the maximum extent at the same time. The stability of light attenuation is maintained in various mechanical and environmental conditions (Williams et al. 2002). The fiber protection tubes are usually high-strength stainless steel tubes with high resistance to lateral pressure and corrosion, while the steel pipe is surrounded by one to two layers of steel wire which is used to enhance the tensile strength of optical fiber cable. The optical fiber cable is covered with sheath, which protects the

fiber cable, making the cable round, adjusting the size and weight, and preventing corrosion and cushion.

Sheath

Sheath can be classified into the inner sheath and the outer sheath. The function of the inner sheath is to make the cables round, provide mechanical buffering, reduce friction and anticorrosion, and keep the structure of the umbilical stable. The outer sheath is the last line of defense of the umbilical. In addition to the likeness of the influences of the inner sheath, the outer sheath also has high anticorrosion performance, UV, and seawater resistance. In some umbilical cables, the outer sheath also has the resistance to mold and marine life (Dupont et al. 2002).

Armor Layer

Different armor types can be chosen according to different uses, environments, and working conditions of the umbilical cables, such as single round steel wire, double (quadruple) round steel wire, and double (quadruple) flat steel wire. Steel wire mainly plays the role of enhancement, providing the ability of tensile, anti-torsion, and anti-pressure for the umbilical cables. At the same time, steel wire also plays a shielding role, which is mainly to shield the magnetic field. Single round steel wire is seldom used, which is mainly used in shallow sea or river where the tension is not high. The advantage of this armored method is the lightness and softness, while the disadvantage is that the tensile strength and resistance are poor. When pulled, the torsion will appear. Different from armored method, the winding direction of the double wire layer is opposite in double (quadruple) round steel wire, so it can keep the torsion balance. Compared with double (quadruple) round steel wire, double (quadruple) flat steel wire has its advantage of reducing the outer diameter of cable and increasing the tensile resistance of umbilical cables. The disadvantage of it is that the cable is so hard that the bending performance is poor. So, we can further combine flat steel wire and round steel wire, which has the advantage of high tensile strength and resistance. Even though the weight is heavier,

it is suitable for high mechanical external force (Lu et al. 2017).

Filler and Wrap

The roles of filler are mainly reflected in the following aspects. Firstly, the filler makes the umbilical cable round, which is convenient for the latter process and which is helpful for external strengthening of components. Secondly, the filler improves the tensile strength of the umbilical cable and reduces the friction between the functional components and the layers. Thirdly, the filler can improve the stability of the structure of the umbilical cable and can be used for water resistance material and can be wrapped around the block.

Classification

In general, umbilical cables contain different component types and specific numbers, according to the actual demand of petroleum and natural gas field engineering, the existing facilities and reserved equipment, and users' requirements. Therefore, it can be concluded that umbilical cable is a typical custom product (SoDahl and Ottesen 2011).

According to the difference between the umbilical cables with pipe component, it can be classified into the umbilical cable with steel tubes and with hoses. According to whether the armor layer is included, it can be divided into the umbilical cable with armor layer and without armor layer. According to the different armor materials, it can be divided into metal armor and nonmetal armor umbilical. According to the application condition of umbilical cables, it can be divided into static and dynamic umbilical cables. There are four common types of umbilical cables as the following:

Single Armor Layer Umbilical Cable with Electric Cable, Optical Fiber, Hydraulic Fluid, and Steel Tube

This type of umbilical cable is capable of transmitting power, control, communication, optical communication, hydraulic power, and chemical simultaneously. And the armor layer is made up of round steel wire (Pesce et al. 2010). There is an

inner sheath and outer sheath inside and outside the armor layer, which is suitable for the subsea production systems with electrohydraulic compound control method.

Double Armor Layer Umbilical Cable with Electric Cable, Optical Fiber, Hydraulic Fluid, and Steel Tube

The function of this type of umbilical cable is the same as the single armor layer one. What makes a difference is that there are two layers of flat steel wire in this type of cable with only one outer sheath and lining inside. It is also suitable for the subsea production systems with electrohydraulic compound control method.

Electrohydraulic Compound Umbilical Cable with Hose

This type of umbilical cable is armored by five layers of round steel wire. The armor can withstand tensile load, increase subsea stability, and protect the internal structure of the umbilical. It is suitable for the subsea production systems with electrohydraulic compound control method. Due to the permeability and self-expansion of the hose, it is suitable for short control distance and low hydraulic requirement particularly.

Electrohydraulic Compound Umbilical Cable with Steel Tube

This type of umbilical cable is suitable for the subsea production systems with electrohydraulic compound control method, which is similar to the one with hose. What is different is that this umbilical cable is composed of hose or tube (Zhu et al. 2015).

Property of Umbilical Cable

Functional Performance

As is mentioned above, the umbilical cables are for the operation demand to meet the corresponding function. Therefore, there is a demand of design indicators in the process according to each function in the production systems. Its design indicators correspond to the functional components (electric cable, optical cable, and hydraulic pipelines). The performances of electric cable and optical cable can be explained

in detail by referring to corresponding entries (Witz and Tan 1995). And the performances of hydraulic pipelines are permeability, corrosion resistance, volume expansion, compatibility, and clean ability.

Mechanical Property of Umbilical Cable

In addition to meeting the above executive functional indicators, the strengthening design of umbilical cable needs to consider the corresponding mechanical safety guarantee to ensure the normal operation of other functional components. The mechanical properties of umbilical cables are as follows:

1. Stiffness

Structural stiffness is called stiffness for short, which has the ability of structure or component to resist deformation. As an important part of the submarine production system, umbilical cable has undertaken various external static and dynamic loads in the marine environment, such as gravity load, wind load, current load, and wave load. These loads on the umbilical cable are as forces or moments (Custódio and Vaz 2002). The transportation and operation of umbilical cable depend on the stiffness of the umbilical itself. Therefore, the stiffness of umbilical cable is a quite important mechanical property which is the key to determine whether the cable can work safely in subsea production system. The main stiffness indexes of umbilical cable are tensile stiffness, torsional stiffness, and bending stiffness. The axial tension and torque are coupled, due to the helical structure in umbilical cable. When there is axial displacement, axial elongation and torsion can occur simultaneously. The relationship between the force and displacement is shown as follows:

$$\begin{bmatrix} F \\ M_t \end{bmatrix} = \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{bmatrix} \frac{\Delta L}{L} \\ \frac{\Delta \theta}{L} \end{bmatrix} \quad (1)$$

In the formula, F represents the axial tension, M_t represents the torque, ΔL represents the

axial elongation, $\Delta\theta$ represents the torsion angle, and L represents the length of umbilical. k_{11} , k_{22} are tensile and torsional stiffness. k_{12} , k_{21} are interaction stiffness. In particular, balance of torsion should be considered during installation to avoid damage because of torsion knotting or torsion accumulation (Skeie et al. 2012).

Umbilical cables have complex structures which are made up of multiple layers of materials and formed by cross-wound laying of different angle components. Due to the unbonded mode of each structural layer of umbilical cable, there will be relative sliding between each structural layer and a lot of contact interface and interface friction with the increase of the curvature while it bends. And the bending deformations are also inconsistent for different types of umbilical cables. Umbilical cable is usually in the full sliding state in practical engineering application. Therefore, in general, bending stiffness is estimated with complete sliding models. The greater the angle of armor winding is, the smaller the bending stiffness of umbilical cables is, the better the performance of flexible bending is.

2. Strength

Strength is the ability of a structure or component to resist damage. Being similar to the stiffness of umbilical cables, strength is another important mechanical performance. In general, hoop stress, radial stress, and axial stress are included in the stress of typical umbilical cables. The total load capacity of umbilical cable is determined by the ultimate stress state of most vulnerable components. Hoop stress, radial stress, and axial stress are usually combined by the strength yield criterion so as to check whether the stress satisfies the yield strength of the materials.

The main strength properties of umbilical cable include maximum tensile capacity and minimum bend radius. Maximum tensile capacity is the failure criterion of umbilical structure with the strain failure of different

components (such as steel tube, armor, polymer, and so on). When the maximum tensile capacity of each component is given, the minimum value of these components is taken as the maximum tensile capacity of the umbilical, which should be greater than the extreme tensile load in the application environment. It is important to ensure adequate safety margin at all operation conditions so as to ensure the normal and safe operation of the umbilical cable. In addition, buckling failure, plastic failure, and functional failure of umbilical cable will be caused by excessive bending during the storage and working operation. Therefore, minimum bend radius is also a key property for the umbilical cable design.

3. Fatigue life

During the working operation of umbilical cables, alternating tension and bending load can be generated as a result of the wave and current. As a result, the armor and steel tube are damaged by the fatigue failure. Fatigue is one of the most common failure modes for dynamic umbilical cables, which directly determines the service life of umbilical cable. Therefore, the estimation of fatigue life of umbilical cable is the most important property in the whole design process.

Ancillary Equipment

In addition to the main equipments, some ancillary equipments of umbilical are needed to ensure that the umbilical cable works normally in subsea production systems. It can be classified as ancillary equipments for connecting and for strengthening. The ancillary equipments for connecting are used to connect the cable and the upper floating body or the underwater development device and to provide functional protection, which include surface umbilical termination, pull-in head, suspension device, J-tube or I-tube seals, subsea umbilical termination, connection box, and weak link. The ancillary equipments for strengthening are used to maintain the overall layout of the umbilical cable and to provide mechanical protection for the joint, including bend stiffeners, bend

restrictors, and buoyancy attachments (Tang et al. 2014).

Key Applications

Umbilical cable is one of the main equipments of the floating production systems. As a lifeline between surface facilities and subsea production systems, it is a key component of the subsea control systems. Generally, umbilical cable is composed of electrical cable, optical fiber, thermoplastic hose and metallic tube, transmitting hydraulic, power, control signal, chemical, and so on. So far, it is widely used for the development of subsea oil and natural gas during the whole production process in shallow water, deep water, and ultra-deep water (Witz and Tan 1992).

The application of subsea umbilical cable has a history of nearly 60 years in ocean engineering. It was in the 1960s that the earliest umbilical cable appeared in a subsea wellhead in Mexico by Shell. In the 1970s, there is a system which is called electrohydraulic compound control system appeared, making it possible to reduce the cross-sectional area of subsea umbilical cable greatly. In addition to subsea hoses, it also includes power cables and data transmission lines. The early subsea umbilical cable was made of thermoplastic hoses until Sandvik used metallic tubes instead of thermoplastic hoses firstly in the 1980s, which was used in Frigg petroleum and natural gas field in northeast Norway successfully. In the twenty-first century, the demand for power increased in the subsea production systems, as the depth increased. Nexans began to develop and produce the world's first umbilical cable with a high voltage cable (24 V) for BP in 2005. Since then, in subsea deepwater petroleum as well as natural gas fields, the subsea umbilical cable which is characterized with high voltage has been widely used (Fleming et al. 2005).

Presently, about 60% of subsea umbilical cables are used in the Atlantic region, mainly in Mexico, Brazil, Norway, and Angola. And about 30% of subsea umbilical cables are used in

Southeast Asia and Oceania, mainly in Indonesia and Australia.

Cross-References

- ▶ [Armor Layer](#)
- ▶ [Electric Cable](#)
- ▶ [Filler and Wrap](#)
- ▶ [Hydraulic Pipe](#)
- ▶ [Metallic Tubes](#)
- ▶ [Sheath](#)
- ▶ [Subsea Production Systems](#)
- ▶ [Thermoplastic Hoses](#)

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Umbilical Termination Assembly (UTA)

- [Decommissioning of Offshore Oil and Gas Installations](#)

UN Convention on the Law of the Sea (UNCLOS)

- [Decommissioning of Offshore Oil and Gas Installations](#)

Uncertainty Theory

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Underwater Acoustic

- [Underwater Acoustic Sensor Network](#)

Underwater Acoustic (UWA)

- [Underwater Acoustic Communication](#)

Underwater Acoustic Channel

- [Underwater Acoustic Sensor Network](#)

Underwater Acoustic Communication

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Synonyms

Analog-to-digital (A/D); Digital-to-analog (D/A); Frequency shift keying (FSK); Intersymbol interference (ISI); Layered space-time codes (LSTC); Low earth orbiting (LEO); Low-density parity check (LDPC); Maximum likelihood sequence detection (MLSD); Maximum likelihood sequence estimation (MLSE); Multiple input multiple output (MIMO); Orthogonal frequency division multiplexing (OFDM); Phase shift keying (PSK); Quadrature amplitude modulation (QAM); Signal-to-noise ratio (SNR); Single-input single-output (SISO); Space-time trellis codes (STTC); Spatial modulation (SM); Trellis-coded modulation (TCM); Underwater acoustic (UWA); Underwater wireless acoustic communication

Definition

Underwater acoustic (UWA) communication is a technique for sending and receiving messages by sound in water. As electromagnetic waves propagate poorly in sea water, acoustics provide the most obvious medium to enable underwater communications. UWA communication is difficult due to limited bandwidth, extended multi-path, refractive properties of the medium, severe fading, rapid time variation, and large Doppler shifts. Compared to terrestrial communication, UWA communication has much lower data rates and shorter communication range.

Scientific Fundamentals

Basic System Model

A typical UWA communication system for the transmitter and receiver in the presence of UWA channels is shown in Fig. 1. Since acoustic signals have low frequency, the passband samples are often directly generated by the modulation module (Zhou and Wang 2014). After digital-to-analog (D/A) conversion, the passband signal is amplified, passed to matching circuits, and matched to the transducer. At the receiver side, the weak signal is increased in level by a pre-amplifier, filtered by a simple bandpass filter, and sampled at the passband. Finally, the bit information can be achieved by the demodulation module. From the signal processing point of view, the channel includes the imperfections of the transmitter and receiving circuits. All the modules that are lumped together are called channel

between the D/A module and analog-to-digital (A/D) module.

UWA Communication Channels

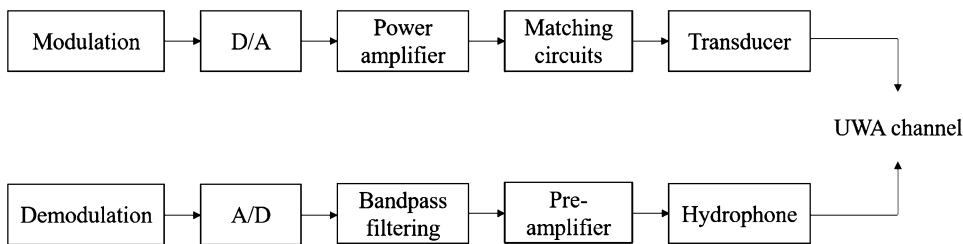
UWA communication channels are generally recognized as one of the most difficult communication media in use today (Stojanovic and Preisig 2009). Acoustic propagation is best supported at low frequencies, and the bandwidth available for communication is extremely limited. Although the total communication bandwidth is very low, the system is, in fact, wideband, in the sense that bandwidth is not negligible with respect to the center frequency. Sound propagates in the water at a very low speed of approximately 1500 m/s, and propagation occurs over multiple paths. Delay spreading over tens or even hundreds of milliseconds results in frequency-selective signal distortion, while motion creates an extreme Doppler effect. All of these factors are the key problems to be solved in UWA communication. Next, we will introduce UWA communication channels from four aspects.

Attenuation and Noise

A distinguishing property of acoustic channels is the fact that path loss depends on the signal frequency. This dependence is a consequence of absorption (i.e., transfer of acoustic energy into heat). In addition to the absorption loss, the signal experiences a spreading loss, which increases with distance. The overall path loss is given by

$$A(l, f) = (1/l_r)^k a(f)^{l-l_r},$$

where f is the signal frequency and l is the transmission distance, taken in reference to some l_r .



Underwater Acoustic Communication, Fig. 1 UWA communication system

The path loss exponent k models the spreading loss, and its usual values are between 1 and 2 (for cylindrical and spherical spreading, respectively). The absorption coefficient is $a(f)$, which can be obtained using empirical formula (Berkhovskikh and Lysanov 1982).

Noise in an acoustic channel consists of ambient noise and site-specific noise. Ambient noise is always present in the background of the quiet deep sea. Site-specific noise, on the contrary, exists only in certain places. For example, ice cracking in polar regions creates acoustic noise as do snapping shrimp in warmer waters. The ambient noise comes from sources such as turbulence, breaking waves, rain, and distant shipping. While this noise is often approximated as Gaussian, it is not white. Unlike ambient noise, site-specific noise often contains significant non-Gaussian components.

The attenuation, which grows with frequency, and the noise, whose spectrum decays with frequency, result in a signal-to-noise ratio (SNR) that varies over the signal bandwidth. If one defines a narrow band of frequencies of width Δf around some frequency f , the SNR in this band can be expressed as

$$\text{SNR}(l, f) = S_l(f) / A(l, f) N(f),$$

where $S_l(f)$ is the power spectral density of the transmitted signal. For any given distance, the narrowband SNR is thus a function of frequency.

Another important observation to be made is that the acoustic bandwidth is often on the order of the center frequency f_c . This fact bears significant implications for the design of signal processing methods, as it prevents one from making the narrowband assumption ($B \ll f_c$) on which many radio communication principles are based. Respecting the wideband nature of the system is particularly important in multichannel (array) processing and synchronization for mobile acoustic systems.

All in all, the fact that the available bandwidth depending on the distance has important implications for the design of UWA networks. Specifically, it makes a strong case for multihopping, since dividing the total distance between a source

and destination into multiple hops enables transmission at a higher bit rate over each (shorter) hop. The same fact helps to offset the delay penalty involved in relaying. Since multihopping also ensures lower total power consumption, its benefits are doubled from the viewpoint of energy-per-bit consumption on an acoustic channel.

Multipath

Multipath formation in the ocean is governed by two effects: sound reflection at the surface, bottom, and any objects and sound refraction in the water. The latter is a consequence of the spatial variability of sound speed. Sound speed depends on the temperature, salinity, and pressure, which vary with depth and location; and a ray of sound always bends toward the region of lower propagation speed, obeying Snell's law. Near the surface, both the temperature and pressure are usually constant, as is the sound speed. In temperate climates, the temperature decreases as depth begins to increase, while the pressure increase is not enough to offset the effect on the sound speed. The sound speed thus decreases in the region called the main thermocline. After some depth, the temperature reaches a constant level of 4 °C, and from there on, the sound speed increases depth (pressure). When a source launches a beam of rays, each ray will follow a slightly different path, and a receiver placed at some distance will observe multiple signal arrivals. Note that a ray traveling over a longer path may do so at a higher speed, thus reaching the receiver before a direct stronger ray. This phenomenon results in a non-minimum phase channel response.

The impulse response of an acoustic channel is influenced by the geometry of the channel and its reflection and refraction properties, which determine the number of significant propagation paths, and their relative strengths and delays. Strictly speaking, there are infinitely many signal echoes, but those that have undergone multiple reflections and lost much of the energy can be discarded, leaving only a finite number of significant paths.

To put a channel model in perspective, let us denote by l_p the length of the p th propagation path, with $p = 0$ corresponding to the first arrival. In shallow water, where sound speed can be taken

as a constant c , path lengths can be calculated using plane geometry, and path delays can be obtained as $\tau_p = l_p/c$.

The surface reflection coefficient equals -1 under ideal conditions, while bottom reflection coefficients depend on the type of bottom (hard, soft) and grazing angle (Jensen et al. 1994). If we denote by Γ_p the cumulative reflection coefficient along the p th propagation path, and by $A(l_p, f)$ the propagation loss associated with this path, then

$$H_p(f) = \frac{\Gamma_p}{\sqrt{A(l_p, f)}}$$

represents the frequency response of the p th path. Hence, each path of an acoustic channel acts as a low-pass filter, which contributes to the overall impulse response,

$$h(t) = \sum_p h_p(t - \tau_p),$$

where $h_p(t)$ is the inverse Fourier transform of $H_p(f)$.

Time Variability

There are two sources of the channel's time variability: inherent changes in the propagation medium and those that occur because of the transmitter/receiver motion. Inherent changes range from those that occur on very long timescales that do not affect the instantaneous level of a communication signal (e.g., monthly changes in temperature) to those that occur on short timescales and affect the signal. Prominent among the latter are changes induced by surface waves, which effectively cause the displacement of the reflection point, resulting in both scattering of the signal and Doppler spreading due to the changing path length.

It is beyond the scope of the present treatment to summarize what is known about the statistical characterization of these apparently random changes in the channel response. Suffice it to say that unlike in a radio channel, where a number of models for both the probability distribution (e.g., Rayleigh fading) and the power spectral density of

the fading process (e.g., the Jakes' model) are well accepted and even standardized, there is no consensus on statistical characterization of acoustic communication channels. Experimental results suggest that some channels may just as well be characterized as deterministic, while others seem to exhibit Rice or Rayleigh fading (Chitre 2007). However, current research indicates K-distributed fading in other environments (Yang and Yang 2006). Channel coherence times below 100 ms have been observed (Preisig 2007) but not often. For a general-purpose design, one may consider coherence times on the order of hundreds of milliseconds. In the absence of good statistical models for simulation, experimental demonstration of candidate communication schemes remains a de facto standard.

The Doppler Effect

The motion of the transmitter or receiver contributes additionally to the changes in channel response. This occurs through the Doppler effect, which causes frequency shifting as well as additional frequency spreading. The magnitude of the Doppler effect is proportional to the ratio $a = v/c$ of the relative transmitter-receiver velocity to the speed of sound. Because the speed of sound is very low compared to the speed of electromagnetic waves, motion-induced Doppler distortion of an acoustic signal can be extreme. Autonomous underwater vehicles (AUVs) move at speeds on the order of a few meters per second, but even without intentional motion, underwater instruments are subject to drifting with waves, currents, and tides, which may occur at comparable velocities. In other words, there is always some motion present in the system, and a communication system has to be designed taking this fact into account. The only comparable situation in radio communications occurs in low Earth orbiting (LEO) satellite systems, where the relative velocity of satellites flying overhead is extremely high (the channel there, however, is not nearly as dispersive). The major implication of motion-induced distortion is on the design of synchronization and channel estimation algorithms.

The way in which these distortions affect signal detection depends on the actual value of factor

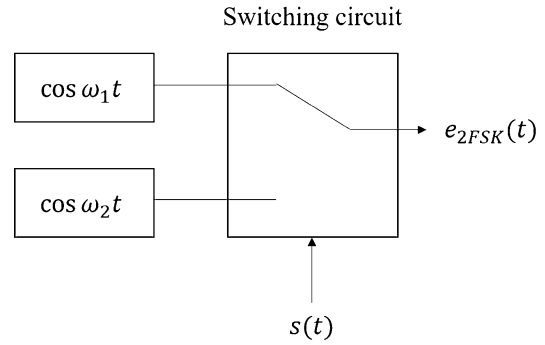
a. For comparison, let us look at a highly mobile radio system. At 160 km/h (100 mph), we have $a = 1.5 \times 10^{-7}$. This value is low enough that Doppler spreading can be neglected. In other words, there is no need to account for it explicitly in symbol synchronization. The error made in doing so is only 1/1000 of a bit per 10,000 bits. In contrast to this situation, a stationary acoustic system may experience unintentional motion at 0.5 m/s (1 knot), which would account for $a = 3 \times 10^{-4}$. For an AUV moving at several meters per second (submarines can move at much greater velocities), factor a will be on the order of 10^{-3} , a value that cannot be ignored.

Non-negligible motion-induced Doppler shifting and spreading thus emerge as another major factor that distinguishes an acoustic channel from the mobile radio channel and dictates the need for explicit phase and delay synchronization in all but stationary systems. In multi-carrier systems, the Doppler effect creates particularly severe distortion. Unlike radio systems, in which time compression/dilation is negligible and the Doppler shift appears equal for all subcarriers, in an acoustic system each subcarrier may experience a markedly different Doppler shift, creating non-uniform Doppler distortion across the signal bandwidth.

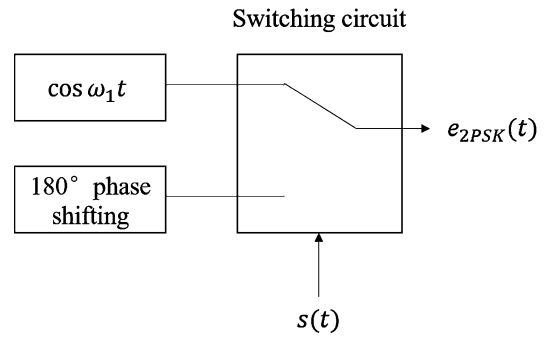
Modulation Schemes

Incoherent Modulation

Most early UWA communications systems used incoherent modulation methods for reasons of simplicity and reliability (Chitre et al. 2008), such as binary frequency-shift keying (2FSK), whose modulation process is presented in Fig. 2. Frequency-shift keying (FSK) is a frequency modulation scheme in which digital information is transmitted through discrete frequency changes of a carrier signal. However, there were a number of exceptions, in particular systems that were used in channels with little boundary action, for example, vertical links in deep water. Vertical links using directional transducers on unmanned underwater vehicles (UUVs) and ships are very clean, experiencing little or no delay spread, with the result that the biggest challenge is tracking the carrier phase that changes with respect to range.



Underwater Acoustic Communication, Fig. 2 Incoherent modulation (2FSK)



Underwater Acoustic Communication, Fig. 3 Coherent modulation (2PSK)

Coherent Modulation

A typical coherent modulation process (binary phase-shift keying, 2PSK) is shown in Fig. 3. Binary phase-shift keying (2PSK) is a digital modulation process that conveys data by changing (modulating) the phase of a constant frequency reference signal (the carrier wave). Through the 1980s phase, coherent communication was used almost exclusively for deep water vertical links, but in the early 1990s phase, coherent communication in multipath channels began to attract attention, as incoherent methods were limited to a bandwidth efficiency of approximately 0.5 bits per Hz.

Single Carrier Modulation

One major step toward high rate communication is single carrier modulation of information symbols from constellations such as phase-shift keying (PSK) and quadrature amplitude

modulation (QAM), whose block diagram is shown in Fig. 4 (Zhou and Wang 2014). With symbols $s[i]$ and pulse shaping filter $p(t)$, the transmitted signal is

$$x(t) = \sum_{i=-\infty}^{\infty} s[i]p(t - iT),$$

where T is the symbol period. The corresponding passband signal can be obtained by

$$\tilde{x}(t) = 2 \operatorname{Re} \{x(t)e^{j2\pi f_c t}\}.$$

The channel introduces intersymbol interference (ISI) due to multipath propagation. When data symbols are transmitted at a high rate, the same physical channel leads to more channel taps in the discrete-time equivalent model. Advanced signal processing at the receiver side is used to suppress the interference; this process is termed channel equalization. Although widely used for slowly-varying multipath channels in radio applications, channel equalization for fast-varying UWA channel is a significant challenge.

Multi-carrier Modulation

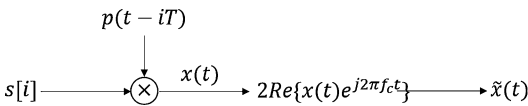
Multi-carrier modulation offers an alternative to a broadband single carrier communication. By dividing the available bandwidth into a number of narrower bands, orthogonal frequency-division multiplexing (OFDM) systems can perform equalization in the frequency domain and eliminate the need for complex time-domain

equalizers. The OFDM modulation process is described in Fig. 5. OFDM modulation and demodulation can easily be accomplished using fast Fourier transforms (FFT). A shallow water experiment in the Mediterranean Sea yielded good OFDM performance ($\text{BER} < 2 \times 10^{-3}$) at ranges by to 6 km (Frassati et al. 2005). At the same ranges, the DSSS performance was found to be significantly poorer.

OFDM systems often use a guard period (often implemented as a cyclic prefix or zero prefix) between consecutive OFDM symbols to avoid ISI. When the delay spread is long, the prefix length can significantly affect the bandwidth efficiency. Maximum likelihood sequence detection (MLSD) on individual subcarriers using a low complexity PSP can combat ISI when the symbol period is smaller than the delay spread (Morozov and Preisig 2006). Other channel shortening techniques such as sPRE may also be used in future OFDM systems to reduce the prefix length and improve bandwidth efficiency.

When using coded OFDM, consecutive symbols are often striped across subcarriers to reduce the error correlation due to fading. However, impulse noise present in some environments can affect multiple subcarriers simultaneously and hence generate correlated errors. The use of a channel interleaver with coded OFDM allows symbols to be distributed over frequency-time plane, thus allowing the code to make maximal use of frequency and time diversity offered by OFDM (Chitre et al. 2005). The knowledge of error correlation due to impulsive noise could be used in future decoding algorithms to improve decoding performance.

OFDM systems are very sensitive to the Doppler shift due to the small bandwidth of each subcarrier as compared to the Doppler shift. As the carrier frequency in UWA systems is typically low as compared to the Doppler shift experienced due to movement, the communication systems have to



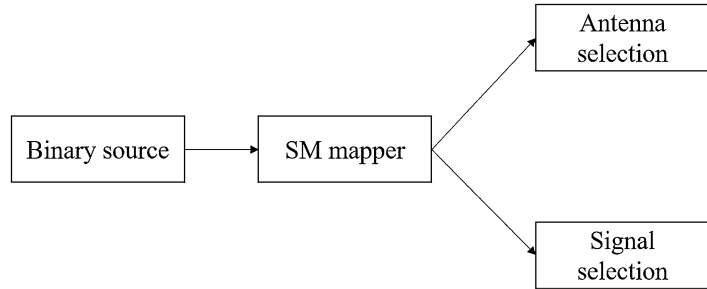
Underwater Acoustic Communication, Fig. 4 Single carrier modulation



Underwater Acoustic Communication, Fig. 5 Multi-carrier modulation

Underwater Acoustic Communication,

Fig. 6 Spatial modulation



cope with wideband Doppler which results in non-uniform Doppler shift across subcarriers. In Stojanovic (2006), the author presents an algorithm for non-uniform Doppler compensation in OFDM systems based on a single adaptively estimated parameter. In Sharif et al. (2000), the authors present a preprocessor that estimates Doppler shift by measuring the time between two known signals and removes the Doppler shift using a computationally efficient linear interpolator. Being a preprocessor, the technique can be used with any type of modulation and equalization.

Spatial Modulation

The process of spatial modulation (SM) is shown in Fig. 6. Information theoretic studies have shown that the capacity of a channel increases linearly with the minimum of the number of transmit and receive antennas. This increase in capacity translates to a corresponding increase in achievable data rate through the use of multiple input multiple output (MIMO) processing techniques and space-time coding.

Optimal detection techniques such as MAP and maximum likelihood sequence estimation (MLSE) exponentially grow in terms of complexity with the number of antennas. To address this problem, space-time trellis codes (STTC) and layered space-time codes (LSTC) can be used with suboptimal decoding techniques (Roy et al. 2004). The benefits of MIMO over single-input single-output (SISO) UWA communication systems were successfully demonstrated through an experiment in the Mediterranean Sea using two transmit projectors for STTC and four transmit projectors for LSTC. In another set of experiments

with six transmit projectors, a spatial modulation scheme with an outer block code, interleaver, and an inner trellis-coded modulation (TCM) was demonstrated (Kilfoyle et al. 2005). The experiments demonstrated that the proposed spatial modulation scheme offered increased bandwidth and power efficiency as compared to signals constrained to temporal modulation. For ISI-limited channels, spatial modulation offers the possibility of increasing data rates when simply increasing transmission power does not. In a MIMO-OFDM experiment, the nearly error-free performance was achieved with a 2-transmitter 4-receiver setup at ranges up to 1.5 km using a $\frac{1}{2}$ -rate low-density parity check (LDPC) code at a coded data rate of 12 kbps (Li et al. 2007).

Key Applications

As efficient communication systems are developing, the scope of their applications continues to grow, and so do the requirements on the system performance (Stojanovic 1999). Many of the developing applications, both commercial and military, are calling for real-time communication with submarines and AUVs, UUVs. Setting the underwater vehicles free from cables will enable them to move freely and refine their range of operation. The emerging communication scenario in which the modern UWA systems will operate is that of an underwater data network consisting of both stationary and mobile nodes. This network is envisaged to provide an exchange of data, such as control, telemetry, and eventually video signals, between many network nodes. The network nodes, located on underwater moorings, robots,

and vehicles, will be equipped with various sensors, sonars, and video cameras. A remote user will be able to access the network via a radio link to a central node based on a surface station. Moreover, UWA communication plays a crucial role in the frogman information system. Compared with other communication forms such as electromagnetic waves, UWA communication has a longer operating range and more reliable performance.

Cross-References

► Underwater Acoustic Communication

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Underwater Acoustic Environment

► Underwater Acoustic Sensor Network

Underwater Acoustic Sensor Network

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Synonyms

Acoustic modems; Ad hoc network; Cross-layer design; Energy efficiency; High latency; Infrastructureless network; Intelligent algorithm based; Medium access control layer; Multihop network; Neighbor discovery; Network experiment; Physical layer; Protocol design; Routing layer; Self-organizing network; Simulation tools; Underwater acoustic; Underwater acoustic channel; Underwater acoustic communication; Underwater acoustic environment; Underwater wireless acoustic communication

Definition

Underwater acoustic sensor network is composed of sensor nodes placed on the seabed and in the sea (including fixed sensor nodes and mobile platforms loaded with sensors), sea surface buoy nodes, and two-way acoustic links between them, which could cover a large area of underwater three-dimensional areas. Underwater acoustic sensor network can collect, process, classify, and compress information, and the information could be relayed back to the land-based or ship-based control center through underwater acoustic sensor network node.

Scientific Fundamentals

Underwater Acoustic Sensor Network Architecture

The network architecture is a basis for designing underwater acoustic sensor network; it can be divided into the following four types.

One-Dimensional Underwater Acoustic Sensor Network

One-dimensional underwater sensor network is composed of a single underwater acoustic communication unit; these underwater acoustic communication units collect the required information, process the collected information, and send it to the surface buoy or base station. This underwater acoustic communication unit can be a relatively fixed unit such as a seabed fixed node or anchor node; it can also be a mobile underwater acoustic node, AUV, and other underwater units with certain mobility. One-dimensional underwater acoustic sensor network usually adopts a star network topology, and the data is transmitted in a single hop between the network unit and the remote station on the surface.

After these independent hydroacoustic units have collected information, the collected information is processed, and then sent to the surface buoy or base station. This underwater acoustic communication unit can be a relatively fixed unit such as a seabed fixed node or anchor node; it can also be an underwater mobile node, AUV, and other

underwater units with certain mobility. Due to the networking architecture and task requirements, the one-dimensional underwater acoustic sensor network usually adopts a star network topology, and the network unit and the remote station on the surface adopt a single-hop method to transmit data.

Two-Dimensional Underwater Acoustic Sensor Network

Different from one-dimensional underwater acoustic sensor network, two-dimensional underwater acoustic sensor network consists of multiple underwater acoustic communication nodes. All nodes in the network are at the same depth underwater. Usually, the nodes in the network are divided into two categories, ordinary nodes and sink nodes. Ordinary nodes only have horizontal communication capabilities, not vertical communication capabilities. The ordinary node collects and processes the required information and sends the data to the sink node through the horizontal link. The sink node has both horizontal and vertical communication capabilities. Sink nodes receive data packets from ordinary nodes through the horizontal link, and after processing, upload the data packets to the surface buoy or base station through the vertical link. Surface buoys or base stations upload data to satellites or send data to shore-based command centers through radio frequency communication. Two-dimensional underwater acoustic sensor network constitutes a variety of network topologies such as star, grid, or ring according to the needs of the tasks performed.

Three-Dimensional Underwater Acoustic Sensor Network

Different from two-dimensional underwater acoustic sensor network, the nodes in three-dimensional underwater acoustic sensor network are not completely at the same depth underwater. Three-dimensional communication can be carried out between network nodes. Nodes of different depths are clustered according to their depths, and nodes of the same or similar depth are clustered in the same cluster. Nodes have both horizontal and vertical communication capabilities.

Underwater Acoustic Sensor Network,

Fig. 1 Layered network model of underwater acoustic sensor network

Underwater acoustic sensor networks	OSI	TCP/IP
	application layer	application layer
	presentation layer	
	session layer	
	transport layer	transport layer
network layer	network layer	network layer
data link layer	data link layer	data link layer
physical layer	physical layer	

Nodes can communicate with nodes in the same cluster through horizontal links or communicate between clusters or with surface buoys or base stations through vertical links.

Four-Dimensional Underwater Acoustic Sensor Network

Four-dimensional underwater acoustic sensor network is composed of a static three-dimensional underwater acoustic sensor network and mobile underwater mobile acoustic communication units. Mobile underwater acoustic communication units include AUVs, mobile nodes, etc. The information collected by the static network can be sent to the mobile underwater acoustic communication unit, and then sent to the surface buoy or base station through the mobile acoustic communication unit. Since the mobile acoustic communication units in four-dimensional underwater acoustic sensor network have good mobility, the underwater acoustic sensor network of this structure can also combine optical communication and radio communication to meet more different needs.

Layered Network Model

Compared with land-based radio communication network, the research on underwater acoustic sensor network is still in its infancy. From the perspective of practical application, in order to reduce the complexity of network design, referring to the open systems interconnection (OSI), model, and transmission control protocol/internet protocol (TCP/IP) proposed by the International Organization for Standardization, underwater

acoustic sensor network is mainly divided into the physical layer, data link layer, and network layer, as shown in Fig. 1. The details are as follows:

Physical Layer

Physical layer is responsible for realizing the modulation and demodulation of information flow data. Its main function is to provide transmission media and interconnection equipment for data communication between nodes, and to provide a reliable environment for data transmission. Modulation is performed at the transmitting end, and the digital information represented by “0,” and “1” is modulated into an acoustic signal for propagation in the underwater acoustic channel. At the receiving end, the received acoustic signal is demodulated, and the original bit stream information is restored. Physical layer only transmits digital modulation information composed of “0” and “1” and does not care about the specific meaning of each bit.

The goal of the physical layer communication algorithm is to improve the reliability and effectiveness of the communication system as much as possible. Because the multipath effect of the underwater acoustic channel causes signal amplitude fading and inter-symbol interference, the channel bandwidth is limited (Stojanovic and Preisig 2009). The main technologies currently used in the underwater acoustic sensor network physical layer include phase shift keying (PSK), multiple frequency shift keying (MFSK), and orthogonal frequency division multiplexing (OFDM). In addition, in order to achieve robust

underwater acoustic communication, the key technologies commonly used in underwater acoustic communication include decision feedback equalization technology, time reversal mirror technology, sparse channel estimation and equalization technology, and wideband doppler compensation technology.

Data Link Layer

Data link layer is the middle layer of underwater acoustic sensor network, and its goal is to achieve fair and effective sharing of underwater acoustic channel resources by each network node. When designing the data link layer, while pursuing high throughput, minimize signal delay, and node power consumption. The design and implementation of the MAC protocol is the main research content of the data link layer of the underwater acoustic sensor network. The design of a MAC protocol with high efficiency, high throughput, and low energy consumption is very important to improve the performance of the underwater acoustic sensor network. Although there are many classic, mature, and efficient MAC protocols for land-based wireless sensor network, due to the big differences between underwater acoustic sensor network and land-based wireless sensor network, the MAC protocol suitable for land-based wireless sensor network cannot work efficiently in the underwater acoustic sensor network, and the MAC protocol needs to be designed according to the characteristics of the underwater acoustic sensor network.

MAC protocols in underwater acoustic communication are usually divided into two categories, namely, scheduled-based and random competition (Climent et al. 2014). The scheduled-based MAC protocols mainly include time division multiple access (TDMA), frequency division multiple access (FDMA), and code division multiple access (CDMA) protocols. Random competition protocols mainly include ALOHA, carrier sense multiple access (CSMA), multiple access with collision avoidance (MACA), etc. Due to the complex application scenarios of underwater acoustic sensor network, sometimes a single multiple access technology cannot achieve good

results, and different multiple access technologies should be used in combination.

Network Layer

Based on the data link layer providing the transmission function of data frames between two adjacent nodes, the main role of the network layer is to further manage the data communication in the network, through several intermediate nodes, and try to transfer data from the source node to the destination node. Its specific functions include addressing and routing, connection establishment, maintenance and termination, etc. Each node in the network must work together to complete the route selection.

The design goals of routing protocols are mainly fast, accurate, and efficient have good scalability, and can quickly find routes. As the topology of underwater acoustic sensor network often changes, routing protocols also could quickly establish routes according to network topology changes (Domingo 2011). While selecting the best path according to the congestion of the network, the control information of maintaining routing should be minimized to reduce the overhead of routing protocol.

Evaluation Index

Evaluation of the performance of underwater acoustic sensor network mainly uses indicators such as average end-to-end delay, throughput, packet loss rate, and overload.

End-to-End Delay

End-to-end delay refers to the average time required from the sending node to the destination node to successfully receive the data packet – the unit is s. End-to-end delay generally consists of three parts: the time the application layer generates the data packet and the time the data packet waits in the cache queue; the time it takes for the data packet to be transmitted and received in the transmitter and receiver; and the time of data packet transmission in the channel.

Throughput

Throughput refers to the ratio of the amount of data packets successfully received to the unit simulation

time; the unit is bps, which is represented by TP in this entry and can be expressed as

$$TP = (Num_{pr} \times p_{size}) / T$$

where Num_{pr} is the number of successfully received data packets, p_{size} is data package size, and T is simulation time. The average throughput is used to evaluate the efficiency of underwater acoustic sensor network data transmission. The higher the throughput, the greater the amount of data transmitted per unit time, and the better the performance of underwater acoustic sensor network.

Packet Loss Rate

Packet loss rate refers to the ratio of lost data packets to the total number of data packets sent in the network, used to evaluate the stability of the underwater acoustic sensor network. It is represented by L_{rate} in this entry and can be expressed as

$$L_{rate} = (Num_{lt} \times Num_{st}) \times 100\%$$

where Num_{lt} is the number of lost data packets and Num_{st} is the total number of data packets sent in the network.

Overload

Overload refers to the ratio of the total amount of data generated by the network protocol control message to the unit simulation time; the unit is bps, which is represented by S in this entry and can be expressed as

$$S = Num_c \times c_{size} / T$$

where Num_c is the number of control packets successfully received, c_{size} is control package size, and T is the simulation time. The greater the overload, the more energy is consumed. Therefore, the overload should be reduced as much as possible when designing the network protocol.

Simulation Tools

In order to evaluate the performance of underwater acoustic sensor network, it can usually be quantitatively evaluated through simulation or

experiment. Underwater acoustic sensor network tests usually require a lot of preparation work and need to wait for suitable sea conditions and other test conditions, which require a lot of manpower and material resources, and due to the limitations of funds and sea conditions, the network performance test verification is not comprehensive. Software simulation is an effective tool for network performance evaluation. There are many software for network simulation, such as NS2, NS3, and OPNET.

NS2

NS2 is an open-source network simulation software based on the split model developed by the University of California, Berkeley. It has the characteristics of expansion and high efficiency. It can work under Windows, Linux, Unix, or Machintosh operating systems. When using NS2 for network simulation, two programming languages are generally used: C++ and OTcl. The former is used to implement the various modules of the network protocol stack, and the latter is used to write some modules and simulation scripts to configure the network topology, protocol stack, simulation time, and other parameters. The simulation results output by NS2 can be further analyzed using related tools. For example, network animator (NAM) can be used to generate a dynamic network topology map, which can clearly show the process of data packet transfer in the simulation process.

Although NS2 has powerful functions and can support wired and wireless network simulation, the publicly released NS2 software cannot be directly used for underwater acoustic sensor network simulation because its wireless channel model and physical layer communication module are not suitable for underwater acoustic sensor network. Therefore, some research institutions have improved the physical layer and channel based on NS2 and developed a series of enhanced versions that can be used for UACN simulation, such as Auqa-Sim, NS-MIRCLE, DESERT, and SUNSET (Petrioli and Petrocchia 2012).

NS3

NS3 is a discrete event simulation software for network systems developed since 2006, using

test			
helper			
protocols	applications	devices	...
internet		mobility	
network			
core			

Underwater Acoustic Sensor Network, Fig. 2 The structure of NS3

GNU GPLv2 open-source agreement. This software is not an extension of NS2 and does not support NS2 application programming interface functions. The structure of NS3 is shown in Fig. 2. The core module is the kernel module of NS3, which is the foundation of the entire NS3 operation and development, including scheduling module, callback module, tracking module, etc. Above the core module is the network module, which contains abstract components such as nodes, network cards, address types, sockets, and packets. The upper layer of NS3 provides helper classes for each module. When writing simulation scripts, users can ignore the details of the module implementation and use the APIs in the helper classes of each module to call the components provided by the module. Above the helper class is the test class, which contains content related to software testing in the module development process.

NS3 provides a visualization tool that can dynamically track the flow of data packets in the simulation process. As one of the cores of NS3, the data collection system allows users to easily track the data or events of interest in all directions, and finally collect and visualize output through the interface provided by the system. NS3 is still under continuous development and improvement. The publicly released version includes a simple underwater acoustic sensor network model (Tracy and Roy 2008) that can simulate the MAC protocol of centralized underwater acoustic sensor network.

OPNET

OPNET is a commercial discrete event network simulation software, which uses a three-layer

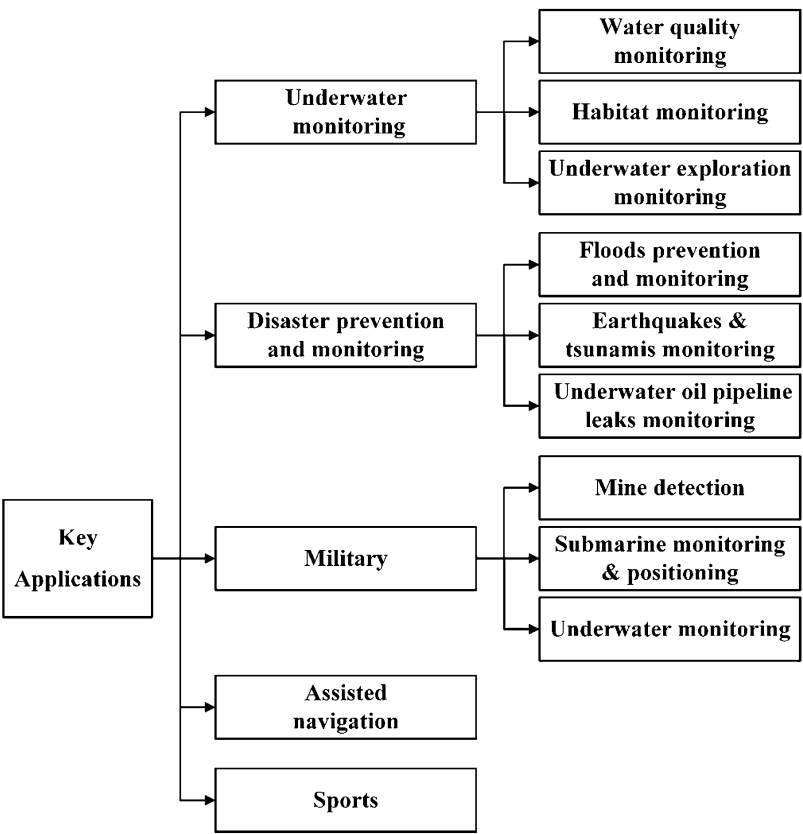
model of process model, node model, and network model to simulate network behavior. The network model is the uppermost layer, which is composed of nestable subnets, communication nodes, and links for communication between nodes. The network topology and model configuration are completed at this layer; the process model is the lowest layer, using a finite state machine (FSM) describing each state and the transition relationship between states. This layer is the specific location for the realization of communication protocol function simulation and simulation-related control behavior. FSM is a communication behavior program described in C language; the node model defines the internal structure of the node, which consists of a transmitter module, a receiver module, a processor module, and a queue module.

Currently, researchers have applied OPNET to underwater network simulation. Mandar et al. (2014) used OPNET to simulate the influence of the channel on the underwater acoustic network, but the lack of reliable underwater acoustic channel model and open source underwater acoustic protocol, as well as the ability to simulate 2D underwater network and other factors limit the simulation effect of OPNET. As a commercial software, the scope of application of OPNET is not as good as the open source simulation software NS2 and NS3, but its convenient way of use is still favored by many scholars.

Key Applications

With the continuous exploitation of land resources, people’s development and competition for marine resources has become increasingly fierce. As one of the most powerful helpers for marine development, underwater acoustic sensor network has a wide range of services in both military and civilian fields. In this section, various applications of underwater acoustic sensor network will be introduced. As shown in Fig. 3, the applications of underwater acoustic sensor network can be divided into five aspects: underwater monitoring, disaster prevention and monitoring, military, assisted navigation, and sports.

Underwater Acoustic Sensor Network,
Fig. 3 Applications of underwater acoustic sensor network



Underwater Monitoring

Underwater monitoring refers to monitoring the characteristics of the waters in a designated area by deploying an underwater acoustic sensor network. The applications of underwater acoustic sensor network in underwater monitoring mainly include water quality monitoring (Faustine et al. 2014), habitat monitoring (Lopez et al. 2009), and underwater exploration monitoring (Srinivas et al. 2012).

Disaster Prevention and Monitoring

Natural disasters are usually unavoidable, and natural disasters in water are often more dangerous and cause more serious losses. Therefore, it is very necessary to establish an effective underwater natural disaster monitoring and prevention mechanism. Underwater acoustic sensor network has been widely used in the monitoring and prevention of underwater natural disasters. It is used to monitor and prevent floods (Udo and Isong

2013), underwater volcanic eruptions (Casey et al. 2010), underwater earthquakes and the resulting tsunamis, underwater oil pipeline leaks (Iwendi and Allen 2011), and other types of water disasters.

Military

The core of modern warfare is the network. As a part of the integrated land, sea, air, and space network, underwater acoustic sensor network is becoming more and more important in the military. An efficient and reliable underwater acoustic sensor network can greatly assist naval operations. Related researchers have conducted a lot of research on the military application of underwater acoustic sensor network. It has been widely used in mine detection (Williams 2010), submarine monitoring and positioning (Hamilton et al. 2010), underwater monitoring (Caiti et al. 2013), and other fields.



Assisted Navigation

The underwater environment has great unknowns, uncertainties, and randomness, and these instabilities tend to increase with the increase of depth. In such an unknown and dangerous environment, ships, boats, and even diving enthusiasts need reliable navigation mechanisms to escort them. Due to the particularity of underwater channel conditions, navigation technology on the land cannot be applied well underwater. Therefore, it is very important to design a navigation system that can work well underwater. Through the unremitting efforts of relevant researchers, underwater acoustic sensor network technology has been widely used in the field of underwater-assisted navigation (Guo and Liu 2013).

Sports

Underwater acoustic sensor network also has a very wide range of applications in water movement. These applications are slightly different from the previous types of applications. Sage et al. (2011) propose an underwater acoustic sensor network for monitoring swimmers' swimming performance. The swimming performance data monitored by the network can be transmitted to swimmers and coaches at the same time. Marshall (2015) focuses on locating swimmers in swimming pools through the underwater acoustic sensor network, and the experimental results show that this method is more accurate than traditional inertial positioning.

Cross-References

- [Ad Hoc Network](#)
- [Underwater Acoustic Sensor Network](#)

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Underwater Connector

► Subsea Connector

Underwater Device

► Photoelectric Detection Technology in Underwater Vehicles

Underwater Distributed Remote Sensing

► Underwater Information Sensing Technology

Underwater Driving Subsystem

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Synonyms

SPDM – Self-propelled type driving mechanism;
TSDM – Tow-sled type driving mechanism

Definition

Underwater Driving Subsystem – A mechanism or device for driving mining vehicles to carry out mining operations on the seabed.

Introduction

The driving mechanism of the deep-sea mining vehicle bears the body weight, interacts with the seabed or overlying sediments to realize the vehicle's movement on the seafloor. Studies on mining

vehicles for polymetallic nodule exploitation have been taken since the 1970s. Because of the relatively flat topography of the seabed in the polymetallic nodule mining area, as well as the mechanical properties of the surface sediment, such as high water content, small bearing strength, and shear strength, researchers have developed a variety of driving mechanisms based on different driving techniques. Typical underwater driving mechanisms include the Tow-sled type, the Archimedes spiral propulsion type, and the crawler self-propelled type.

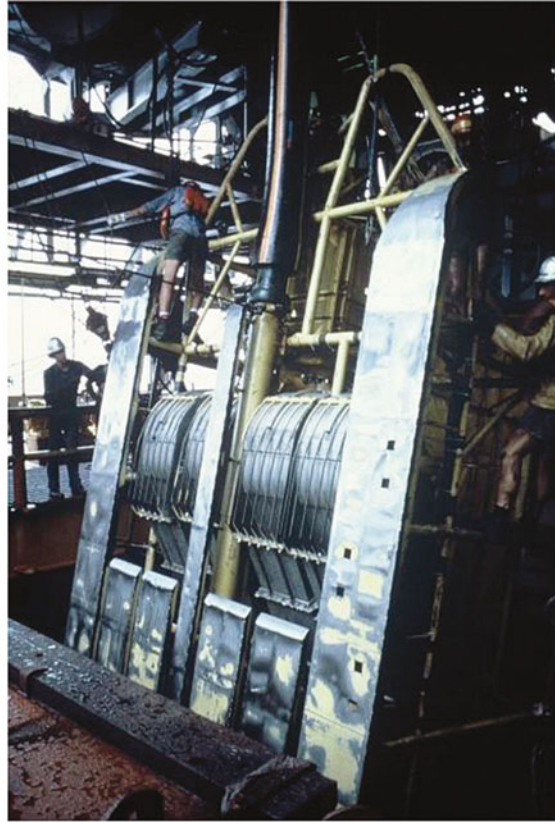
In the late 1970s, some countries carried out sea trials of Towed type and Archimedes type mining vehicles. Although these two ways are technically feasible, they are not suitable for commercial mining due to low mineral collection rates and difficulty in avoiding obstacles. After that, the focus of researchers gradually turned to the development of crawler driving technology and mechanism.

The Tow-Sled Type Driving Mechanism (TSDM)

TSDM usually uses a sled-type bearing chassis. With the traction of the sea surface mining vessel through the lifting pipe, it is available to drive on sediments with low bearing capacity or shear force, and to cross some slopes and small obstacles. In 1978, Ocean Management Incorporated Corporation (OMI) successfully completed three mining tests at a depth of more than 5000 m using a tow-sled hydraulic mining machine (as shown in Fig. 1a) jointly developed by researchers from Germany, Japan, and the United States (ISA 2008).

The mechanism has the advantages of a simple structure and little disturbance or damage to the seabed. However, the mechanism cannot accurately control the direction of travel or avoid obstacles autonomously, and cannot travel in accordance with the predetermined track and preset speed, which will reduce the efficiency of seabed mineral resources collection and cause resource waste. Therefore, this kind of mechanism

Underwater Driving Subsystem, Fig. 1 Tow-sled type driving mechanism. (Yamazaki et al. 1999). (a) Bottom view. (b) Improved tow-sled mining vehicle



(a) bottom view



(b) improved tow-sled mining vehicle

is only suitable for the principle experiments, not for commercial exploitation.

Figure 1b shows an improved tow-sled mining vehicle used in towing mining experiment on a sea mountain with a depth of 1700 m in 1997 (Yamazaki et al. 1999).

The Self-Propelled Type Driving Mechanism (SPDM)

The self-propelled type driving mechanism is powered by the surface mining vessel through armored cables which could deliver a high

power level. According to the operator's remote-control instructions, it has the ability to drive along the preset path and control the mode with good surmounting and pass obstacles. Moreover, it can change the driving speed according to nodule abundance to maintain constant production capacity. The self-propelled type driving mechanism is mainly divided into the Archimedes spiral propulsion type and the crawler self-propelled type.

Archimedes Spiral Propulsion Type Driving Mechanism

The Archimedes spiral propulsion type driving mechanism was originally developed by the US Navy to use as a vehicle in swampy areas. In 1979, the Ocean Minerals Company Corporation (OMCO) carried out a mining test in the Pacific polymetallic nodule mining area. The mining vehicle used in the test adopted this kind of driving mechanism, as shown in Fig. 2a (Schick 1980). Figure 2b shows that the mining vehicle in the shuttle boat shape, developed by French scientists, also uses this type of mechanism (Herrouin et al. 1989).

The driving mechanism is driven by hydraulic pressure and has a simple structure. The propelling force generated by the high-speed rotation of the spiral blade in the sea mud makes the mechanism move forward or backward. Despite good traveling performance in swampy areas and snowfields, it is almost impossible to move on hard rock.

Subsequently, researchers from France, Japan, Germany, Russia, and China conducted extensive research on the Archimedes type of mechanism. After comparing with the crawler type driving mechanism, the following conclusions were drawn on its disadvantages (Wang 2015).

- (a) The Archimedes type's static depression depth is much lower than that of the crawler type, which means its bearing capacity is poor.
- (b) The traction force per vehicle weight of the Archimedes type is far less than that of the crawler type, while the driving power is far more than that of the crawler type. Also, the ratio is about 40:7.4.

- (c) It has difficulty in obstacles surmounting and turning around.
- (d) The spiral groove is easily trapped by the sediment, which will significantly affect the driving performance.

Therefore, whether the Archimedes spiral propulsion type driving mechanism is suitable for commercial mining on the soft seabed remains to be further studied.

Crawler Self-Propelled Type Driving Mechanism

The crawler self-propelled type driving mechanism adopts the mature universal construction machinery driving technology. A benefit from the large grounding area is that it can be well attached to the seabed and produce high traction. Meanwhile, its good maneuverability, as well as its small impact on the environment, makes it especially suitable for polymetallic nodule mining areas.

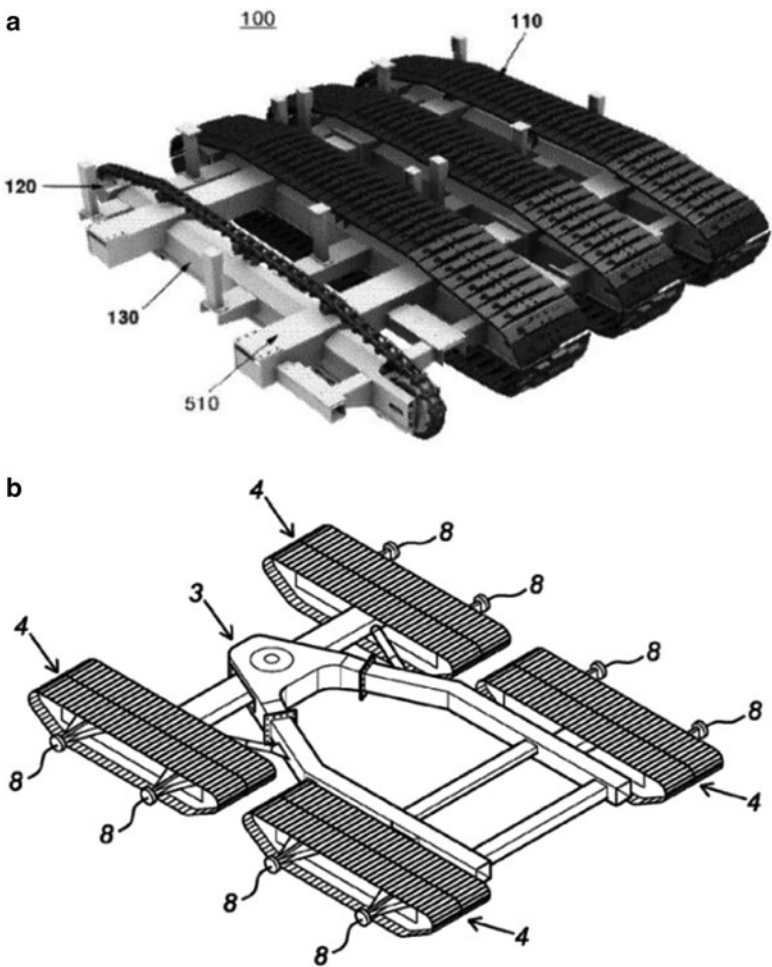
Crawler Driving Technology and Mechanism on Soft Sediment in Polymetallic Nodule Mining Areas

During the operation of a deep-sea mining vehicle, its driving mechanism interacts with the seabed surface sediment. On the one hand, the driving mechanism cuts and depresses the sediment. On the other hand, the sediment drives and bears the driving mechanism. Compared with land soils, seabed sediments (within 20 cm below the seabed surface) are characterized by high water content, large fluidity, low shear strength and compressive strength, and small friction angle. Within a certain range, the shear strength and compressive strength of the sediment gradually increase with the increase of depth. The deeper the driving mechanism is depressed, the greater the traction is generated. However, the driving resistance will be sharply increased if the depression depth is too large. Ultimately, it affects the driving performance and causes a great disturbance to the marine environment. This puts forward a larger emphasis on the design of the mining vehicle's driving mechanism, that is, to reasonably design the weight of the mining

Underwater Driving Subsystem,
Fig. 3 Double-crawler mining vehicle of 3500-meter deep-sea polymetallic nodules developed in China. (COMRA 2020)



Underwater Driving Subsystem, Fig. 4 Four-crawler driving mechanisms developed by South Korea (a) (Sup et al. 2016) and GSR of Belgium (b) (Johan et al. 2018), respectively



double-crawler mining vehicle of 3500-meter deep-sea polymetallic nodules developed in China.

Figure 4 shows the four-crawler driving mechanism of mining vehicles, respectively, developed by South Korea (a) (Sup et al. 2016) and GSR of Belgium (b) (Johan et al. 2018). These two mining vehicles have completed the sea test.

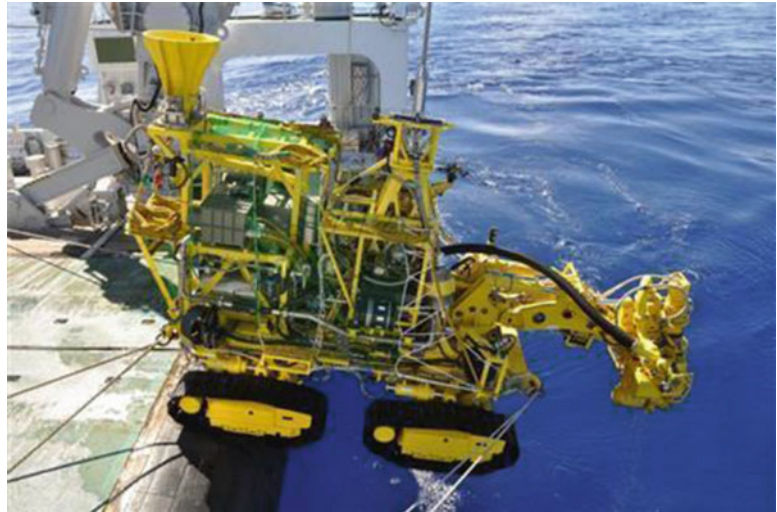
Crawler Driving Technology and Mechanism on Hard Ground and Complex Terrain

According to previous exploration data, the polymetallic massive sulfide deposits and cobalt-rich crust plate deposits at the top of the seamount

have complex micro-topographic features, and the mining compressive strength of the block ore body is no less than 20 MPa, which requires the driving mechanism of the mining vehicle to have good obstacle crossing abilities and adhesion. To develop driving technology and mechanisms of mining vehicles for polymetallic sulfide and cobalt-rich crust, it is possible to learn from some relatively mature technologies and mechanisms of heavy-duty operations in complex terrains, such as vehicles from mountainous areas, or combat vehicles. With different mining processes and test purposes, the driving mechanism developed may differ.

Underwater Driving Subsystem,

Fig. 5 Polymetallic sulfide mining vehicle. (Nobuyuki et al. 2018)



Underwater Driving Subsystem, Fig. 6 Four-crawler driving mechanisms of cobalt-rich crusts mining vehicle. (CIMR 2018)



Aiming at the commercial exploitation of polymetallic sulfide in the Sorawa-1 mining area of Papua New Guinea, Nautilus Minerals has developed a three-step process of mining area leveling, ore cutting, and particle collection in reference to the open-pit mining process. Based on this process, Nautilus has developed three operating vehicles with double crawler driving mechanisms and carried out tests underwater. The auxiliary cutter begins the work by grinding down the seafloor to make it level enough for the second piece of equipment, the bulk cutter. That machine grinds the resulting slurry up fine enough for the collection machine to suck it up before it is sent to a ship on the surface. Two four crawler mining vehicles were developed based on the mining process of cutting and collecting simultaneously, as shown in Figs. 5 and 6. In 2017, a polymetallic sulfide mining test at a depth of 1700 meters was completed in the Okinawa sea area (Ishiguro et al. 2013; Nobuyuki et al. 2018; Saekyeol et al. 2019). In 2018, a verification test of mining sampling at a depth of 2019 meters was completed in the cobalt-rich crusting contract area of the Western Pacific Ocean (CIMR 2018).

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Underwater Equipment Fix Techniques

- [Subsea Equipment Installation Technology](#)

Underwater Glider (UG)

- [Glider](#)

Underwater Information Awareness

- [Underwater Information Sensing Technology](#)

Underwater Information Sensing

- [Underwater Information Sensing Technology](#)

Underwater Information Sensing Technology

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Synonyms

Acoustic correlation current profiler (ACCP); Acoustic Doppler current profiler (ADCP); Autonomous underwater vehicle (AUV); Conductivity, temperature, depth (CTD); Environmental characterization optics (ECO); Sound, velocity, temperature, pressure (SVTP); Underwater distributed remote sensing; Underwater information awareness; Underwater information sensing; Underwater surveillance; Unmanned surface vehicle (USV)

Introduction

Definition

Underwater information perception is a process of gathering sea environment and target signal with acoustic, optical, magnetic, electrical, and other sensors; processing the signal; and obtaining the characteristics of the sea environment and target. Since the sound wave can transmit for a long range in water, the acoustic information attracts extraordinary attention. The final purpose of underwater information perception is to develop ocean underwater surveillance system, understand the variation law of ocean environment, construct a transparent ocean, and help the mankind develop and utilize the ocean.

Function of Ocean Information Perception

In the era of informatization, if we humans want to study and utilize the ocean, we should first perceive the ocean, gather ocean information, and explore the law of ocean.

Ocean information involves a wide range of subjects, including all the aspects of ocean physics, ocean chemistry, ocean biology, ocean

geology, etc. Mankind has invented a variety of ocean surveillance equipment to observe the ocean, including coast-based ocean wave monitoring station, ocean meteorological station, ocean tide level station, coastal hydrophone station, high-frequency ground wave radar, etc. The buoy and ocean measuring ship are equipped with the synthesis aperture sonar (SAS), the acoustic Doppler current profiler (ADCP), the acoustic correlation current profiler (ACCP), the optical apparatus, and multiple kinds of seawater sensors. The meteorological satellite and water-color satellite are equipped with a variety of remote sensing and remote testing apparatus. The seabed-based platforms are equipped with a variety of sensors to measure the ocean dynamic parameters. All the sensors work together to sensing the ocean in three dimensions. The information is analyzed, computed, deduced, and signal processed, to form the ocean wind speed, wind direction, air temperature, air pressure, relative temperature, amount of rain, visibility, tidal wave, sea current vector, sea current profile, surface wind field, surface water temperature and salinity, profile water temperature and salinity, sea depth, ocean magnetic field, seabed configuration, evaporation flux, and the formation mechanism of atmospheric duct, the formation mechanism of sea area flow field and convergence zone, and other ocean physical knowledge library and ocean parameter knowledge library. All the knowledge of ocean environment factors forms a grand ocean ambient situation map, used for evaluating and forecasting the effect of ocean environments on all kinds of ocean exploiting activities (Fang 2005).

Functions of Underwater Information Perception

The physical fields to perceive include underwater acoustic field, ocean magnetic field, ocean electrical field, ocean optical field, etc. We are interested in the ocean environment information and target information, such as the characteristics of sea ambient noise, underwater target, and sound propagation, which are very important to underwater acoustic research and engineering applications. The ocean environment is complex and variable,

changing with area and time. The factors affecting sound propagation include the temperature-salt-depth profile, ocean current profile, turbulence, internal wave, frontal surface, sea-bottom sound speed, etc. (Ling 2013).

Ocean underwater information is of great importance in mankind's exploring activities. For example, the density discontinuity layer in seawater affects the safety of underwater vehicles. As a vehicle goes into the lower density layer, the buoyancy reduces suddenly, resulting in a sharp descent of the vehicle, which may cause serious accidents. The internal wave has effects on the controllability and safety of the vehicle. Tide waves and ocean currents may deviate the vehicle from its course.

Sound propagation characteristic is the most important factor that affects the acoustic information perception in the sea. Sound propagation in the sea mainly depends on the distribution of temperature, salinity, and density of seawater. In the deep sea, the difference in sound speed at vertical plane forms the convergence zone and acoustic shadow zone. An underwater vehicle may detect an object at a long range by making use of the convergence zone, and hide itself in the shadow zone. In the shallow sea, sound propagation is affected by meteorological and seabed conditions, which makes the propagation characteristics more complex than in the deep sea (He et al. 2015).

As a background interference to underwater information perception, the sea ambient noise has important effect on sonar operating range forecasting and sonar signal processing. Studying the characteristics of the sea ambient noise, including the time and frequency domain feature, spatial distribution, and correlation coefficient, taking full advantage of the differences between sea ambient noise and moving sound sources, to suppress the interferences, are very important to detecting moving source signal buried in strong interferences. The regularity in sea ambient noise may be exploited to estimate the relative ocean parameters, such as the surface wind speed, the amount of rain, the shipping distribution, etc. The above information is also used for the study of oceanography and ocean climate (Chen 2006).

Information perception for the moving sound sources is necessary for national defense security as well as the study of the sea environment. The moving sound sources include military ships, merchant ships, and other underwater vehicles. Information perception of underwater moving sources includes two cases. In the first case, the target passes not far away from the observer and may be looked as a dominant concentrated source, and the noise information acquired can be used for target acoustic characteristics analysis. In the other case, when the moving source is far from the observer, it actually contributes to the sea ambient noise. Usually, the contribution is within the lower frequency band below several hundred Hertz. Sometimes it reaches the frequency band above 1 kHz near a busy sea route.

Scientific Fundamentals

Underwater Information Perception System

An underwater information perception system includes shore-based sonars, ship-carried sonars, seabed-based sensor arrays, anchored buoys, floating buoys, autonomous underwater vehicles (AUVs), underwater gliders, wave gliders, unmanned surface vehicles (USVs), etc. Those systems carry sensors to observe and gather underwater information.

Ocean observing and information gathering in a great period and space is an important developing trend of ocean information perception. The rapid development of underwater unmanned system provides a wide-range underwater information perception capability with an effective approach. Underwater unmanned systems can be divided into two types: the stationary type and the mobile type. The stationary type mainly includes the seabed-based sensor array and anchored buoy, etc. The mobile type mainly includes AUVs and underwater gliders, etc. Both types have advantages and disadvantages. The mobile type is more flexible when performing a task since it can move. It can move into dangerous or sensitive areas to observe clearly at a short range. The advantage of stationary type are relatively simple in structure, having no dynamic energy expenditure,

supporting long-time observation and data recording, not generating self-noise, and observing quietly.

Structure of Underwater Information Perception System

The underwater information perception system mainly includes the carrier, central control unit, sensors, signal conditioner, data recorder, and battery. A digital signal processor (DSP) will be arranged when real-time data processing is required. A communication unit will be allocated when real-time transmission of the information is requested. The structure of an underwater information perception system is shown in Fig. 1.

The underwater information perception system can carry a variety of sensors. Sensors used to sensing acoustic signals include sonars, hydrophones, acoustic vector sensors, etc. For the purpose of making spatial directivity and improving the sensibility to weak signals, the sensors sensing the target and noise information usually operate as an array, such as the hydrophone array, magnetic sensor array, electric sensor array, etc. Sensors used to sense the attitude of a platform are magnetic compasses and pressure sensors. Sensors used to sense the ocean environment include CTD (conductivity, temperature, depth), SVTP (sound, velocity, temperature, pressure), ADCP, ACCP, DO (dissolved oxygen) sensor, ECO

(environmental characterization optics) sensors, gamma-ray sensors, etc.

The signal conditioner is used to filter and amplify sensor signals. The data recorder is used to record the collected digital signals, with a capacity of several months. The DSP is used to analyze and detect the collected signals, extract the signal features, compute the spatial distribution and motion state, and recognize the attribute of the source. The communication unit is used to transmit information among the distributed sensor nodes with the underwater acoustic modem. Sometimes the underwater information is needed to transmit to the radio data link, and the underwater acoustic/radio unit is used, or a communication buoy is released up to the surface. The mobile nodes such as the AUVs or underwater gliders can rise up to the surface, and transmit information with a radio unit.

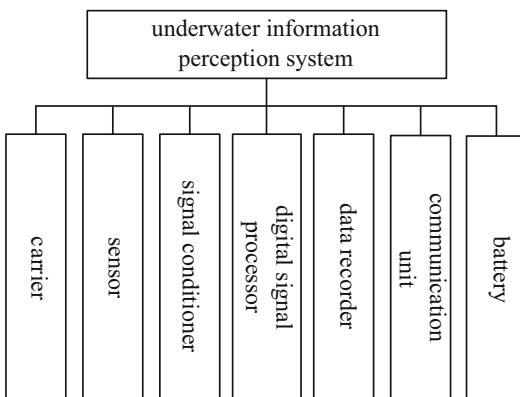
Underwater Information Perception System of Stationary Type

An underwater information perception system of stationary type gathers information of underwater target or environment by sitting at the seabed or mooring in water.

The real-time transmission sub-buoy is a single-point moored system used to monitor ocean environment, as shown in Fig. 2. It can carry ADCPs, CTDs, vertical linear arrays, and vector acoustic hydrophones to gather environmental and sound source information. It boasts the advantages of both sub-buoy and surface buoy in that it is capable to conduct long-time continuous secret monitoring of deep-sea oceanographic environment and timely transmit the collected data to shore-based station. This highly integrated system features secrecy, robust against sabotage, and timely transmission and can be applied to early warning of ocean disaster and military attack.

The main performance index is as follows:

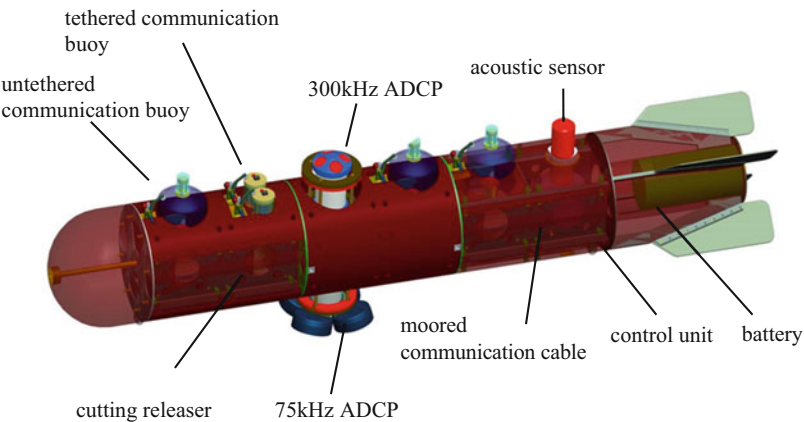
1. Operation depth: 4000 m, deployment depth of the mother buoy: 80~300 m
2. Effective range of current measurement and profiling: 0~500 m (double ADCP); temperature and salinity depth: 800 m



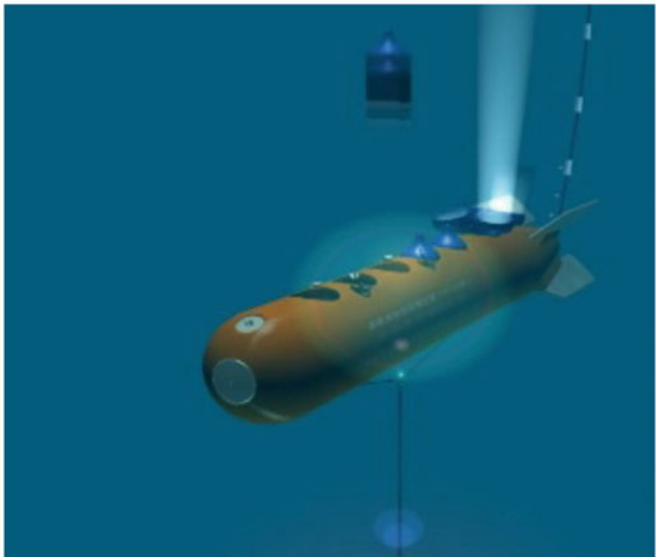
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Fig. 1 Structure of underwater information perception system

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Fig. 2 Sub-buoy made by Yichang Research Institute of Testing Technology. **(a)** Sub-buoy. **(b)** Releasing untethered communication buoy



(a) Sub-buoy



(b) Releasing untethered communication buoy

- 3. Automatic alarm for unexpected surfacing of the mother buoy (with its real-time GPS location and survey data transmitted)
- 4. Underwater working life: 12 months

Underwater Information Perception System of Mobile Type

An underwater information perception system of mobile type gathers information about underwater target or environment by cruising in the sea.

Take an underwater glider, C-Glider, as an example. C-Glider, as shown in Fig. 3, is an

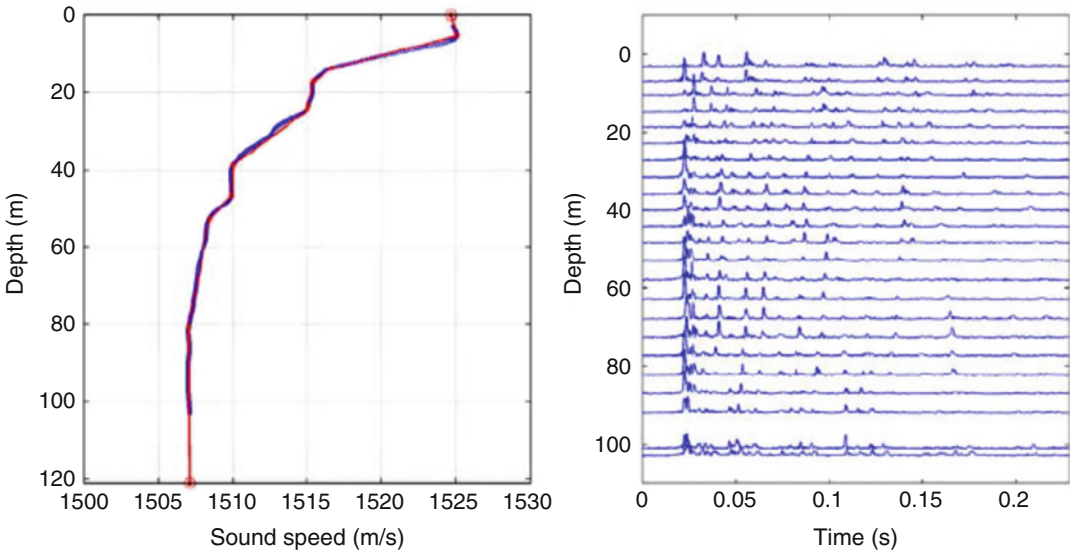
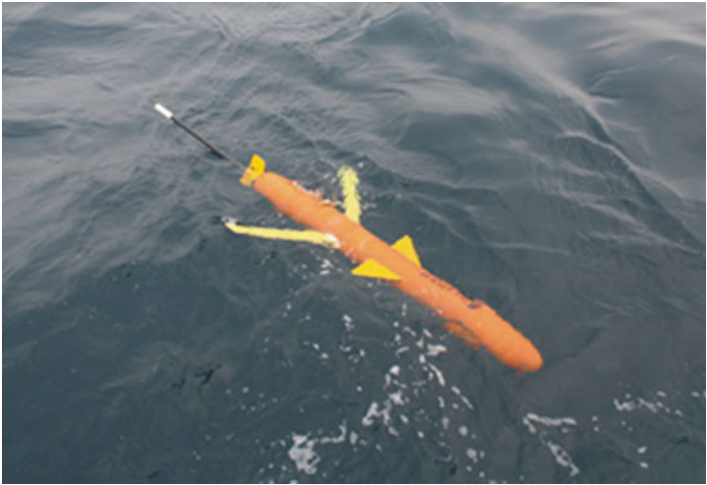
innovative platform used to monitor marine environment in middle-/long-distance waters. It features autonomy, real-time transmission, and mobility. It can be equipped with such sensors as CTDs, ADCPs, hydrophones, vector acoustic sensors, etc.

The main performance index is as follows:

- 1. Mass weight:<90 kg; total length:<2.2 m (without the antenna), housing diameter:<0.24 m
- 2. Measurable parameters: temperature, salinity, pressure, and current

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Fig. 3 C-Glider made by Yichang Research Institute of Testing Technology



Underwater Information Sensing Technology, Fig. 4 SVP and pulse signal collected with a glider (Jiang and Osler 2013)

- 3. Survey depth:1200 m
- 4. Range: ≥ 1500 km @ 0.5kn; ≥ 500 km @ 1.2kn
- 5. Maximum horizontal speed:1.5kn
- 6. Communication and location: BeiDou satellite and radio, or iridium and radio

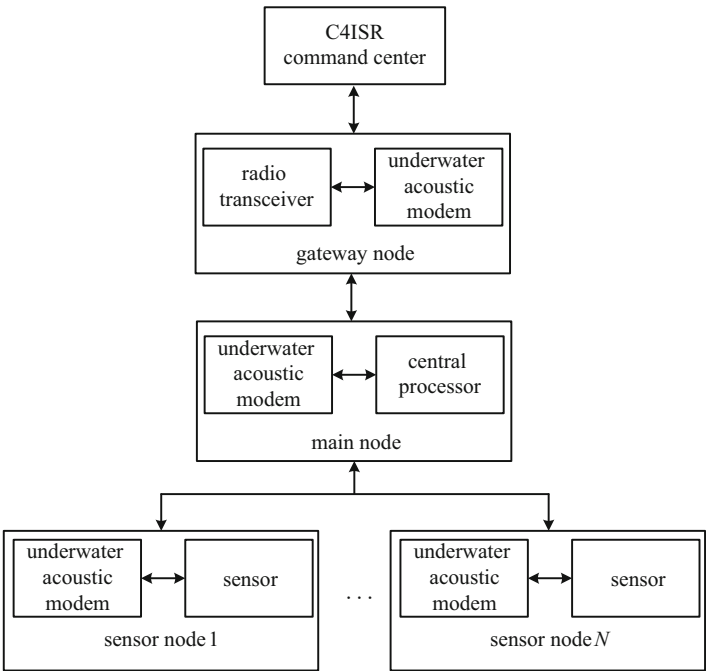
Figure 4 is the sound velocity profile and acoustic pulse signals collected with a glider (Jiang and Osler 2013).

Distributed Underwater Information Perception Network

With the rapid development of underwater sensor, autonomous processing, unmanned vehicle, communication, and network techniques, forming an area underwater information perception network with autonomous unmanned sensor nodes is the future developing trend of underwater information perception. The advantage of the unmanned sensor network is low cost, unmanned

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Fig. 5 Composition of distributed underwater information perception network (Ling 2013)



operation, and wide-range and long-time information collection.

A distributed underwater information perception network consists of sensor nodes, main node, and gateway node (Ling 2013), as shown in Fig. 5.

Sensor Node

The sensor nodes are used for gathering ocean environment information, processing array signals (feature analysis, extraction, detection, localization, and classification), and transmit processing results to the main node. The sensor nodes mainly consist of sensors and underwater acoustic modems. Different nodes use different sensors, including stationary arrays (e.g., shore-based sonars), retrievable arrays (e.g., vertical arrays released from moored buoys), ocean hydrologic sensors, etc.

Main Node

The main node is the core of a network. It is used to receive information transmitted from all the sensor nodes and fuse them. The other function of the main node is to manage the network, and each main node can manage tens of sensor nodes. The main node transmits the fused information to

the gateway node and receives the instructions from the command center through the gateway node.

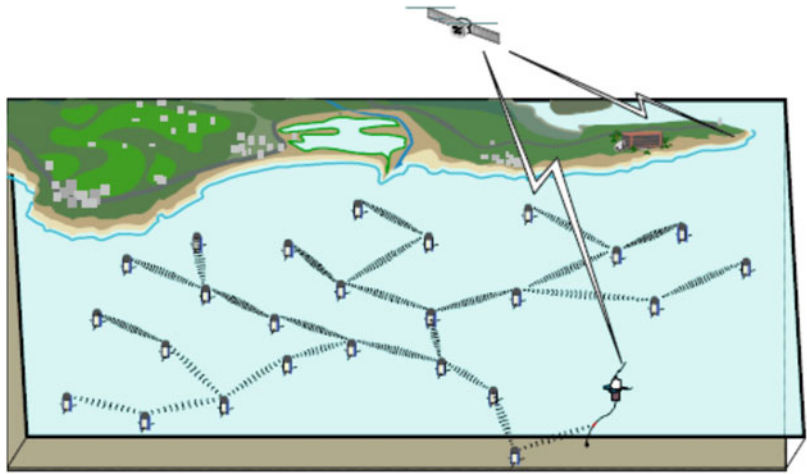
Gateway Node

The gateway node is used to communicate with the command center to transmit and receive information. Usually the gateway node is appointed to the underwater acoustic/radio buoy, tethered communication buoys, AUVs, or underwater gliders. The command center receives the information from the gateway node, and combines with information from other ISR (intelligence, surveillance, reconnaissance) systems, to obtain a fairly complete picture of the ocean environment and the target situation.

In the view of logical function, the underwater acoustic network mainly consists of the physical layer, data link layer, and network layer. The physical layer mainly resolves the problem of effective data transmission, by utilizing acoustic propagation characteristics and selecting proper modulation mode. The data link layer mainly resolve the problem of exploiting the acoustic channel in multi-user mode, including the MAC (media access control) and error correction

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Fig. 6 Sketch map of Seaweb (Rice et al. 2000)



control. The network layer mainly resolves the routing problem, studying the propagation route from source to receiver, which depends on the topology structure of the network (Li et al. 2006).

As an effective approach to collecting information, the distributed underwater information perception network undertakes the tasks of intelligence, surveillance, and reconnaissance, i.e., detecting, locating, classifying, and tracking the target (Shen 2008).

A representative distributed underwater information perception network is the Seaweb, as shown in Fig. 6 (Rice et al. 2000). Off-board Seaweb nodes of various types may be readily deployed from high-value platforms including submarines, ships and aircraft, or AUVs and unmanned aerial vehicles (UAVs). The architectural flexibility afforded by Seaweb wireless connections permits the mission planner to allocate an arbitrary mix of node types with a node density and area coverage appropriate for given telesonar propagation conditions and for missions at hand. Seaweb underwater acoustic networking provides digital wireless links enabling deployable autonomous distributed systems (DADS). Gateways to manned control centers include radio links to space or shore and telesonar links to ships.

Key Techniques of Underwater Information Perception

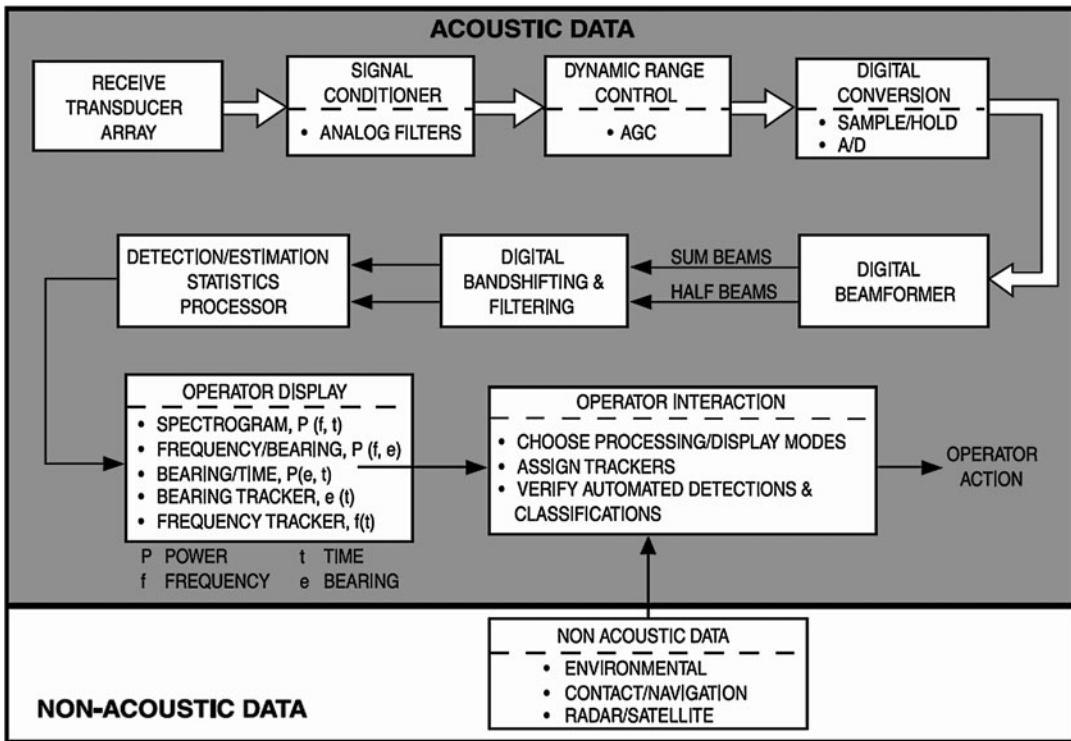
The underwater information perception system collects data of the ocean environment and the

targets, performs characteristics analysis, signal detection, and classification, and extracts information from the complex sea ambient noise. The main key techniques are discussed below.

Signal Detection

The conventional method of signal detection includes the following procedures: Receiving signals with a sensor array, filtering and amplifying the signals, controlling signal dynamic range, digital beamforming, broadband analysis, and shallow-band analysis, detecting signals and estimating the target parameter. A representative passive sonar detection system is shown in Fig. 7 (Carter et al. 2001).

However, when it comes to unmanned autonomous detection for the sensor node, there are several restrictions: Firstly, the apertures of both stationary nodes and mobile nodes are relatively small, usually a few meters, which is very difficult for arranging large-scale array to improve the spatial gain. Second, the sea ambient background is very complex. Except for the sea ambient noise covering the weak signal, there are a large number of coherent interferences at off-shore areas. If the detection system is unable to distinguish multiple sources, it is very difficult to perceive weak signal in those complex interference background. Especially for the autonomous detection system, it is very difficult to adapt to all kinds of complex background to make a correct decision automatically. Third, the detection is limited by energy



Underwater Information Sensing Technology, Fig. 7 Generic passive sonar system for signal detection, estimation, and classification (Carter et al. 2001)

which is supplied by the battery. It is necessary to lower the system complexity and power consumption.

To resolve the problem of underwater autonomous detection in complex sea background, the array self-unfolding technology needs a breakthrough to improve the spatial gain, and the artificial intelligence technique is needed to improve the environment adaptive ability.

Target Classification

Target classification is the most challenging aspect of anti-submarine warfare with a passive sonar, especially in the shallow sea with complex topology. An acoustic intelligence operator needs several years of training to be qualified for target classification. The computer algorithm to complete this function automatically is not very effective yet. It extracts the time domain features, continuous and line spectrum, modulation spectrum, cepstrum, spatial distribution

characteristics, etc., and input them into a neural network for training and testing after purification and compression. The distributed system can provide a variety of views and enhanced features, and may be helpful, though there are still many problems left to be resolved.

Underwater Communication

In the underwater information perception network, acoustic communication is usually used to transmit information among the distributed sensor nodes, while the gateway node connecting the network to the higher level network usually uses the acoustic/radio buoy. The performance index such as data rate and error rate are seriously limited by the underwater acoustic channel. The most impressing characteristics are limited available bandwidth, long-spread multipath, fast channel fluctuation, and strong Doppler effect, which cause low communication rate, high error rate, high probability of interruption, and long

propagation delay. In some adverse hydrological conditions, the communication is unavailable since the receiving node may be in the shadow zone of the transmitting node. Furthermore, it is hard to be covert when communicating by underwater acoustic modem. The probability of being intercepted may be reduced by using directive beam and broadband coding, but it is still possible to be intercepted by energy detection. Cables are another option for communication, but their deployment remains a problem to be solved.

Underwater Network

With all the nodes interconnecting with each other, the underwater information perception system will expand the range of data transmission and surveillance area, and achieve information exchange to improve the perception ability, by connecting all the nodes to make a network. In the underwater network, MAC protocol mainly resolves the problem of sharing the channel within multiple users. Due to long propagation time delay and low data rate in underwater acoustic channel, there are serious conflictions in multiple user conditions. The TDMA master-slave mode MAC protocol to allocate the time slot in turn is usually adopted in small-scale underwater acoustic networks. Every sensor node may need to wait for a few minutes for their turn to send a message, with very low efficiency. The mixed protocol combining different multiple access modes and access protocols is the hot spot in acoustic MAC protocol study at present (Xu et al. 2016).

Poor acoustic propagation conditions also bring great difficulties to route selection. Due to the moving of some mobile nodes, and the severity of marine environment, underwater acoustic communication is prone to breakdown or temporary failure, causing the underwater network topology being highly dynamic, leading to a variable route, which needs regular maintenance. The underwater data link is unstable with high packet loss rate, making low efficiency of route construction and packet transmission, which causes retransmission repeatedly. The above two characters result in high power consumption in route maintenance.

Distributed Detection with Network

Looking from the outside, the conception of distributed detection has many advantages, and a variety of detection opportunities brought by the wide spread sensors is attractive. However, passive detection is basically restricted by signal level, noise level, and available array gain. So, comparing the large-scale and high-gain sensor arrays, how do many distributed sensors with small gain work properly to observe and detect?

On the other hand, it should be noted that it is the comprehensive detection ability of the distributed network that should be paid more attention to, rather than the detection ability of a single sensor node.

In the distributed network, AUVs or gliders may come close to a target to enhance the signal, exploiting weak signal characteristics for target classification. Since there is obvious uncertainty with a single sensor's detection performance, some form of measurement is required to optimize the sensor configuration, making the distributed network being exploited and controlled efficiently.

Cross-References

- [Underwater Acoustic Sensor Network](#)
- [Underwater Information Sensing](#)
- [Underwater Surveillance](#)

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Underwater Lander

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Synonyms

[Diffusive equilibration in thin films \(DET\)](#); [Diffusive gradient in thin films \(DGT\)](#); [Intensified Silicon Intensifier Target \(ISIT\)](#); [Seabed](#); [Seafloor](#)

Definition

Lander is a general term for an autonomous and unmanned oceanography research vehicle, which descends by free-fall to the ocean floor (without remote assistance) and operates independently on the ocean floor. The platforms have the purpose of collecting physical, biological, and chemical variables over a predefined period of time. They operate independently of the surface for periods of a few days, for example, biological studies, up to a few years, for example, physical oceanography studies. After completing its mission, the lander releases the ballast weight by an acoustic signal transmitted from the surface or by a preprogrammed device. The vehicle comes back to the surface by merit of its positive buoyancy (Parker et al. 2003; Tengberg et al. 1995).

Scientific Fundamentals

The seafloor plays an important role in the regulation of the chemical composition of water masses in the oceans. Processes in sediments are important links or sinks in the biogeochemical cycles of elements in aquatic systems. Besides the seabed is the habitat for a great variety of higher organisms, and as such constitutes a distinct stratum for benthic life and consequently for numerous biological processes. The conventional approach to study these geochemical and biological processes is to collect a sediment sample from the seabed, bring it up to the surface, and scientists make observations and carry out experiments on it either on-board ship or in the laboratory. Methods that have been used to evaluate biogeochemical activity in sediments and at the sediment-water interface include direct measurements of benthic fluxes during incubations of sediment together with overlying water and measurements of solute distributions in and above sediments (Tengberg et al. 1995).

Accurate data are difficult if not impossible to obtain from the deep sea because artifacts are induced when the samples are subjected to large changes in hydrostatic pressure and (often) temperature as they are brought up to the surface. It is therefore preferable to carry out experiments and measurements directly at the seafloor (in situ). Even when working at quite shallow depths in shelf seas there is general agreement that it is preferable, if possible, to carry out experiments in situ, to avoid artifacts caused by excessive disturbance. Such studies can be carried out using benthic landers or other instruments deployed on the seafloor. When designing and using these instruments an important goal is to minimize the disturbance they cause to the seabed both during landing and operation. An additional benefit of using landers for deep seafloor studies is that, when they work properly, they are less demanding of ship time (and thus of money) than conventional sediment coring techniques. Once the lander has been deployed the ship is free to undertake other operations after until it is time for recovery (Tengberg et al. 1995).

When long-term operations are desired, lander platforms are the most economically suitable option. These stable platforms are autonomous and unmanned, oceanography research instruments to be deployed independently on the seabed over extended periods (Ferreira et al. 2014). Ocean Bottom Seismometer is a good example of these platforms, and it is a receiver for seismic signals that can be heard near the seabed. The major challenge when developing this type of instrument is to minimize the interference of the currents occurring at these depths, with the focus to receive a clean signal by the lander (Tengberg et al. 1995). Exploration of the ocean floor can bring both great business opportunities and new scientific discoveries. Therefore, it is of great interest to be able to obtain information about it at a low cost.

When designing a lander there are numerous points that need to be considered if it is to be successful in achieving its designated tasks at the seafloor. These include the methods of launching and recovery, the choice of construction materials for various components, the design, descent, and ascent speeds, the landing technique, selection of the techniques for sampling, observation, and measurement, and the choice of electronics and energy requirements.

Historical Development

The first in situ experiments on the seabed were carried out by scuba divers. Fluxes into or out of sediments have been obtained for a number of substances, by deploying a chamber on the seafloor isolating an area of sediment surface and the overlying water and then drawing discrete samples at specific times intervals for analysis in the laboratory.

Since the number of scientific manned submersibles in the world has always been, and is likely to remain limited, for most experiments, the use of submersibles for such deployments is impractical nor is it generally affordable. So, the development of autonomous free-vehicle landers is one obvious solution to solving these practical difficulties. In 1979, the HINGA lander, as a

representative of the first generation of landers, was designed to be deployed at the bottom of a mooring (Hinga et al. 1979).

Based on the combined experience gained with the early devices, the second-generation landers were developed during the 1980s, for example, the DEVol lander and the IHF lander. The USC lander was developed as a result of the need to have a device capable of making multiple flux incubations with a short turn-round, and of being deployed in a wide range of environments. The development of the BECI lander was based on the philosophy of keeping the concept as simple as possible, in reaction to the adverse experiences gained during the development of the highly complex MANOP lander (Tengberg et al. 1995).

During the 1990s, there was the third generation of landers which involved various new developments including FVGR-2 lander (Smith Jr 1987) and the GOMEX lander (Rowe et al. 1994) which was designed not only to estimate fluxes but also to attract and capture benthic organisms.

After entering the twenty-first century, the design and research of landers has entered a new era. Based on the original functions, the lander has been given more automation capabilities, such as fixed depth operation, active adjustment, automatic positioning, and rapid recovery.

Material for Lander Frames

Frame Material

An autonomous benthic lander must have a basic support frame on which is mounted the instruments, the ballast weights, the release mechanism, and the buoyancy. The main objective when constructing this frame is to keep its weight as low as possible without compromising its mechanical strength. Keeping the weight low not only makes the whole instrument easier to handle but also minimizes the need for expensive buoyancy material. While deployed the lander and its frame are exposed to relatively trivial static stresses, so the main risks of mechanical damage occur during launch and recovery.

For ease of transport and storage, particularly in containers, most constructors have designed outer frames that are readily dismantled. The material most commonly used is aluminum, which has the advantages of being relatively inexpensive and light. However, aluminum is not as strong as some other materials, for example, stainless steel, and many ships do not carry the special welding equipment required for repairs. Alternative materials for frames are stainless steel, titanium, and composite materials such as glass fiber reinforced with aramid fiber and with epoxy resin as binding material.

Regardless of metals used in constructing the frame, considerable care has to be taken if any combination of different metals is used (e.g., stainless-steel bolts in an aluminum frame, titanium pressure house in a stainless steel frame). Failure to insulate the metals from one another, by coating or painting, will result in the rapid corrosion of the metal with the lower reduction potential (Tengberg et al. 1995).

Ballast Weights

A wide range of materials is used for the ballast weights including concrete, scrap iron, cast iron, and lead. Lead has the highest density, but should be avoided because of its potential environmental impact may be serious. Scrap iron (e.g., railway waste) is cheap and easy to obtain but is awkward to use because of its irregular shape. Iron blocks cast into conical shapes are expensive but serve to minimize disturbance upon landing. Concrete ballast is best avoided, because of its density is relatively low, and its behavior in the deep-sea environment is poorly known.

Ballast weights can be fixed underneath the frame and used as footpads. Since most of the sediment suction is applied to the footpads, dropping them off allows the rest of the lander frame to escape easily to begin the ascent back to the surface. While this method of attaching ballast weights decreases the risk of a lander becoming stuck in the sediment, it is not a complete guarantee of success. One minor inconvenience of this method is that it complicates the deck handling, since either a special rack is needed or the lander

has to be rifted off the deck to attach the weights (Tengberg et al. 1995).

Descent and Landing

Descent

Currents within the oceanic water column will affect a lander during its descent and ascent and may carry it away from the desired position (Hendricks and Rodenbusch 1981). The faster a lander sinks or rises, the less it will drift out of position, rates of 0.5–1 m/s are acceptable, but this still means that a lander will take 1–2 h to traverse a water column 3,600 m deep. Another reason to maximize the descent and ascent rates is to save ship time, by minimizing the waiting time after launching and weight release, and if a lander takes too long a time to ascend, there may be problems relocating it at the surface. However, if the lander's descent rate is too fast, it is liable to cause excessive disturbance to the sediment-water interface at the measurement site. Also, it will be driven deeper into the sediment and so run the risk of the lander becoming irretrievably stuck in the bottom. Some landers have become trapped temporarily in sticky bottom sediments. Pull-out forces can be increased by adding more buoyancy, but this incurs penalties of increased cost and the need to use more ballast to attain the same descent speed (Tengberg et al. 1995).

Landing

As long as the footpads are large enough to prevent too deep a penetration of the frame in the sediment, most landers work successfully by simply crash-landing on to the bottom at a modest speed. Holes in the footpads can help to reduce the suction on lift-off. Some landing techniques have been devised to minimize the impact both to reduce disturbance of the experimental site and the risks of the lander becoming irretrievably stuck. The landing technique consists of the descent weights being suspended on ropes 0.6 m under the lander frame. When these weights hit the bottom, the lander is positively buoyant, stops about 0.1 m from the bottom (filmed during scuba diving), and then stays suspended above the

bottom. No feet are required on these landers because the frame never touches the bottom. This technique eliminates the risk of the frame getting stuck in the sediment (Tengberg et al. 1995).

Main Equipment

Microprocessors and Lander Electronics

Chamber stirring, water sampling, and measurements must all be controlled, monitored, and recorded electronically. The choice of controller varies greatly from the lander to lander depending on the design requirements. While designers may still opt to develop their systems depending on personal knowledge and experience in electronics, a wide range of microprocessors are now commercially available which are suitable for lander applications (small, good capacity, easy to program, and low energy consumption). It is essential to protect the electronics against seawater and so they have to be housed in pressure-cases, usually of aluminum, stainless steel, or titanium.

Other Equipment

Some landers have video or still cameras mounted to survey the landing and the bottom deployment. The lander has a camera inside the chamber to take photographs of animal movements on the sediment surface. Their use tends to be of greatest interest, during the initial stages of deployment when the behavior of the lander in water and on its arrival at the seafloor is poorly known.

Some chamber landers are equipped with communication systems that relay back to the ship the successful operation of all stages of the lander's performance, its landing, chamber penetration, lid closure, and lift-off. The simplest system is to release floats at particular phases of the operation to signal their successful completion but over deep water, the floats may not be spotted. Acoustic communication systems are expensive and maybe superfluous if the lander is to be deployed only for a few days. They could be of importance for status checks during long-term deployments (Glud et al. 1995).

Electrical Energy

For many of its functions, a lander requires energy. The longer the deployment, the greater the energy storage capacity is needed, which for chamber landers mostly used for stirring, so designs and components need to minimize energy consumption.

The energy can be supplied by different types of batteries. Lithium type batteries combine high capacity with small size. The nickel-cadmium batteries are generally rechargeable and can give high currents, but they tend to discharge at zero current and so they are unsuitable for long-term deployments. The batteries most widely used on landers are oil pressure compensated lead-acid batteries.

When calculating requirements for energy capacity, a very important aspect to take into consideration is that at cold temperatures most batteries lose up to half of their nominal capacity. Some constructors recommend splitting up the power supplies into several independent units and keeping high current and stable voltage supplies separated. Security systems and computer memories for data storage and programming normally have their individual energy supplies (Glud et al. 1995).

Ascent and Recovery

The need for an excess of float capacity after ballast release is an important consideration. If too little buoyancy is used, the lander may fail to pull out of the bottom so that it is either lost or has to be recovered by dredging.

Most landers have two independent systems for ballast release. Acoustically controlled releases are handy but expensive; their advantage is that using a system controlled from the surface, the time of recovery can be chosen to provide the most appropriate conditions (weather and time). Most acoustic releases produce precision time "pings" which can be used to estimate the distance to the lander and the direction the ship needs to be tuned to lessen the distance.

The greatest risk of damage to the lander (and human handlers) occurs during launching and recovery. It is very important to prevent the lander from swinging and bouncing while hanging from

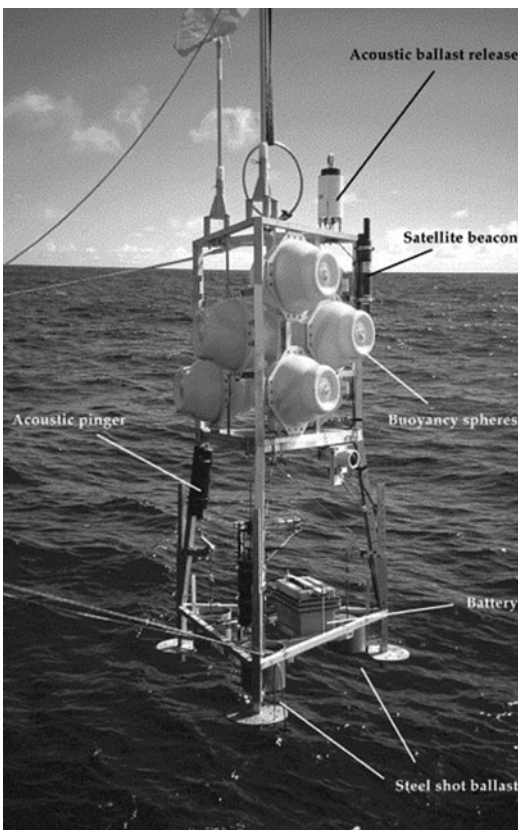
a crane or A-frame. A successful launch and recovery avoiding damage to the lander is more a question of the skill of the ship crew and the sea state than of lander design. Some landers have been lost or damaged during recovery because of a lack of communication between people working on deck and the bridge (Glud et al. 1995).

Key Applications

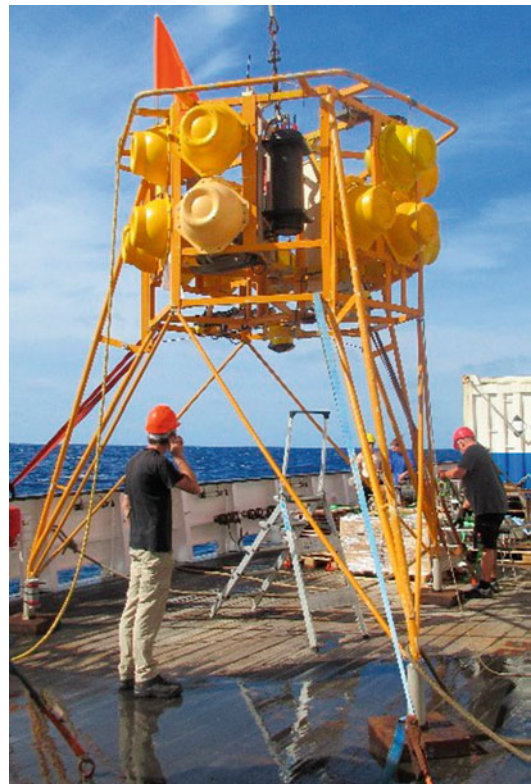
The BENBO benthic lander, which was designed and constructed in collaboration with a commercial engineering company (Denmark), is shown in Fig. 1. The lander incorporates a number of interchangeable modules. These include (i) an enclosed chamber for determination of total

benthic community respiration and solute flux, (ii) a micro-profiling module which can insert up to four oxygen micro-electrodes and two pH micro-electrodes into the sediment to measure pore water concentrations at very high spatial resolution ($\sim 50 \mu\text{m}$ intervals), and (iii) a unit to insert DGT (diffusive gradient in thin films) and DET (diffusive equilibration in thin films) gel probes through the sediment-water interface to measure fluxes of trace metals and concentrations of major ions at a spatial resolution of $100 \mu\text{m}$ to 1.5 mm (Black et al. 2001).

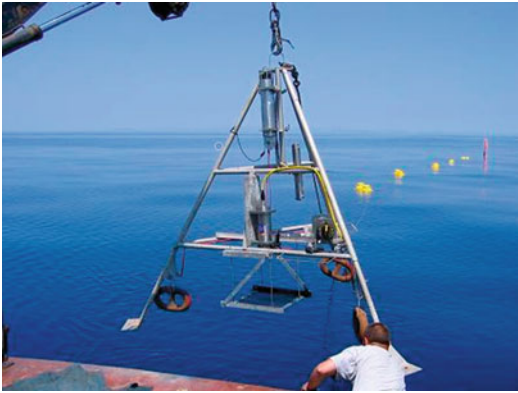
Netherlands Institute of Sea Research developed the Bottom Boundary Observatory (Fig. 2), a four-meter tall lander for deep sea missions. Its structure was designed to remain on the seabed for long periods, sometimes more than 1 year. This lander is adaptable to several configurations; it can measure near-bottom salinity, currents speeds,



Underwater Lander, Fig. 1 Photograph of the BENBO benthic lander with the oxygen micro-profiling module during deployment (Black et al. 2001)



Underwater Lander, Fig. 2 Bottom Boundary Observatory (Gonçalves 2019)



Underwater Lander, Fig. 3 ISIT lander (Heger et al. 2007)



Underwater Lander, Fig. 4 Tsunami Warning and Early Response system of Cyprus (Gonçalves 2019)

currents directions, temperature; and it is equipped with cameras (Person et al. 2006).

Heger et al. (2007) developed a lander that can be equipped with different types of sensors, capturing images of deep-sea creatures or analyzing water quality. It is named Intensified Silicon Intensifier Target (ISIT) lander, as shown in Fig. 3, in which the purpose is to research stimulated and spontaneous bioluminescence emissions in the water column and benthic boundary layer to depths around 1000 m, as well as fauna attracted to artificial food falls.

In Fig. 4, one can see a flat shape system, called the Tsunami Warning and Early Response system of Cyprus, which is one of the offshore communications backbones. This lander belongs to the OceanWorks International's seafloor network systems (Gonçalves 2019).

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Underwater Mining System

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Synonyms

CBL – Continuous Line Bucket; PTM – Passive Towed Mining

Introduction

The underwater mining subsystem is mainly used to collect ore from the seabed by underwater mining vehicles or other equipment. Since the exploration of the polymetallic nodule buried in the deep seabed was first started in the 1960s, it has now been extended to the other mineral resources such as polymetallic sulfide and cobalt-rich crusts. Correspondingly, the underwater mining technology and equipment have also been developed from a simple bucket mining subsystem to towed mining subsystem, then to shuttle subsystem and to current self-propelled mining subsystem to achieve accurate mining.

Bucket Mining Subsystem and Towed Mining Subsystem

In 1960, the technical scheme of towing cable with a bucket to obtain the nodule was put forward. Deep-sea mining test was completed to collect the nodule with the bucket mining subsystem, as shown in Fig. 1. It is difficult to obtain better economic benefits because the bucket cannot be controlled on the seafloor and a lot of resources are lost.

In 1967, continuous line bucket (CLB) mining technology scheme was put forward. The CLB system refers to all mining systems, which have been inspired by bucket chain dredging or by oceanographic chain dredges, that is, buckets have been fixed to a continuous line in the manner

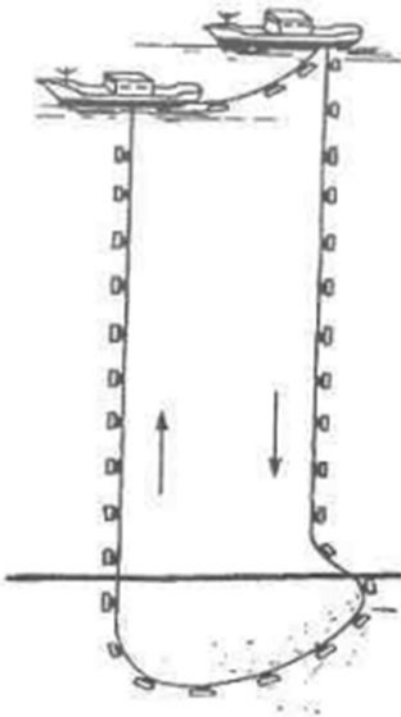


Underwater Mining System, Fig. 1 Bucket to obtain the nodules in sea test (ISA 2008)

that a defined number of buckets are towed across the seafloor thus collecting loose materials at the top of the seabed (Fig. 2). In 1970, the CLB system was tested and achieved the expected results. The CLB system is simple in structure, and low in investment, and operation cost. However, the bucket and rope are easy to entangle, and the acquisition track on the seafloor is almost uncontrollable. It is difficult to avoid the cliff and obstacles on the seafloor, resulting in a low acquisition rate, large loss of resources and a great impact on the marine environment. The CLB system is proven to be unable to meet the requirements of underwater mining and has been abandoned by researchers.

Passive Towed Mining Subsystem and Towed Vehicle

The passive towed mining subsystem is composed of a towed collector, a lift pipe system, an airlift system, a pump lift system, an onboard



Underwater Mining System, Fig. 2 The continuous line bucket system (ISA 2008)

handling system, instrumentation and control system, and a test mining ship, as shown in Fig. 3a.

The collector, as shown in Fig. 3b, will be towed by a flexible hose made of hard rubber, collecting nodules with waterjet flow based on Coanda effect separating seafloor sediment, crushing nodules to desirable size distribution, and feeding them into the lift pipe.

During the sea trial in the 1970s, Ocean Management Incorporated corporation (OMI) and OMA both adopted the towed vehicle, as shown in Fig. 3. In 1997, an improved test was carried out on a sea mountain at a depth of 1700 m.

The test showed that the towed mining vehicle has the advantages of simple structure, light disturbance, and damage to the seabed, but it cannot accurately walk according to the planned mining route, low collection efficiency, large mining loss, and more difficult to avoid obstacles. Therefore, it is almost impossible to be used in industrial trial

production. OMI thinks it is necessary to turn to the research of remote-controlled self-propelled mining vehicles.

Shuttle Subsystem and Mining Vehicle

In 1972, French Academy of Ocean Development (IFREMER) began to study a technical scheme of shuttle subsystem integrating mining and upgrading.

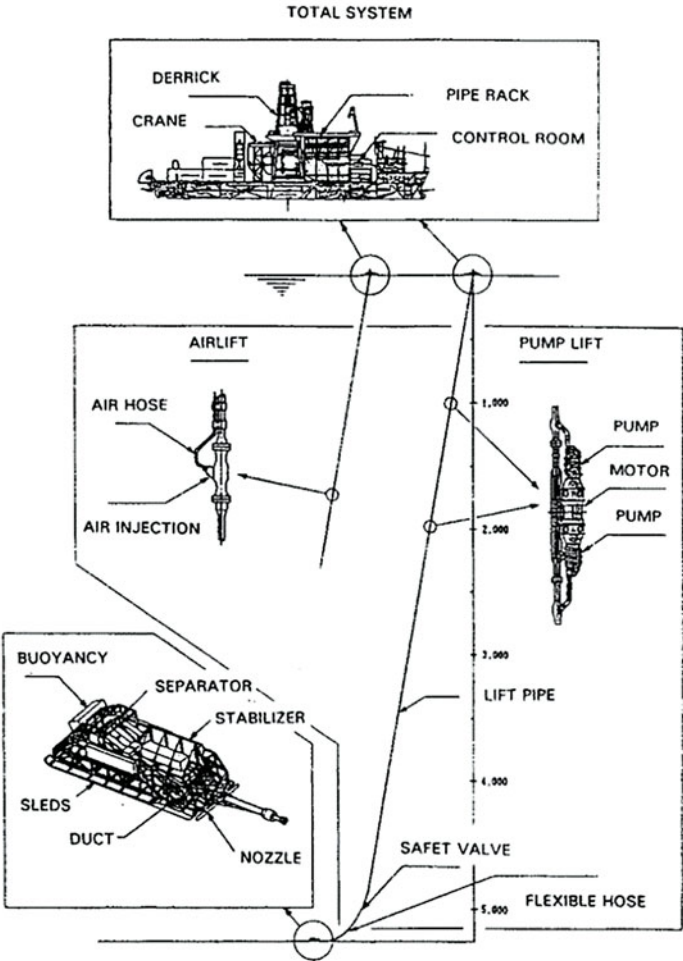
The technical scheme of the shuttle subsystem is composed of a semi-submersible platform and several shuttle mining vehicles. Each mining vehicle is controlled by the platform through the acoustic system. The mining vehicle with zero self-weight in the water using buoyancy material is driven by Archimedes propeller to collect nodule at the sea bottom, and the ballast with equal weight to the collected nodules is discarded. Before all ballast is discarded, the collection is stopped, and the rest ballast is discarded until the weight of the mining vehicle in the water is zero. Under the action of the propeller, the vehicle rises to the semi-submersible platform to remove the nodules, replenish the ballast, and start a new round of mining.

As shown in Fig. 4, the vehicle PLA 1 is mainly composed of propeller mechanisms, thruster, storage bin, collection mechanism, buoyancy, battery power and navigation, positioning, telemetry, and remote control unit. The propeller mechanisms are four independently driven Archimedes screw propeller mechanisms. The four thrusters are to provide lifting and lowering power, control the height and trajectory, realize fluctuation, swing, and rotation. The storage bin was used for placing ballast and nodule or ballast, the buoyancy was used for balancing the self-weight in the water to be zero. The battery power was installed on both sides of the mining vehicle.

Only a small proportion of the model machine of the mining vehicle was tested in the laboratory. In December 1985, the PLA 2-6000 mining vehicle was upgraded and developed. In October 1987, a series of deep-water diving was carried out in the Mediterranean Sea, demonstrating its

Underwater Mining System, Fig. 3 Japanese passive towed mining subsystem and towed vehicle (Yamazaki and Yoshida 1999). (a) Japanese mining system for at-sea test. (b) The towed vehicle tested by OMI in the 1970s

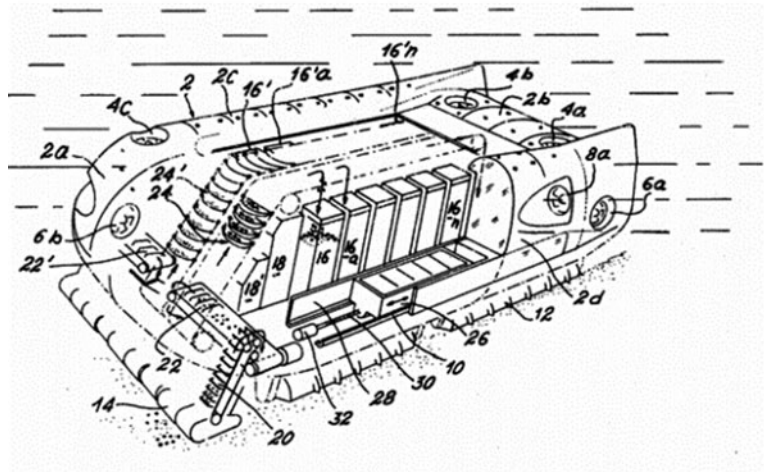
a



b



Underwater Mining System, Fig. 4 The PLA I (Lemerrier et al. 1982)



Underwater Mining System, Fig. 5 The PLA II (Herrouin et al. 1989)

dynamic flight, landing, and movement capabilities on the seabed. See Fig. 5 (Charles et al. 1990).

The vehicle has the advantages of flexibility, good adaptability to seabed topography, small mining loss, and little impact on the production capacity of the system with the failure of one vehicle. However, each vehicle can only mine 250 tons of nodules in 8 hours, and a large amount of ballast waste rock needs to be transported from the land. Furthermore, a single vehicle may be lost. It must have a method to recover the failed single vehicle. The battery weight is too heavy. The manufacturing and operation costs are high. After the PLA 2-6000 was tested, the research of this system was abandoned.

Self-Propelled Mining Subsystem and Mining Vehicle

In the 1970s, a series of experiments to mine deep-sea polymetallic nodules were carried out mainly in the Pacific Ocean with a depth of 5000 meters. The testing results show that the self-propelled mining vehicle has outstanding advantages such as good mobility, easy to avoid obstacles, accurate acquisition path control, high recovery rate, and relatively constant output during variable speed driving. Subsequently, the mining vehicle of deep-sea mineral resources has gradually turned to the research of self-propelled mining vehicle, which is considered to be the most valuable mining vehicle for commercial mining.

The self-propelled mining vehicle is powered by a sea surface mining ship through armored cable. The operator on sea surface mining ship remotely controls the mining vehicle to operate on the seabed according to the set mining path. At present, there are mainly Archimedes spiral mining products and crawler mining vehicles verified by sea testing.

Self-Propelled Archimedes Mining Vehicle

At first, Archimedes spiral vehicle was developed by the US Navy in the swamp area. Ocean Minerals Company Corporation (OMCO) was applied to the deep-sea mining field. In 1979, OMCO led the mining test in the polymetallic nodule mining area in the Pacific Ocean.

The mining vehicle consists of a propulsion device, a mechanical collecting mechanism, a crushing mechanism, and a control specific embodiment. The propulsion apparatus adopts an Archimedes screw propulsion mechanism. The driving roller of the Archimedes screw propulsion mechanism has built-in buoyancy material to adjust the weight of the mining vehicle in the seawater. The nodule pick-up and handing mechanism of the miner vehicle include the following components: a rake type pick-up for lifting the nodules (and a selected amount of cushioning mud) from the ocean floor; a conveyor for transferring the picked-up nodules from the rake to a conveyor surface and for holding the nodules in a stabilized position on the conveyor surface while subsequently conveying the nodules first to a washer and then to a crusher; a washer for separating the nodules from the picked-up mud and for removing the mud from the conveyor surface; a stripper for removing the nodules from the conveyor surface at the inlet to the crusher; a crusher for crushing the nodules to a nodule slurry; and a pumping mechanism for pumping the nodule slurry from the miner vehicle to the buffer storage. A single, rake-type, nodule pick-up and associated nodule handling mechanism are used, and this pick-up and associated mechanism are located in the interior of the miner vehicle between the propulsion screws. The controls include electrical, electronic, and hydraulic controls. The operations that are actively controlled include initial placement of the miner vehicle in the mining area; the speed and direction of movement of the miner vehicle; the depth to which the rake of the pick-up penetrates the ocean bottom; the angle of inclination of the pick-up to the ocean bottom; the amount of soil or mud picked up with the nodules; the separation of the nodules from the soil or mud; the transport of the nodules to the crusher; the crushing of the nodules; the recovery of fines; the avoidance of obstacles; and the rejection of nodules or other objects above a certain size.

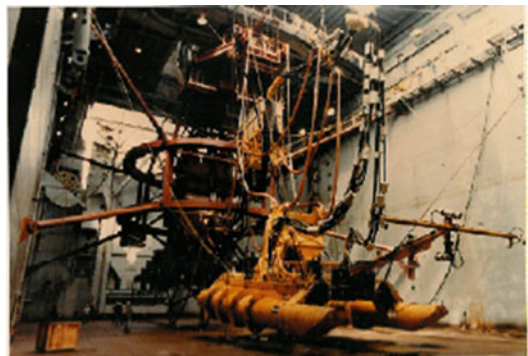
In 1979, OMCO carried out mining experiments in the polymetallic nodule mining area in the Pacific Ocean (Jin 1994).

Subsequently, scientists and technicians from France, Germany, Russia, and China conducted extensive research on the Archimedes screw propulsion mechanism. Through the comparative test with the crawler mechanism, it was found that Archimedes screw propulsion mechanism has high static indentation depth and walking power, low bearing capacity, difficulty in obstacle surmounting, and the spiral groove is easy to be covered by sediments, resulting in slipping. Although Archimedes screw propulsion mechanism has good performance on soft mud, swamp, and snow, it is almost impossible to walk on hard rock. Therefore, the Archimedes screw mechanism is still to be further studied and improved when it is actually used for driving on the thin soft seabed (Fig. 6).

Self-Propelled Crawler Mining Vehicle

As we all know, crawler vehicles have the following characteristics: large grounding area, stable movement, strong load capacity, strong adaptability of the road surface, and strong obstacle surmounting ability, etc. Scientists and technicians from Germany, France, Russia, South Korea, India, Japan, Belgium, China, and other countries have carried out relevant research on crawler mining vehicles for deep-sea mineral resources mining.

The crawler mining vehicle is powered by the sea surface mining ship through armored cable and operated remotely by the surface operators. The mining vehicle has two or more independent running tracks, each of which is driven by a



Underwater Mining System, Fig. 6 Self-propelled Archimedes mining vehicle (ISA 2008)

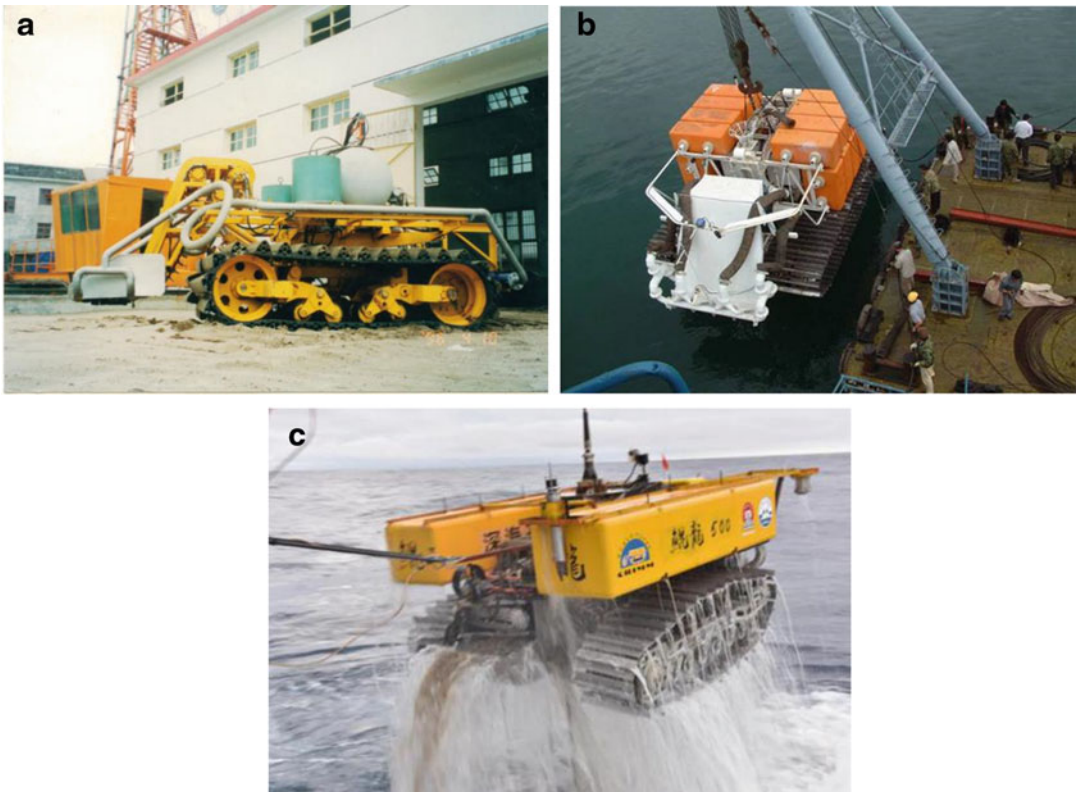
hydraulic motor, and travels on the seabed according to the set operation path. The collector picks up the ore such as polymetallic nodules, polymetallic sulfides, and cobalt-rich crusts covered on the seabed. The collected ore can be broken into 10–50 mm particles by the crusher on the mining vehicle according to the design needs, and then sent to the lifting pipe through a flexible hose. The mining vehicle is generally composed of a traveling mechanism, collector, crushing mechanism, hydraulic power system, landing balance device, buoyancy body, electrical and monitoring subsystem, navigation, positioning system, etc.

Self-Propelled Crawler Mining Vehicle for Seabed Polymetallic Nodules

In the seabed polymetallic nodule mining area, the water content of the surface sediments is too high.

The shear strength and bearing strength of surface sediments are exceedingly small, so it is difficult to support the operation of mining vehicles. It puts forward higher requirements for mining vehicles to collect polymetallic nodules from the soft surface sediment to minimize disturbance to the ocean floor to the greatest extent.

Researchers from China, India, South Korea, Belgium, and other countries have carried out the dynamic calculation and virtual prototype design of the mining vehicle, developed the principle prototype of the mining vehicle, and carried out the corresponding principle test. On this basis, a certain scale of commercial mining vehicles has been completed, and the mining vehicles have been carried out mining tests. Up to now, the deep-sea mining test of whole subsystems contained a self-propelled crawler mining vehicle subsystem, and other subsystems such as rise



Underwater Mining System, Fig. 7 China's Self-propelled crawler mining vehicle. (a) Hydro-mechanical combined mining model machine (Jian and Wang 1998).

(b) Prototype of the mining vehicle (Wang 2015). (c) 500-meter sea trial mining vehicle (COMRA 2018)

transportation subsystems have not been carried out.

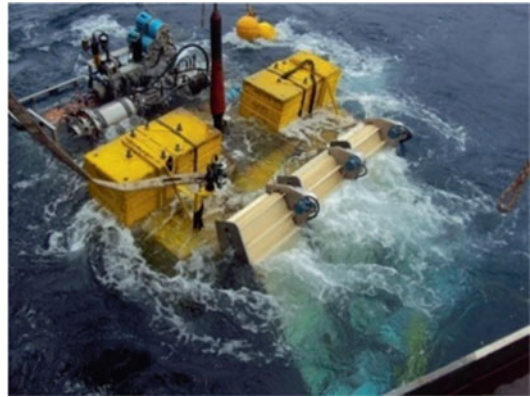
From 1991 to 1995, two types of hydraulic and hydro-mechanical combined mining model machines were developed in China. From 1996 to 2000, in-situ testing of the geo-mechanical properties of seabed sediments, traction test research of walking mechanism, and driving dynamics research of crawler vehicles was studied, and a prototype of the mining vehicle was developed. The mining test was completed at the depth of 135 m in Fuxian Lake located in Yunnan Province, China in September 2001. Subsequently, the improved design of the miner was completed in 2005. A mining vehicle test pool with a length of 60 m, a width of 50 m, and a depth of 5 m were established in 2006 and the improved mining vehicle was tested in the pool in 2007. Then, another mining vehicle was developed from 2010 to 2017 and was completed a 500-m sea test in the South China Sea in 2018.

The developed self-propelled crawler mining vehicle developed by China (Xie 1995; Jian and Wang 1998), India (Nidhi 2015; Masuda et al. 2014), South Korea (Lee et al. 2013; Sup et al. 2016) and Belgium (GSR 2019) are shown in Figs. 7, 8, 9, and 10.

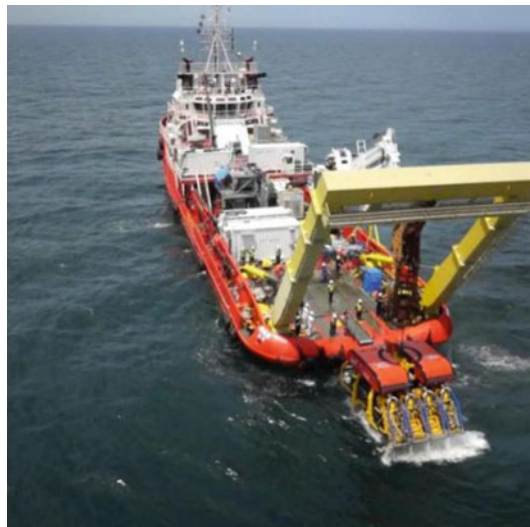
Self-Propelled Crawler Mining Vehicle for Seabed Cobalt-Rich Crust

As for the mining of seabed polymetallic sulfides, the design of mining vehicles for seabed cobalt-rich crusts can also draw lessons from the mining technology of open-pit mines on land.

With the support of the 12th China Ocean Project, a cobalt-rich crust mining test vehicle was developed for sampling by adopting open-pit continuous mining technology and simultaneously mining and collecting. As shown in Fig. 11, the vehicle mainly consists of a four-track walking mechanism that is independently suspended and driven, an acoustic thickness probe, spiral drum, slurry pump, and other auxiliary devices. When it works, the acoustic thickness probe obtains the thickness of the ore body on the line, the spiral drum is controlled to excavate and crush the crusts according to the obtained thickness of cobalt-rich crust, and the broken crust



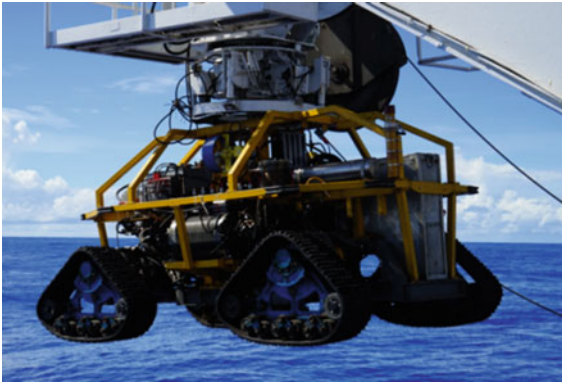
Underwater Mining System, Fig. 8 India's Self-propelled crawler mining vehicle (Nidhi 2015)



Underwater Mining System, Fig. 9 South Korea's Self-propelled crawler mining vehicle (Sup et al. 2016)



Underwater Mining System, Fig. 10 BGR's Self-propelled crawler mining vehicle (GSR 2019)



Underwater Mining System, Fig. 11 Four-track cobalt rich crust mining vehicle (CIMR 2018)

particles are collected into the sampler by the slurry pump. In 2018, the sampling test was completed at the depth of 2019 meter in the cobalt crust contract area of COMRA in the Pacific Ocean and a total of 150 kg of crust samples were collected in four tests (CIMR 2018).

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Underwater Navigation and Positioning

► Simultaneous Localization and Mapping

Underwater Production System

► Subsea Production System

Underwater Surveillance

► Underwater Information Sensing Technology

Underwater Target

► Photoelectric Detection Technology in Underwater Vehicles

Underwater Vehicle, Underwater Robot, UUV, ROV, HOV

► Obstacle Avoidance Technology for Underwater Vehicle

Underwater Welding

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Synonyms

Welding methodology

Definition

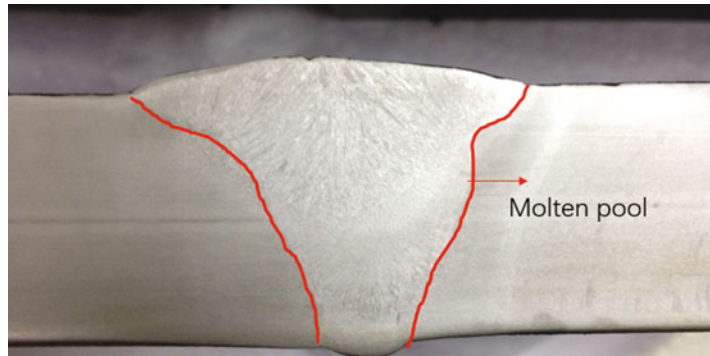
Underwater welding, just as the name implies, is a specific welding process that is performed while

the welder is submerged in the underwater environment, often at elevated barometric pressures.

Introduction

Welding technology is a powerful fabrication technology that joins materials by employing highly concentrated power energy to melt the components. This technology is widely employed in onshore and offshore structure constructions due to the high productivity during the assembly procedures (Smith and Smith 2009; Fu et al. 2014, 2016). Typically, two stages are included in the welding process, the heating and the cooling stages. Firstly, during the heating stage, the base material is melted, and the melted filler material is deposited simultaneously to the joint. In the heating stage, the melted base and deposited materials form a pool of molten material (molten pool), as shown in Fig. 1. Due to the reduction of power energy input in the cooling stage, the temperature of the molten pool decreased significantly. Finally, the melted base material and deposited filler is cooled to join the construction. Usually, the resistance of the welded configuration (butted joint, full penetration, fillet joint, etc.) is higher than the base material (parent metal).

In some specific working environment, the welding technique is also an efficient way to join the structures (Jeffus 1997). The underwater welding is a common technique to repair the underwater structures in the ships and offshore structure constructions industry, as shown in Fig. 2. Underwater welding, just as the name implies, is a specific welding process that is performed while the welder is submerged in the underwater environment, often at elevated barometric pressures. However, due to the limitation of the in-field working conditions, such as the existence of the electrically conductive medium (water) and the lack of the welding protection, the underwater welding is different from the open-air welding process, and some specific techniques are commonly employed to promote the underwater welding application. For instance, in the offshore platform modules assembly and structure repairing, shielded metal arc welding (SMAW) is

Underwater Welding,**Fig. 1** Molten pool region after the welding (Fu et al. 2014)**Underwater Welding,****Fig. 2** Underwater wet welding process

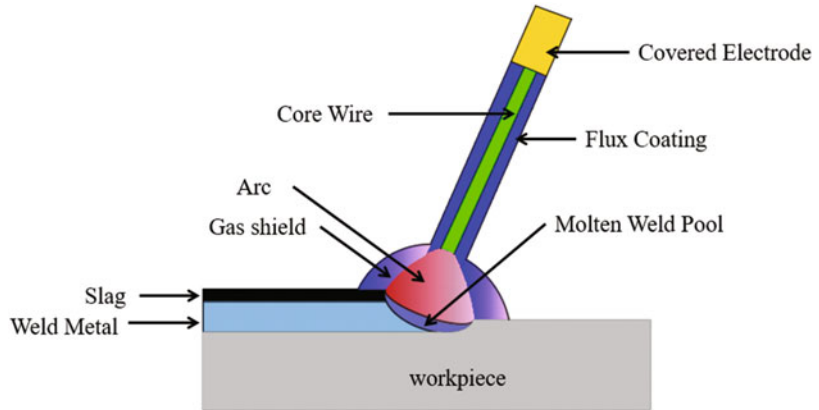
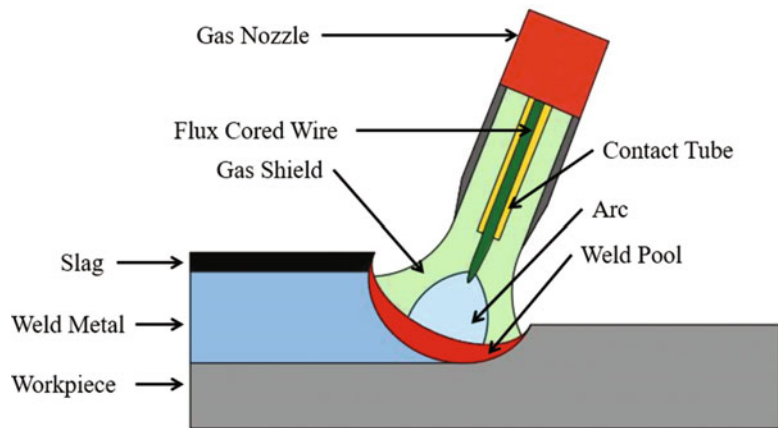
the popular fabrication process (Muransky et al. 2012; Jeffus 2002; Caprace et al. 2017). However, in controlled environments, specifically for the underwater condition, employing inert gases technique to shield the weld cannot be readily adopted in some situations, due to the unpredictable situation which can result in a poor weld.

Welding Methodology

With the development of the industry construction, a number of welding methods are developed, such as the manual welding methods including shielded metal arc welding (SMAW) and semiautomatic and automatic processes that are the most popular welding technique, such as gas metal arc welding (GMAW), submerged arc welding (SAW), gas tungsten arc welding (GTAW), and flux-cored arc welding (FCAW).

The SMAW, also known as manual metal arc (MMA) welding, or electric welding, is a manual welding technology that employs a welding electrode to weld metals together (Jeffus 1997). Different from the open-air welding, the direct current generated by the energy supply is employed to form the welding arc between the electrode and the workpiece metals in the underwater welding process. Because of the simplicity of application of the welding process, the SMAW is one of the most popular welding processes in the steel structural construction (Fig. 3).

The GMAW commonly termed by its welding technique subtypes, MIG (metal inert gas) welding and MAG (metal active gas) welding, is the welding technique in which a welding arc is generated between the consumable welding electrode and the base metals to be joined, which melts the welding wire and workpiece metal, joining them together (Muransky et al. 2012). With

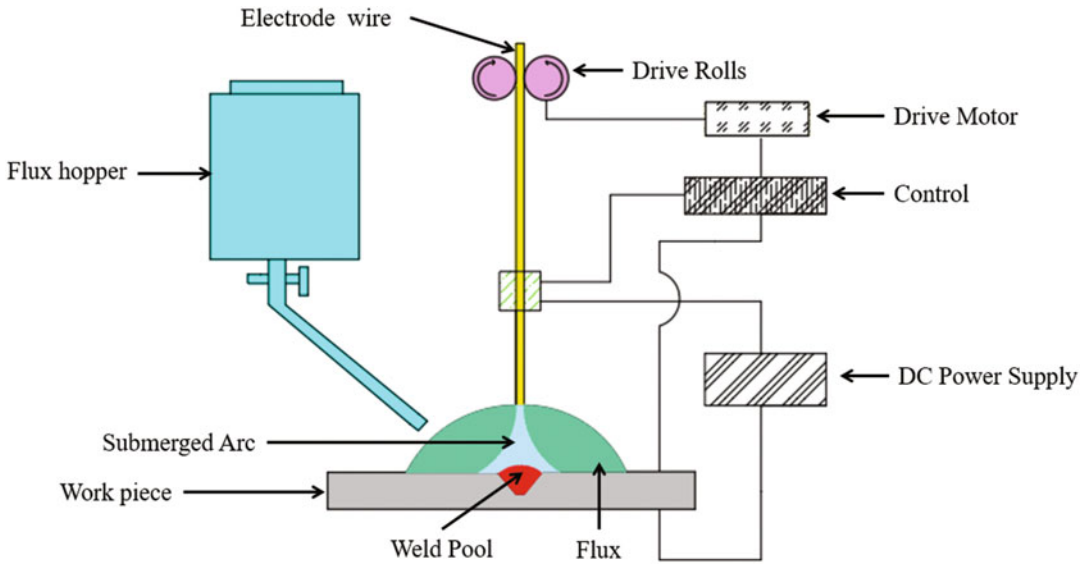
Underwater Welding,**Fig. 3** The diagram of gas metal arc welding (GMAW) and weld area**Underwater Welding,****Fig. 4** The gas metal arc welding (GMAW) and welding zone

the continuous feeding of the welding wire, a shield gas feeds through a welding gun to protect the welding arc and molten pool from the atmospheric contamination. In the MIG welding process, the inert gas, for instance, argon-based shielding gas or a mix of argon and CO_2 , is employed to protect the welding from the contaminants in the air. However, the active welding gas such as 75% argon and 25% CO_2 is adopted as the shield gas to protect the welding arc and molten pool in MAG welding process. The GMAW welding process is acceptable for most materials and filler wires, and it's one of the most popular welding techniques because of its high productivity and ease of implement to robotic automation (Fig. 4).

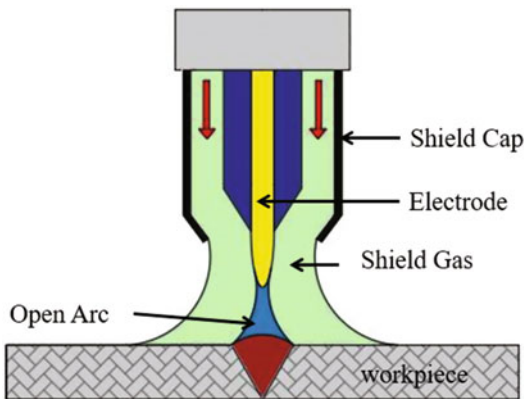
The SAW is a common automatically and continuously fed welding wire and a blanket of

granular fusible flux. The welding arc and molten pool are shielded from the contamination in the air by the submerged flux blanket consisting of lime silica, manganese oxide, calcium fluoride, and other components. When the blanket is molten, the flux becomes conductive and provides a current path between the welding wire and base metal. This existence of solid layer of flux covers the weld zone completely, thus preventing spatter and sparks as well as suppressing the intense ultraviolet radiation and fumes (Fig. 5).

The GTAW, known as TIG (tungsten inert gas) welding, employs a nonconsumable tungsten welding wire to perform the welding. The welding arc and molten pool are shielded from the atmospheric contamination and oxidation by the continuously injected inert shielding gas, and a filler metal is usually needed to melt and join the



Underwater Welding, Fig. 5 The diagram of submerged arc welding (SAW) process



Underwater Welding, Fig. 6 The gas tungsten arc welding (GTAW)

structure. The GTAW is usually employed to fabricate the thin stainless steel structures and the nonferrous alloys such as aluminum. Comparing with the other welding technique, the GTAW is relatively complex and difficult to perform, and it is slower than most other welding process (Fig. 6).

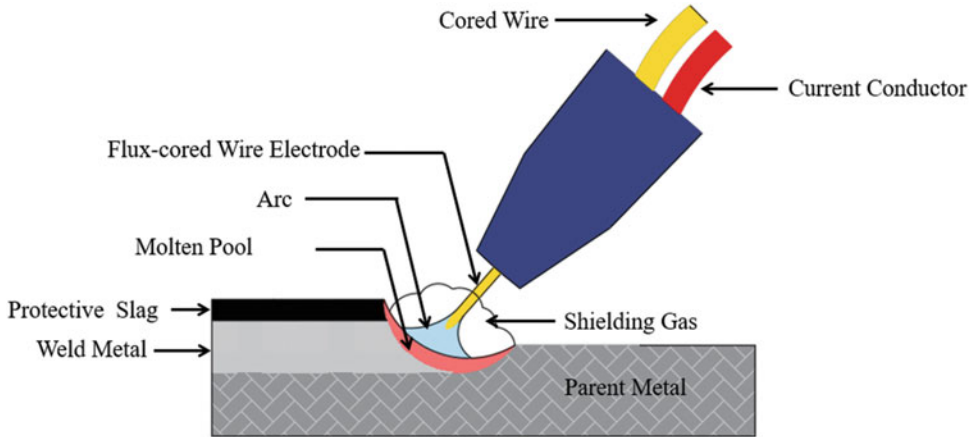
The FCAW is commonly identified as MIG while it employs a special tubular electrode filled with powder materials. Usually, this cored electrode is more expensive than the normal solid electrode and can produce slag and fumes during

the welding process. The FCAW can be performed with or without using the shield gas, depending on the filler (Fig. 7).

Recently, with the development of the high concentrated energy equipment, the laser technology, the electron beam welding, beam welding, and magnetic pulse welding have been used widely in the structural construction. Moreover, the solid-state welding processes such as friction welding in which the base metal does not melt have been developed recently.

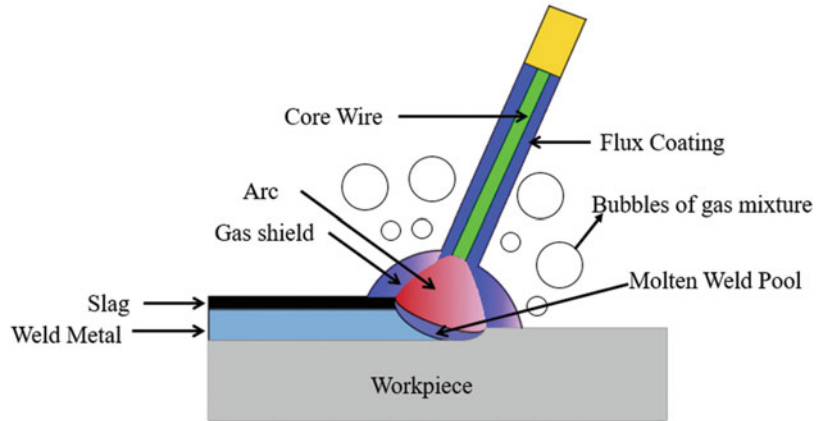
Underwater Welding Methodology

The welding technique described in the previous section is the common welding technology used widely in the steel construction industry. Moreover, the above described welding technique can be used for the underwater welding process. However, different from the ordinary welding process, in the underwater welding process, the arc is similar to submerged arc welding in underwater combustion and burned in the generated bubbles. When the welding electrode is burned, the coating surrounding the electrode generates a sleeve to stabilize the bubble. In order to make the welding



Underwater Welding, Fig. 7 The flux-cored arc welding (FCAW)

Underwater Welding, Fig. 8 The underwater welding diagram



electrode burn stably underwater, it is necessary to apply a certain thickness of the coating on the electrode core and impregnate it with paraffin or other waterproof material. Bubbles are composed of hydrogen, oxygen, water vapor, and the bubbles generated by the electrode burning and other oxides of turbid smoke. In order to overcome the difficulty of arc stabilization caused by water pressure and the lower temperature, the arcing voltage is higher than the one in the open-air welding process, and the current is 15–20% larger than the welding current in the ordinary welding process.

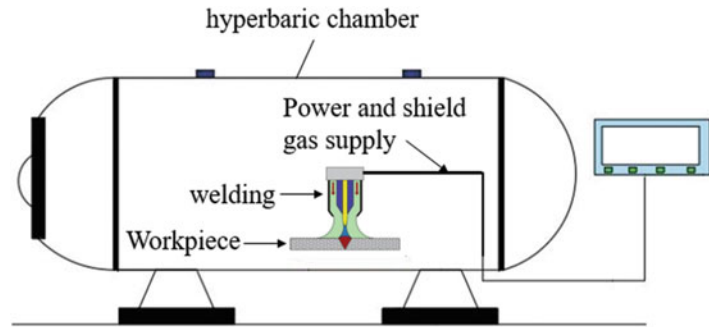
Normally, the underwater welding can be performed by three methods: wet welding, dry welding, and local dry welding. For wet welding,

usually the divers are entailed in the water to perform the weld directly, as shown in Fig. 2, with the help of the specific equipment. The basic aim of that specific equipment is to protect the diver safety. Usually, it involves using a specially fabricated welding rod and employs a similar process used in ordinary welding, as shown in Fig. 8. For instance, the SMAW is usually adopted for underwater welding in the offshore platforms construction and subsea pipeline repairing. The FCAW and TIG are also common for the underwater welding application.

The second method of underwater welding is dry welding or hyperbaric welding, as shown in Fig. 9. The dry welding is the process by which a chamber is sealed around the weld piece that is to

Underwater Welding,

Fig. 9 The underwater welding with a hyperbaric chamber



be welded. The chamber is then filled with a mixture of helium and oxygen, or argon gas, which then expels the water outside of the chamber and creates a cavity. This allows for a dry environment in which to perform the weld. In the specific designed hyperbaric chamber, the TIG, MIG, FCAW, and SMAW can be conveniently performed. Usually, it has two typical dry welding techniques, high-pressure dry welding and normal-pressure dry welding. For the high-pressure dry welding, the inner pressure of the chamber is higher than the pressure in the welding location; moreover, the inner pressure of the chamber is increased with increase of welding location water depth to expel the water. Usually, the manual metal arc welding and the flux-shielded arc welding are employed. For the normal-pressure dry welding, the inner pressure of the chamber is maintained as the atmospheric pressure. The welding process will not be influenced by the water depth. However, the chamber configuration for the normal-pressure dry welding is more complex in order to maintain the constant inner pressure (1 atm).

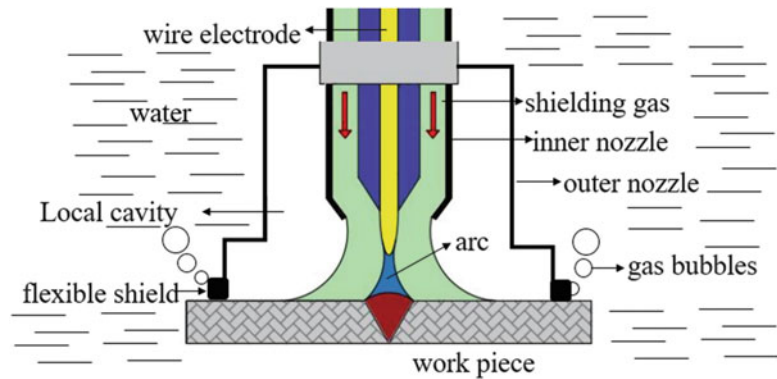
Similar to the wet welding, underwater local dry welding entails the diver preforms the welding in the water directly. Normally, a specific designed cover is employed to prevent the water into the weld zone. The gas is injected into the cover to separate the welding arc from the surrounding water. The welding process can be performed in the relatively dry environment. Usually, there are three types of dry welding technique, dry welding technique with large-scaled cover, dry welding method with medium-scaled cover, and dry welding process with small-scaled cover. In the

dry welding process with large-scaled cover, the welding is performed conveniently in a relatively large cavity, and the water was expelled out. In the dry welding process with medium-scaled cover, a special designed cover is employed, and the weld wire is conveniently inserted from a pre-designed channel. The welding process can be performed after the water is expelled by the inner rotating airflow. During the welding process, the gas generated by the welding will be expelled synchronously. In the third underwater local dry welding technology, as shown in Fig. 10, a small cover is composed together with the welding torch. The welding shield gas can separate water from the welding location. During the welding process, the local cavity can move with the welding torch to protect the welding arc.

Summary

The underwater environment makes the welding much more complicated than the open-air welding process. In addition, the underwater welding also involves many factors such as diving operation technology. Usually, the continuous welding procedure is hard to perform due to the limitation of the underwater condition. Normally, the section of welding process is employed and that will induce the discontinuities of the welding. Except that, some other characteristics of the underwater welding can be summarized.

Since the effect of light underwater reflection, absorption, and refraction is much stronger than that of air, the welding visibility is worse in the underwater condition. Moreover, a large amount

Underwater Welding,**Fig. 10** Underwater local dry welding process

of air bubbles and smoke are induced around the welding arc, and the visibility of the welding arc is even worse in underwater condition. The underwater arc will cause thermal decomposition of the surrounding water, resulting in an increase in hydrogen dissolved in the weld. The poor quality of the welded joint is strongly dependent on the hydrogen content generated during the underwater welding.

Since the heat transfer coefficient of water is high, which is about 20 times that of air, the cooling rate is high in the underwater welding process. When the workpiece is to be welded directly in the water, the cold-water quenching effect on the weld is obvious, and it is easy to produce a hardened structure with high embrittlement.

Cross-References

- ▶ [Dynamic Behavior and Fatigue](#)
- ▶ [Welding Technology](#)

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Underwater Wireless Acoustic Communication

- ▶ [Underwater Acoustic Communication](#)
- ▶ [Underwater Acoustic Sensor Network](#)

Uneven Seabed

- ▶ [Station-Keeping System for VLFS](#)

United Nations (UN)

- ▶ [Decommissioning of Offshore Oil and Gas Installations](#)

Unlimited Tension Interface

- ▶ [Shallow Foundations](#)

Unmanned Surface Vehicle (USV)

- ▶ Semisubmersible Vehicle Autopilot Control System
- ▶ Underwater Information Sensing Technology

Unmanned Underwater Vehicles (UUVs)

- ▶ Environmental Perception for Underwater Vehicles

Unmanned Underwater Vehicle (UUV)

- ▶ Autonomous Underwater Vehicle (AUV)
- ▶ Remotely Operated Vehicle (ROV) in Subsea Engineering
- ▶ Semisubmersible Vehicle Autopilot Control System

UWA – University of Western Australia

- ▶ Pile Capacity

V

Variable Weight

- [AUV/ROV/HOV Hydrostatics](#)

Vertical Load Anchor

- [Taut Mooring](#)

Very Large Floating Structures (VLFS)

- [Mega-Float](#)

Very Large Floating Structures (VLFS): Overview

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Definition

Very Large Floating Structures (VLFS) constitute a unique concept of ocean structures because of their unprecedented length (on the kilometer

scale), displacement (millions of tons), and nearly billion-dollar predicted cost (Suzuki et al. 2006).

VLFS also have the following characteristics:

1. Hydroelastic response in structural design (Cui et al. 2000).
2. Modular configuration with connectors
3. Use under inhomogeneous environmental sea conditions (Wu et al. 2013).
4. Long design life: 50 ~ 100 years
5. Deployed semipermanently or permanently with complex station-keeping systems

Scientific Fundamentals

Historical Development

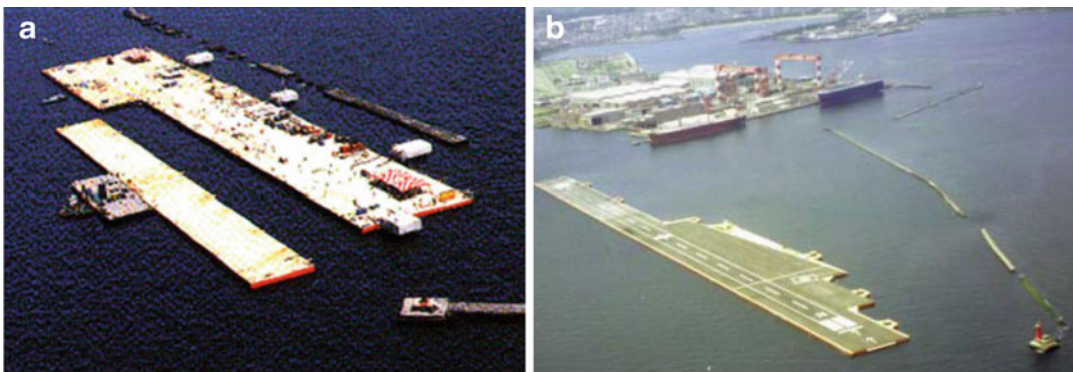
The idea of the Very Large Floating Structures (VLFS) first appeared in the modern world in the form of the floating island, which was described in the nineteenth century by the French novelist Jules Verne (Lamasparto et al. 2015). The first to be promoted in serious terms was in circa 1920, when the American Edward R. Armstrong proposed a sea station to be used as an “aeroplane supply and navigating station” (McAllister 1997). Although this proposal was rejected, Armstrong’s early idea that incorporated the principle of a small waterplane area and column-supported platform to minimize motion has been commonly used currently in the design of semisubmersible mobile offshore drilling equipment. Until the period of the Second World War, the US Navy

constructed a miniature floating airport measuring $552 \text{ m} \times 83 \text{ m} \times 1.5 \text{ m}$ with a draft of 0.5 m (Cui et al. 2007).

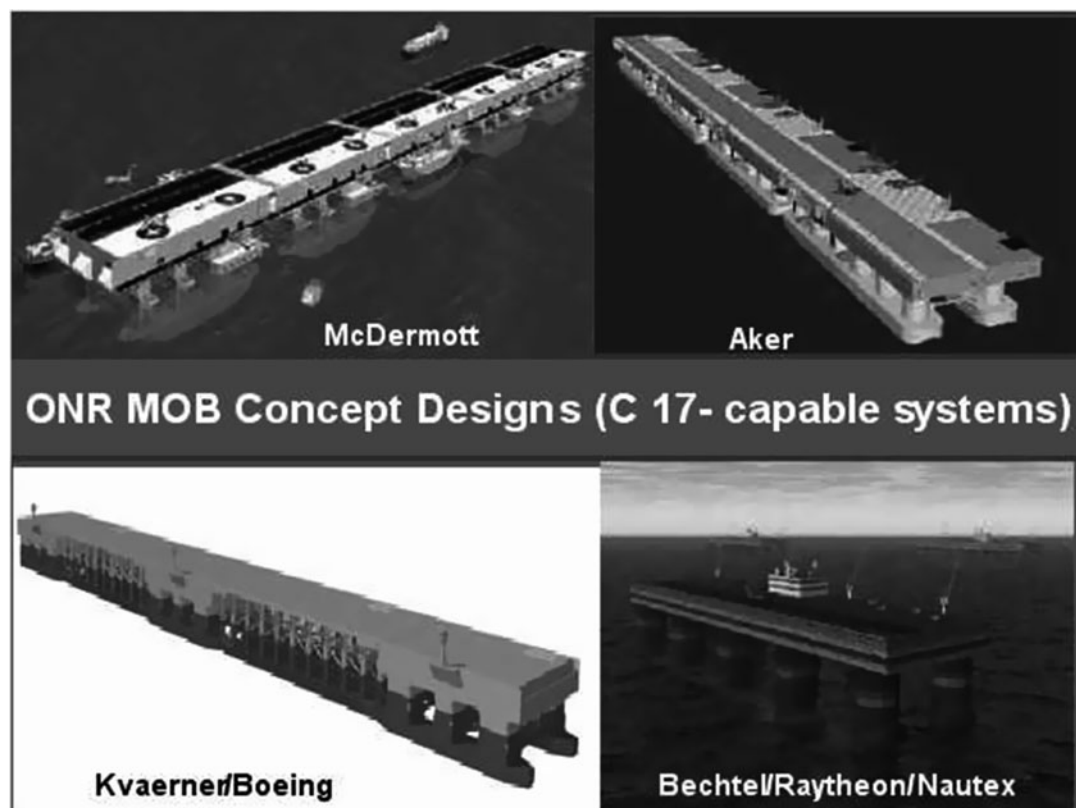
In recent decades, studies of VLFS were carried out in two major projects. One is the Mega-Float project of Japan. The other one is the Mobile Offshore Base (MOB) project of the USA. In addition, four international conferences on VLFS have been held in the USA and Japan (VLFS 1991, 1996, 1999, 2003). In 2006, the International Ship Structure Conference (ISSC) conducted a comparative study on the hydroelastic response of VLFS (Wu et al. 2017).

In Japan, due to the limited availability of civil land and frequent earthquakes, the Mega-Float project conducted by the Technological Research Association of Mega-Float (TRAM), was established in 1995. The Mega-Float project was divided into two phases. In phase I, from 1995 to 1997, a large floating experimental model 300 m long, 60 m wide, and 2 m deep was built to test the practical issues such as basic design, construction, and technical methods of a Mega-Float, as shown in Fig. 1a. Phase II, which began in 1998, was mainly to examine application technologies required to support the Mega-Float-type airport. In addition, TRAM built a model 1000 m long, 60 m wide (partially 120 m), and 3 m deep in Yokosuka to experiment with landing and takeoff of small airplanes (Suzuki et al. 2006), as shown in Fig. 1b.

MOB is a multifunctional floating comprehensive military base that can be placed in coastal or



Very Large Floating Structures (VLFS): Overview, Fig. 1 Two Mega-Float experimental models (Suzuki et al. 2006)



Very Large Floating Structures (VLFS): Overview, Fig. 2 Mobile Offshore Base Concepts (Suzuki et al. 2006)

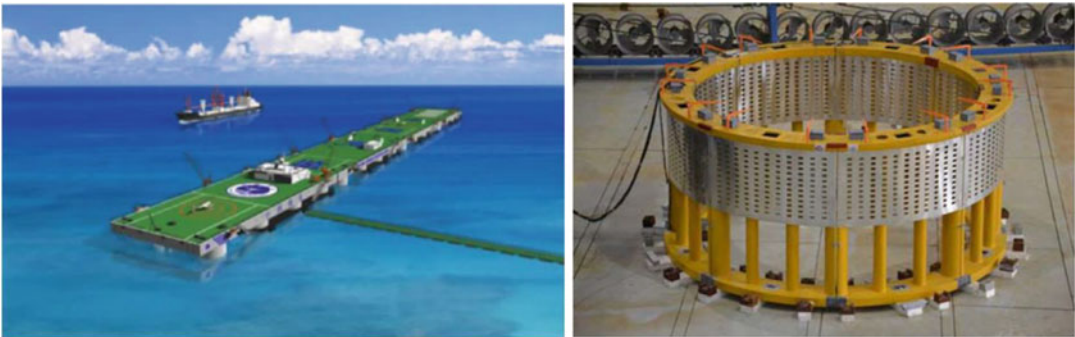
international seas and is assembled/disassembled at sea by several self-propelled semisubmersible structural modules, as depicted in Fig. 2. With the MOB concept, the USA could have a base anywhere in the world within 1 month (Lamasparto et al. 2015). In the USA, the first engineering project similar to the MOB was sponsored by the Defense Advanced Research Program Agency (DARPA) with a budget of \$40 M in the early 1990s (McCallister 1997). The Office of Naval Research (ONR) continued to fund the MOB program from 1997 to 2000.

To date, the world's major coastal countries have engaged in the research of VLFS. To develop the fishing and tourism resources near the coral reefs, and to assume the function of wave absorption and island protection, China has funded some research projects on VLFS (Cui et al. 2000; Wu et al. 2013) (see Fig. 3). Because the water depth

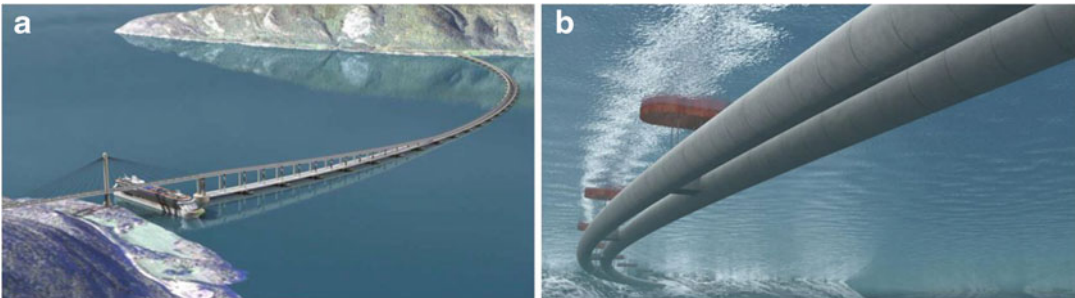
in fjords is so great, nearly 500 ~ 1500 m, the Norwegian Public Roads Administration (NPRA) is planning to build several floating bridges/submerged floating tunnels for crossing the fjords on the E39 (Xiang et al. 2016), as shown in Fig. 4. South Korea has also initiated several VLFS projects, such as three floating islands on the Han River and a floating cruise terminal for Seoul (Wang and Tay 2011).

Types of VLFS

VLFS can be generally classified as either pontoon-type or semisubmersible type according to their structure type (Wang and Tay 2011). Pontoon-type VLFS have lesser construction and maintenance costs, while the semisubmersible structures have better wave damping capability (Cui et al. 2000). Figure 5 shows a schematic representation of the two types of VLFS.

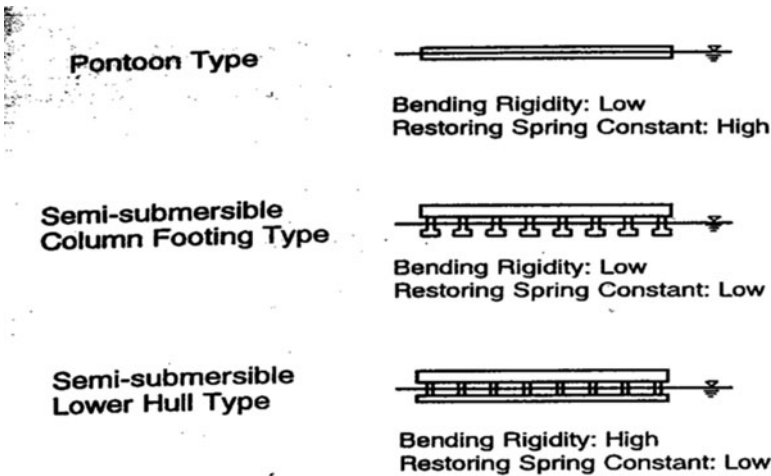


Very Large Floating Structures (VLFS): Overview, Fig. 3 VLFS in the South China Sea (Wu et al. 2017; Kou 2011)



Very Large Floating Structures (VLFS): Overview, Fig. 4 VLFS in Norway (<https://www.vegvesen.no/en/home>)

Very Large Floating Structures (VLFS): Overview, Fig. 5 Typical geometry types of VLFS (Cui et al. 2007)



(I) Semisubmersible Type

Generally, the semisubmersible-type VLFS have a raised platform above sea level using columnar tubes. There are mainly two types of semisubmersible VLFS proposed (Cui et al. 2007). One

type has pontoons supported on columns of large diameter and small depth (the column footing type). The other type has pontoons supported by the column and lower hull, whose shape is obtained by extending the present semisubmersible rigs with two lower hulls (the lower hull type).

Semisubmersible VLFS are more suitable for deployment in high seas with large waves and can be stationed in coastal or international waters.

(II) Pontoon Type

Pontoon-type VLFS are also known in the literature as either Box Type or Mat-like because of their box shape and small draft in relation to the length. The pontoon-type VLFS platform rests on the water surface and is intended for deployment in calm waters such as in a cove, a lagoon, or a harbor (Wang and Tay 2011).

Key Technologies

From the investigations of Mega-Float and MOB, the key technical problems involved in the development of VLFS can be summarized in six aspects.

Inhomogeneous Environments

With their relatively small size, the traditional marine structures are generally assumed to be under homogeneous environments. However, for VLFS with several kilometers of length, the sea environments at both ends of the head and tail may be quite different. The differences may include (1) wave conditions; (2) current conditions; (3) wind conditions; and (4) nonuniform topography (Cui et al. 2007). Moreover, the VLFS is usually deployed in nearshore seas with varying water depths (Utsunomiya et al. 2001), offshore seas sheltered by breakwaters (Nagata et al. 1998), or fjords/islands with complex topography (Wei et al. 2019; Wu et al. 2013). In addition, the ocean waves can be easily affected by islands, breakwaters, or the shores of fjords/islands when propagating from the open sea with certain angles, and the wavefield will then become inhomogeneous. These inhomogeneous environments can lead to large loads for the connectors, mooring systems, and other components of the VLFS.

Hydroelastic Response

Distinguished from traditional marine structures by their large size and relatively low structural stiffness, VLFS exhibit responses in waves that

include rigid body motions and obvious structural deformations. Because of the strong coupling between structural deformations and fluid motions, the responses of VLFS should be determined by hydroelasticity theories. Hydroelastic responses are the main causes of the structure and connector failure such as buckling and fatigue. Thus, the accurate prediction of the hydroelastic responses of a VLFS is a prerequisite for a safe and economic design. In the last several decades, frequency domain hydroelasticity theories have been developed from 2D (Bishop and Price 1979) to 3D (Wu 1984), from linear (Wu and Wang 1993) to nonlinear theories (Chen et al. 2003b), and applied to estimate hydroelastic responses of VLFS. However, the studies of hydroelasticity analysis of VLFS have been mainly restricted to homogeneous waves. When the oceanic environment around the VLFS is inhomogeneous, the traditional hydroelasticity theory based on homogeneous waves is no longer applicable. Even so, the studies of hydroelastic responses under inhomogeneous sea environments are still relatively few (Wu et al. 2013, 2017).

Station-Keeping Systems for VLFS

The station-keeping system ensures that the VLFS is kept in position so that the facilities installed on the structure can be reliably operated and to prevent the structure from drifting away under critical sea environments (Wang and Tay 2011). Similar to traditional marine structures, the station-keeping systems for VLFS can be grouped into three main types: (1) the dolphin fender mooring system, (2) the mooring line system, and (3) the dynamic positioning system. In shallow water area, a dolphin fender mooring system is the most reliable, albeit expensive. Catenary chain mooring is available even in shallow water if a midpoint sinker or buoy is used. If the water depth is over 500 m, compliant type mooring, taut mooring, or dynamic positioning systems are preferable to the catenary mooring system (Cui et al. 2007). Due to the massive size of the VLFS, the station-keeping system design and analysis have become more complex.

Connectors for VLFS

The massive continuum structure of a VLFS is usually divided into several separate modules.

The modules are fabricated in the shipyard and then connected on-site in the open sea by welding or using connectors. For multimodular floating systems, the connector between modules becomes the weakest component in the whole system, which means it is subjected to large loads and easily damaged. For thousand-meter-scale floating systems, the connection load may exceed 10,000 t in the sixth sea state (Riggs et al. 2000). Various connector designs have been proposed for VLFS, including rigid or hinge connectors (Maeda et al. 1979), semirigid connectors (Fu et al. 2007), flexible connectors (Wu et al. 2016), and other compliant connectors (Xia et al. 2016). However, the strength and economic issues of the connectors have still been a long-standing technical bottleneck for developing multimodular floating systems.

Structural Analysis and Design of VLFS

The purpose of structural design is to develop a structure that can satisfy the maintainability and safety requirements in an economic manner. For innovative structures such as VLFS, structural design needs to be based on reasonable structural response analysis and explicit design criteria (Suzuki et al. 2006). However, there is no complete specification or standard to follow during the design of VLFS. It is mainly designed according to functional requirements, and then the safety of the structure is evaluated by direct numerical calculation or model test methods. The safety design limit states for VLFS include the ultimate limit state (ULS), the fatigue limit state (FLS), the serviceability limit state (SLS), and the progressive collapse limit state (PLS) (Fujikubo 2005). The manner of establishing the corresponding design criteria and procedures has always been the focus of scholarly research (Cui et al. 2000).

Experimental Techniques for VLFS

The development of any new theory/method is inseparable from its experimental verification. The two main approaches to carrying out experimental research are full-scale measurement and model tests. Compared with full-scale measurements, model tests of VLFS are more practical and economical (Cui et al. 2007). Though model tests of a VLFS are very important, there are several

peculiar problems in model experiments. To conduct model experiments in wave basins, it is necessary to reduce the scale of the very large



Very Large Floating Structures (VLFS): Overview, Fig. 6 Floating Pier at Ujina, Japan (Watanabe et al. 2004)



Very Large Floating Structures (VLFS): Overview, Fig. 7 Kamigoto floating oil storage base, Japan (Wang and Tay 2011)



Very Large Floating Structures (VLFS): Overview, Fig. 8 Performing stage at Marina Bay, Singapore (Wang and Tay 2011)

(kilometer scale) prototype to an extraordinarily small scale. This may cause a problem in the wave simulation capability of wave basins and the accuracy of measuring the behavior of the model. Moreover, the simulation of inhomogeneous sea environments is also a great challenge to wave-making technology in wave basins. In short, the characteristics of VLFS greatly increase the difficulty of model tests, and it is quite necessary to carry out further research (Chen et al. 2003a).



Very Large Floating Structures (VLFS): Overview, Fig. 9 Seoul Floating Islands, South Korea (www.archdaily.com)

Key Applications

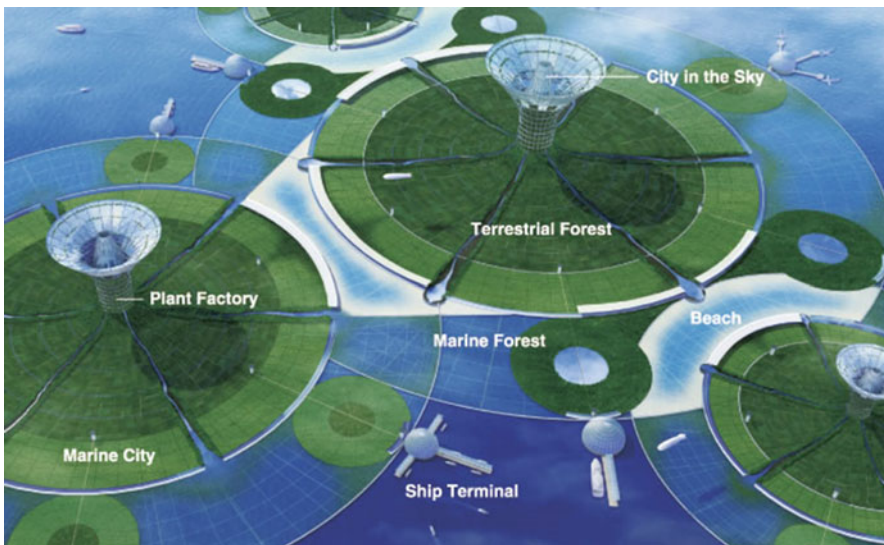
VLFS are designed primarily for floating airports (see Fig. 1), floating military bases (see Fig. 2), and floating bridges (see Fig. 4). However, VLFS could be used for a variety of purposes.

Floating Ports

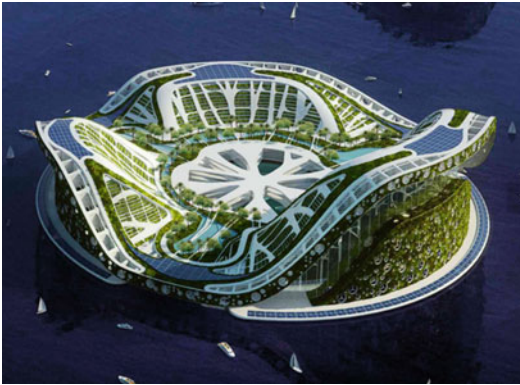
Floating structures are ideal choice for ports, because the ships can come alongside them since their positions are constant with respect to the waterline (Watanabe et al. 2004). Moreover, the floating ports may also be beneficial in siting offshore termini for potentially hazardous vessels, such as LNG carriers (Suzuki et al. 2006). An example of a floating port is Ujina Port at Hiroshima, which was built by Japan in 1993 (see Fig. 6).

Energy Storage

Another important use of VLFS is the development of facilities for fuel storage. The Kamigoto Oil Storage Base is a facility of 5 units of $390 \times 97 \times 27.6$ m, which is one of the nation's ten national oil stockpile bases, for which see Fig. 7 (Suzuki et al. 2006; Wang and Tay 2011). This floating oil storage is equipped to store different kinds of oil, such as gasoline, domestic fuel, and jet fuel.



Very Large Floating Structures (VLFS): Overview, Fig. 10 Floating city, Japan (www.wordlesstech.com)



Very Large Floating Structures (VLFS): Overview, Fig. 11 Floating city, Belgium (www.vincent.callebaut.org)

Recreation and Residential Areas

VLFS could be constructed as recreation and residential facilities, including restaurants, hotels, and performance stages. Singapore has built the world's largest floating performance stage. It is 120 m long, 83 m wide, and 1.2 m deep, as shown in Fig. 8, and is located at Marina Bay. With convention halls, restaurants, and other tourist attractions, the Seoul Floating Islands have been built by South Korea, as shown in Fig. 9.

Floating Cities

With the development of VLFS technologies, it is possible that large human habitation and floating cities will be built on the ocean surface in the future. These facilities can either be fixed to coastal cities or free-floating in international waters. A design concept of the future water city proposed by Japan's Shimizu Technology Corporation (see Fig. 10), focusing on the use of green science concepts to build a carbon-neutral city, is planned to be completed in 2025. The famous Belgian architectural designer Vincent Callebaut proposed the "Lily Petal" water mobile city model, which is a floating amphibious city that can float on the earth according to different wind directions and climates, as shown in Fig. 11.

Cross-References

- [Connectors of VLFS](#)
- [Floating Bridge](#)

- [Hydroelasticity Theory](#)
- [Mega-Float](#)
- [Station-Keeping System for VLFS](#)
- [Structural Analysis and Design of VLFS](#)

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Vessel Dismantling

- [Complete and Partial Dismantling](#)

Vessel Positioning

- [Thruster-Assisted Mooring](#)

Vessel Recycling

- [Complete and Partial Dismantling](#)

Vibration and Noise of Icebreakers

- [Ice-Induced Vibration and Noise of Ships](#)

Vibration and Noise of Ships Under Ice Collision

- [Ice-Induced Vibration and Noise of Ships](#)

VIM (Vortex-Induced Motion)

- [Semi-submersible Platform](#)

Viscous Resistance

- [AUV/ROV/HOV Resistance](#)

Vortex Shedding and VIV Suppression

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Introduction

Vortex-induced vibration (VIV) is a fluid-structure interaction (FSI) phenomenon in which the structure is excited into vibration under the forces induced by the vortices alternatively shed from the surface of a bluff body. VIV is a major concern which has the potential to cause the fatigue failure of the structures in many engineering applications. The suppression of VIV minimizes the fatigue effects prolonging the integrity of the structures and avoiding or deferring the

costs associated with replacement or repair of the structures.

Vortex Shedding

Definition of Vortex Shedding

Vortex shedding is the phenomenon in which vortices are shed alternatively at either side of a bluff body at a certain frequency, and consequently the wake has an appearance of a vortex street as illustrated in Fig. 1.

Mechanism of Vortex Shedding

Flows past bluff bodies with different cross-sectional shapes, to some extent, share similar characteristics. Herein, the flow features around a circular cylinder are taken as an example due to the facts that the key structural components in offshore engineering are usually of a cylindrical shape and the flow past a circular cylinder has been extensively studied in the literature.

When a fluid particle flows toward the leading edge of a circular cylinder, the pressure in the fluid particle experiences a rise from the freestream pressure to the stagnation pressure. The high fluid pressure near the leading edge impels the developing boundary layers about both sides of the cylinder. However, the pressure forces are not sufficient enough to force the boundary layers around the back side of the cylinder at certain

flow velocities. Near the widest section of the cylinder, the boundary layers separate from each side of the cylinder surface and form two free shear layers which trail aft in the flow and bound the wake. Since the innermost portion of the free shear layers moves much more slowly than the outermost portion of the layers, which are in contact with the free stream, the free shear layers tend to roll up into two discrete swirling vortices. The pair formed by these two vortices is actually unstable when exposed to small disturbances for Reynolds numbers $Re = \frac{UD}{\nu} > 40$, where U is the freestream velocity, D is the diameter of the cylinder, and ν is the kinematic viscosity of the fluid. Consequently, one vortex will grow larger than the other if $Re > 40$. The larger vortex (Vortex A in Fig. 2a) presumably becomes strong enough to draw the opposing vortex (Vortex B) across the wake, as sketched in Fig. 2a. The vorticity in Vortex A is in the clockwise direction, while that in Vortex B is in the counterclockwise direction. The approach of vorticity of the opposite sign will then cut off further supply of vorticity to Vortex A from its boundary layer. This is the instant where Vortex A is shed. Vortex A is then convected downstream by the flow once it becomes a free vortex.

Following the shedding of Vortex A, a new vortex will be formed at the same side of the cylinder, namely, Vortex C (Fig. 2b). Vortex B will now play the same role as Vortex A, namely, it will grow in size and strength so that

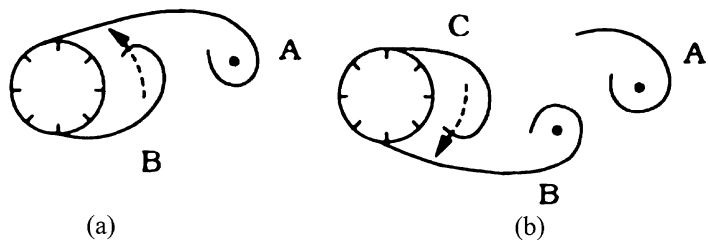
Vortex Shedding and VIV Suppression,

Fig. 1 Appearance of vortex shedding behind a circular cylinder in a stream of oil at $Re = 161$ (Homann 1936)



Vortex Shedding and VIV Suppression,

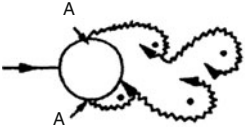
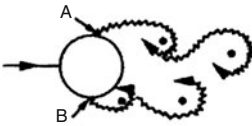
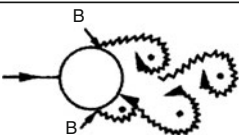
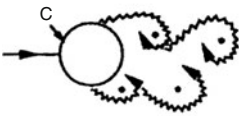
Fig. 2 Mechanism of vortex shedding (Sumer and Fredsøe 2006)



it will draw Vortex C across the wake (Fig. 2b). This will lead to the shedding of Vortex B. This process will continue each time a new vortex is shed at one side of the cylinder where the shedding will continue to occur in an alternative manner between the sides of the cylinder.

Flow Regimes Around a Circular Cylinder

The flow around the cylinder can be classified into different regimes as Re is increased from zero. Figure 3 presents different flow regimes around a smooth circular cylinder as summarized in Sumer and Fredsøe (2006).

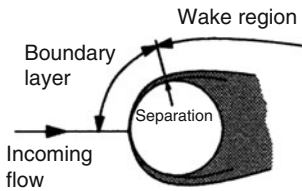
a		No separation. Creeping flow	$Re < 5$
b		A fixed pair of symmetric vortices	$5 < Re < 40$
c		Laminar vortex street	$40 < Re < 200$
d		Transition to turbulence in the wake	$200 < Re < 300$
e		Wake completely turbulent. A: Laminar boundary layer separation	$300 < Re < 3 \times 10^5$ Subcritical
f		A: Laminar boundary layer separation B: Turbulent boundary layer separation; but boundary layer laminar	$3 \times 10^5 < Re < 3.5 \times 10^5$ Critical {Lower transition}
g		B: Turbulent boundary layer separation; the boundary layer partly laminar partly turbulent	$3.5 \times 10^5 < Re < 1.5 \times 10^6$ Supercritical
h		C: Boundary layer com- pletely turbulent at one side	$1.5 \times 10^6 < Re < 4 \times 10^6$ Upper transition
i		C: Boundary layer com- pletely turbulent at two sides	$4 \times 10^6 < Re$ Transcritical

Vortex Shedding and VIV Suppression, Fig. 3 Regimes of flow around a smooth circular cylinder (Sumer and Fredsøe 2006)

When $Re < 5$, the flow attaches to the cylinder surface and no separation occurs. With the increase in Re , due to the adverse gradients imposed by the divergent geometry of the flow environment, the boundary layers separate from each side of the cylinder surface becoming two free shear layers in the wake. Flow separation as sketched in Fig. 4 first appears when Re becomes 5. For the range of $5 < Re < 40$, a pair of symmetric vortices forms in the wake of the cylinder. The vortex formation length increases with increasing Re (Batchelor 1967). When Re is further increased, the wake becomes unstable, which would eventually give birth to the vortex shedding phenomenon.

Vortex Shedding Frequency

The frequency at which the alternative vortices are shed, f_s , can be normalized with the flow velocity U and the cylinder diameter D to form the so-called Strouhal number.



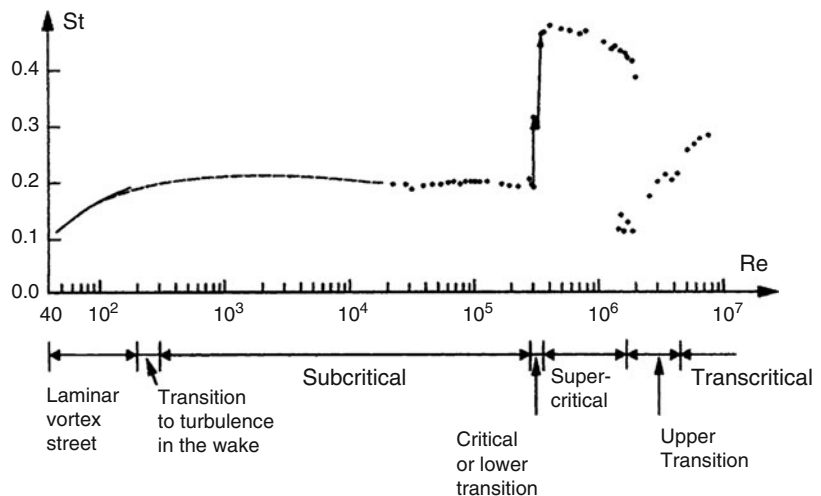
Vortex Shedding and VIV Suppression, Fig. 4 A sketch of flow separation (Sumer and Fredsøe 2006)

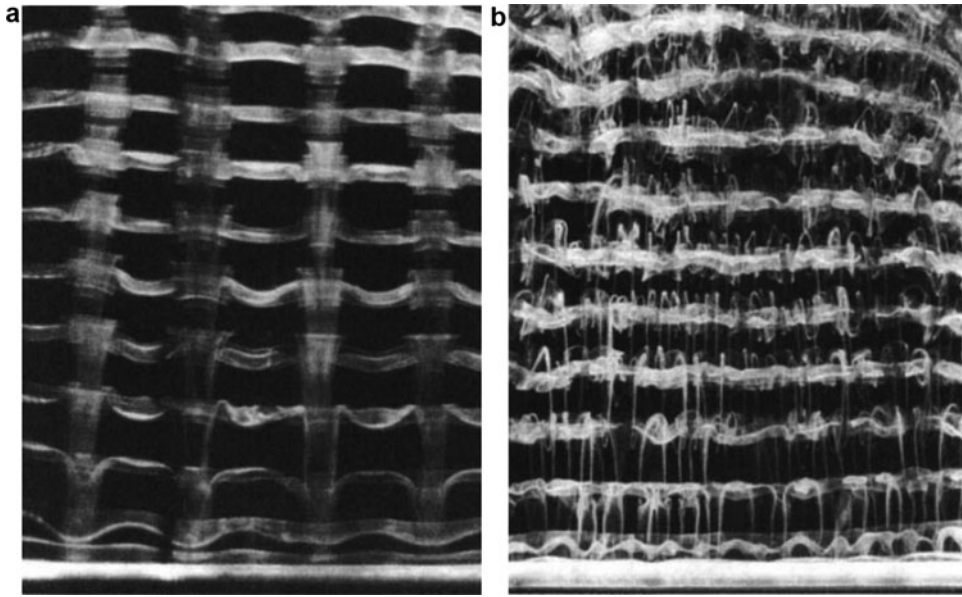
$$St = \frac{f_s D}{U}$$

Since the flow around a circular cylinder is governed by Re , St can on dimensional grounds be seen to be a function of Re . Figure 5 presents the variation of St with Re . The report of St starts from $Re = 40$. For $40 < Re < 200$, the vortex street is laminar, and the vortex shedding is essentially 2D. In this range, St increases linearly with Re . The transition in the wake region occurs when $200 < Re < 300$. Two discontinuities are observed in St in the experimental data by Williamson (1992). The dislocation of the vortices leads to the first discontinuity as the flow changes from 2D to Mode A (see Fig. 6a). The second discontinuity occurs as the flow transits from Mode A to Mode B as shown in Fig. 6b. In the Re range of $300 < Re < 3 \times 10^5$, the wake is completely turbulent, whereas the boundary layer over the cylinder surface remains laminar. This regime is known as the subcritical flow regime where St is almost constant at $St \approx 0.2$. With a further increase in Re , the flow enters the critical regime ($3 \times 10^5 < Re < 3.5 \times 10^5$). In the critical regime, the boundary layer separation is turbulent on one side of the cylinder and laminar on the other side. The next flow regime is the supercritical regime where $3.5 \times 10^5 < Re < 1.5 \times 10^6$. In this regime, the boundary layer on both sides of the cylinder is turbulent at the separation points. This results in a

Vortex Shedding and VIV Suppression,

Fig. 5 Variation of the Strouhal number with the Reynolds number for a smooth cylinder (Williamson 1989; Roshko 1961; Schewe 1983)





Vortex Shedding and VIV Suppression, Fig. 6 Mode A and B three-dimensional vortex shedding. (a) Mode A represents the inception of streamwise vortex loops,

and (b) Mode B represents the formation of finer-scale streamwise vortex pairs (Williamson 1992)

delay in the boundary layer separation where the separation points move downstream. This means the vortices would interact at a faster rate. Therefore, a dramatic increase in St from 0.2 to 0.45 is seen in this regime. The Strouhal number experiences another discontinuity when Re reaches the value of 1.5×10^6 . At this Re , transition to turbulence in one of the boundary layers has been completed (Fig. 3h). The boundary layer on one side of the cylinder is completely turbulent, and that on the other side is partly laminar and partly turbulent. The asymmetric situation causes the formation of the lee-wake vortices, and it prevails over the whole upper transition regime ($1.5 \times 10^6 < Re < 4 \times 10^6$). The regular vortex shedding is re-established in the transcritical flow regime where St takes the value of 0.25–0.3.

Vortex-Induced Vibration (VIV) Suppression

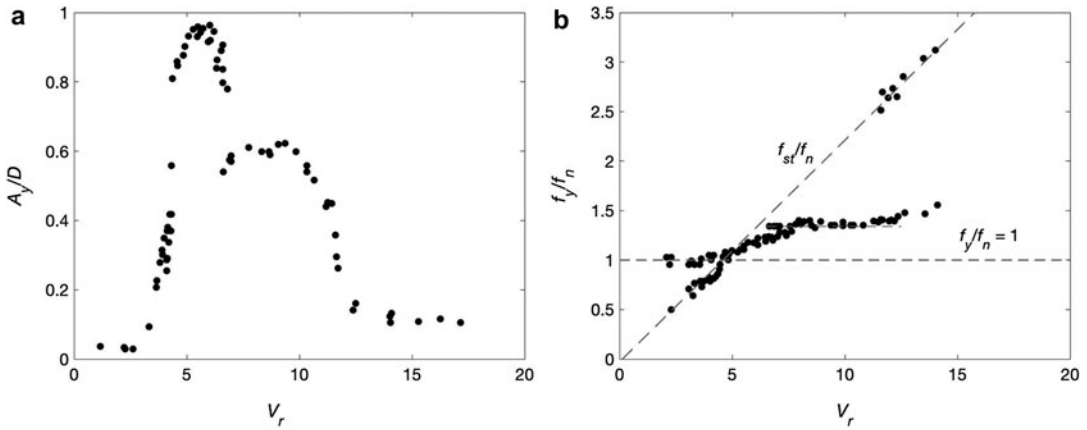
Vortex-Induced Vibration (VIV) Suppression and Its Principles

As VIV is induced by the alternatively shed vortices, adjusting the locations of the separation

points, varying the fluid forces, and controlling strengths of the vortices in the wake could mitigate the effects of VIV. Vortex-induced resonance rises when the structural vibration frequency is close to the natural frequency of the structure. Therefore, modifying the characteristic length, changing the cross-sectional shape, or adding appendages can effectively avoid the occurrence of vortex-induced resonance.

Modifying Structural Parameters

Figure 7(a) shows the one-degree-of-freedom VIV response of an elastically mounted circular cylinder with a low mass ratio $m^* = 2.4$ (Khalak and Williamson 1997). It can be seen that when the reduced velocity ($V_r = U/f_n D$) is low, the amplitude of the circular cylinder is relatively small, and the oscillation frequency f_y follows the stationary-cylinder Strouhal frequency f_{st} until it approaches the natural frequency of the circular cylinder f_n . From this point on, with a further increase in V_r , f_y deviates from f_{st} , and the ratio of f_y to f_n remains constant. The V_r region in which f_y is close to f_n is referred to as the lock-in range (Williamson and Govardhan 2004). In this



Vortex Shedding and VIV Suppression, Fig. 7 One-degree-of-freedom VIV responses of a circular cylinder: (a) amplitude response and (b) frequency response (Khalak and Williamson 1997)

corresponding range, the cylinder undergoes large-amplitude vibrations. Beyond the lock-in range, the vibration amplitude of the cylinder reduces and f_y returns to f_{st} .

Based on the descriptions above, it can be concluded that the VIV amplitude reaches its maxima when the cylinder is subject to vortex-induced resonance. Therefore, the large-amplitude resonant response of the cylinder under certain incoming flow conditions could be inhibited by modifying the structural parameters. The characteristic length or the natural frequency might be changed to keep away from the lock-in-reduced velocities. In order to suppress VIV of risers or pipelines, the statistics about the flow velocity range of the sea area are a priori. Although the sizes of risers or pipelines are usually restricted by the rules and regulations, their characteristics lengths could be modified by adding insulations. Varying the natural frequencies of risers or pipelines can be achieved by changing their masses, lengths, bending stiffness, and/or tensions (if applicable).

Adjusting Boundary Layer Separation Point Locations

The locations of the boundary layer separation points are related to the cross-sectional shape, the surface roughness, and the Reynolds number. For bluff bodies with sharp corners (e.g., a square cylinder), the separation points are usually at the

corners. The cross-sectional shape has to be changed in order to change the separation point locations.

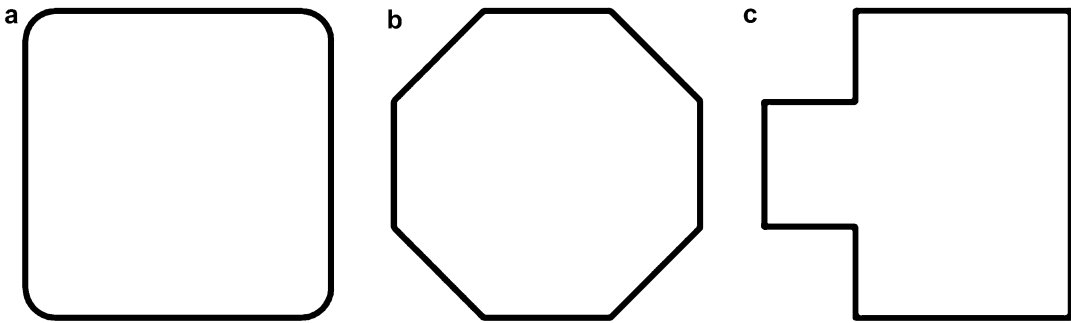
Figure 8 demonstrates three modifications to the cross-sectional shapes which could adjust the locations of boundary layer separation points. By modifying the corners, the reattachment of the separated shear layers is promoted. As a result of the reattachment, the strength of the vortex street is decreased, and the wake width is narrowed (Tamura and Miyagi 1999; Ueda et al. 2009).

For circular cylinders, the boundary layer separation locations can be directly shifted by changing their cross-sectional shapes, for example, installing a fairing on the rear of a cylinder (see Fig. 9). In this way, the bluff circular cylinder is converted to streamlined shape so that the boundary layer separation point locations are modified.

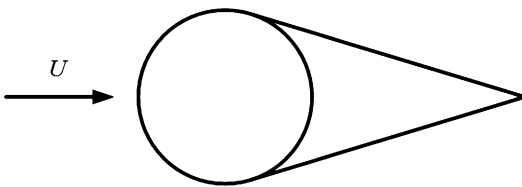
In addition, the locations of the boundary layer separation points might be delayed by injecting momentum into the wake flow. As shown in Fig. 10, a nozzle is installed on the rear side of the cylinder (Feng et al. 2010, 2011; Gao et al. 2019). By pumping extra fluid into the wake as a high-speed jet, the shear layers on both sides of the cylinder move toward the centerline to shift the boundary layer backward.

Suppressing Development of Wake Vortices

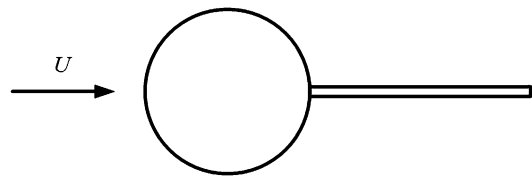
The fluid forces acting on the structures are dependent on the formation and convection of the



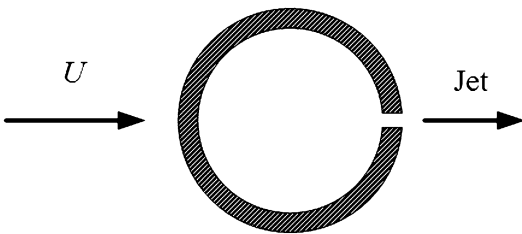
Vortex Shedding and VIV Suppression, Fig. 8 (a) Square cylinder with round corners, (b) square cylinder with chamfered corners, and (c) square cylinder with cut corners at the front edges



Vortex Shedding and VIV Suppression, Fig. 9 A sketch of flow past a circular cylinder with a sharp-tail fairing



Vortex Shedding and VIV Suppression, Fig. 11 A sketch of flow past a circular cylinder with a splitter plate



Vortex Shedding and VIV Suppression, Fig. 10 A sketch of flow past a circular cylinder with synthetic jet into its wake

vortices. The fluid forces are larger when the vortices are formed, and the magnitudes of the fluid forces decrease as the vortices are convected downstream. Therefore, VIV could also be suppressed by interfering the formation and development of wake vortices.

Splitter plates as shown in Fig. 11 are typical devices used to suppress the development of wake vortices. They separate the wake region into the upper and lower parts so that the separated shear layers do not have enough space to roll up into

large vortices but are forced to break into small vortices or convected along the surfaces of the splitter plates. Furthermore, the presence of splitter plates also modifies the shear layer interaction in the near wake of the cylinder.

Typical VIV Suppression Devices

Helical Strakes

A widely used method for suppressing VIV of long slender bodies of circular cross section is the attachment of helical strakes. As shown in Fig. 12, helical strakes consist of one or more fins that spiral along the span of the structures. The fins cause the vortices along the span to be broken up into shorter and weaker segments. Furthermore, the breaking up of the vortices reduces their ability to correlate along the span resulting in a series of vortices randomly phased in time.

The helical strakes were invented by Christopher Scruton and D. E. J. Walshe at the National Physics Laboratory in the UK (Scruton and Walshe 1957) and have been subsequently employed in the wind engineering since the early 1960s. It was not until the late 1970s that the



Vortex Shedding and VIV Suppression, Fig. 12 Helical strakes

helical strakes were seen on slender marine structures for VIV suppression. The main advantage of helical strakes is that they are less sensitive to the direction of the flow whereas they suffer from two major problems: the first being that they increase drag and the second that for a given strake height, their effectiveness reduces with decreases in the response parameter $m\zeta$. Allen et al. (2006) discovered that helical strakes with a triple-start strake geometry of $0.1D$ high strakes with a $5D$ pitch-per-start can suppress VIV very efficiently at a wide range of Re .

Fairings

A second device for suppression of the VIV of offshore oil and gas tubulars is the fairing. As shown in Fig. 13, fairings consist of a structure having two sides with the aim of streamlining the flow past the structure. The short fairing was pioneered by Allen and Henning (Allen 1989; Allen and Henning 1995), while longer fairings were moderately successful when used on offshore drilling risers in the late 1970s and early 1980s. Allen and Henning (2007) found that short and even ultrashort fairings could be very effective at VIV suppression. Moreover, fairings have lower drag than helical strakes. Khorasanchi and Huang (2014) analyzed the vibration response of a circular cylinder with fairing. As fairings were added to the cylinder, the center of gravity of the system would shift from the cylinder axis, which may lead to the instability problem. Consequently,



Vortex Shedding and VIV Suppression, Fig. 13 Marine riser fitted with fairings

the fairings should be designed in such a way that the center of gravity of the system stays close to the axis of the cylinder. Additionally, the structural damping could be increased to ensure the stability of the system.

Splitter Plates

For helical strakes, sharp-edged separation with its accompanying high drag is required to maintain three-dimensional separation and suppress VIV. At values of $m\zeta$ typical for risers (less than 0.1), it seems that three-dimensional solutions for VIV suppression are unlikely to provide satisfactory effectiveness of VIV suppression with reduced drag. There are a number of two-dimensional control devices to weaken vortex shedding and reduce drag, with the most well-known being the splitter plate. The primary vortex shedding can be suppressed given that the splitter plates are sufficiently long. The experimental data by Huera-Huarte (2014) indicates that it is not necessary to cover the whole risers or pipelines with splitter plates. The splitter plates could be distributed at certain spacings. This kind of arrangement is not only cost saving but also can achieve good VIV suppression effectiveness.

Cross-References

- [Design of Pipelines and Risers](#)
- [Flexible Pipes and Umbilicals](#)
- [Steel Pipelines and Risers](#)

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Vortex-Induced Motions (VIMs)

- [Steel Pipelines and Risers](#)

Vortex-Induced Vibration (VIV)

- [Hydrodynamics for Subsea Systems](#)

Vortex-Induced Vibrations (VIV)

- [Steel Pipelines and Risers](#)

Warships

- [Surface Warships](#)

Waste Recovery System

- [New Technologies in Auxiliary Propulsions](#)

Water Depth

- [Shallow Foundations](#)

Water Surface Support (Layout and Recovery) System

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Synonyms

[Deep-sea mining](#); [Deployment and recovery](#);
[Water surface support system](#)

Introduction

The water surface support system is not only a floating platform for seabed mining, slurry lifting and conveying, a control center for seabed mining equipment, and a living place for staff, but also has special functions for dewatering the slurry lifted from the seabed and unloading the ore to the transport barge. It provides space for underwater mining equipment and mineral storage area, and supports water surface distribution and recovery and underwater mining operations (Liu et al. [2015](#)).

Main Components

The distribution and recovery system are mainly composed of the following subsystems:

Layout and recovery system of mining vehicle
Umbilical deployment and recovery system
Pipe connection and dismantling device
Hoisting system
Heave compensation system
Steel tower
Base platform
Pipeline storage transfer and auxiliary connection system
Water surface support cooperative control system
Pipe ship coupling device

Water Surface Support (Layout and Recovery)

System, Fig. 1 Water surface support (layout and recovery) system



Placing transfer device in underwater relay station
Hose placement and recovery system
Pipe clamping device (Fig. 1).

Function

Under the predetermined sea conditions, the reasonable, orderly, safe, and efficient placement and recovery operation of the underwater equipment and facilities involved in the seabed ore gathering and navigation positioning system, underwater transportation system, and power transmission and control system can adapt to the adverse impact caused by the heave and swing motion caused by wind, wave, and current.

Key Application

Under the specified sea conditions, all underwater equipment, such as the submarine ore collection and navigation and positioning system, underwater transport system, power transport, and control system, shall be laid out and recovered in accordance with reasonable and orderly lifting, docking, and releasing processes to support and ensure the smooth operation of seabed mineral collection (Yang et al. 2019).

Important Scientific Issues or Technologies

Heave Compensation Technology for Heavy Haul and Long Stroke

In the process of deep-sea mining, the pipeline transportation system weighing hundreds of tons

and thousands of meters long is suspended under the water surface support platform, and the end of the hard pipe is hundreds of meters away from the seabed. The vertical heave movement of long-distance pipeline transportation system will occur under the induced load of complex marine environment, which will affect the operation safety of surface support platform and the stability of slurry transportation.

Overall Integration Technology of Placement and Recovery

The layout and recovery system needs to complete the layout and recovery of many systems or equipment involved in the seabed ore gathering and navigation positioning system, underwater transportation system, and power transmission and control system under the predetermined sea conditions. There are many objects to be laid out, the interface difference is large, and the operation time is long.

Layout and Recovery System Equipment

Layout and Recovery System of Mining Vehicle

The layout and recovery system of mining vehicle mainly realizes the layout and recovery of mining vehicle from the stern deck to the seabed, the smooth water inflow and outflow of the mining vehicle, the tension compensation in the process of mining vehicle lowering or recovery, and the compensation for ship heave movement during landing or leaving the bottom, so as to ensure the safety of mining vehicle (Fig. 2).

Water Surface Support (Layout and Recovery) System, Fig. 2

The layout and recovery of mining vehicle



Water Surface Support (Layout and Recovery) System, Fig. 3 The umbilical winch

automatic operation of low constant tension in the process of laying or recovering each cable, to ensure the safety of the system (Fig. 3).

Pipe Connection and Dismantling Device

The main function of the pipeline docking and dismantling system is to realize the rapid installation and disassembly of thousands of meters long lifting pipe which needs to be put into water during the process of laying or recycling in deep-sea mining (Fig. 4).

Lifting System

The lifting system mainly realizes the lifting and lowering of laying objects, and also takes into account the centralized operation, data information interaction, and control center involved in the process of pipeline processing (docking or dismantling, lifting, or placing) (Fig. 5).

Umbilical Deployment and Recovery System

The umbilical cable distribution and recovery system can realize the synchronous action of the umbilical cable in the process of laying or recovering the mining vehicle from the stern deck to the seabed, and the synchronous action of the umbilical cable in the process of laying or recovering the underwater equipment from the moon pool to the seabed. At the same time, it can realize the

Heave Compensation System

The heave compensation system can provide vertical support force and heave compensation force for the hard pipe through the up and down movement of the oil cylinder piston rod of the oil cylinder system, to control the stress and displacement of the hard pipe, and keep the tension of the hard pipe basically constant under the condition of the vertical movement of the mining ship, to reduce the risk of fatigue damage and dynamic

Water Surface Support (Layout and Recovery) System, Fig. 4 The casing tongs



Water Surface Support (Layout and Recovery) System, Fig. 5 The lifting system



Water Surface Support (Layout and Recovery) System, Fig. 6 The model of heave compensation system

compression of the underwater transportation pipeline system, and keep the pulp transportation relative to the sea bottom height stable, to improve the overall operation safety (Fig. 6).

Steel Structure Tower and Base Platform

The steel structure tower is mainly used to install the crane of the hoisting system and



Water Surface Support (Layout and Recovery) System, Fig. 7 The steel structure tower and base platform

hang the traveling equipment, and assist in the placement and recovery of the equipment. The base platform provides support for steel structure tower, lifting system, and related equipment, and provides installation position and operation space for equipment operation. In the process of lowering and lifting hard pipe, it is used as temporary suspension bearing body of underwater transportation system, and is also a platform for pipeline butt joint and disassembly (Fig. 7).

Pipeline Storage Transfer and Auxiliary Connection System

The pipeline storage and transfer and auxiliary connection system mainly realizes the safe transfer of the hard pipe and lifting pump between the deck storage area and the base plane in the laying and recycling stage, so as to meet the requirements of laying and recycling operation efficiency, and provide reasonable storage space and storage position for the hard pipe and lifting pump (Figs. 8 and 9).



Water Surface Support (Layout and Recovery) System, Fig. 8 The pipe grabbing crane

Water Surface Support Cooperative Control System

The water surface support cooperative control system mainly realizes the function of relay station and mining vehicle-coordinated lowering. In the process of laying or reclaiming operation, “OPC + Ethernet” mode is used for communication, and the tail winch is controlled to lay the mining vehicle, to ensure that the height difference between the mining vehicle and the deployment relay station of moon pool in the ship is maintained, and to realize the coordinated laying process of the relay station and the mining vehicle. At the same time, the real-time status, parameter information, and alarm information of key equipment are collected, processed, and displayed in the process of placing or recycling (Fig. 10).

Pipe Ship Coupling Device

The pipe ship coupling device is used to compensate the heave displacement and angular displacement of the pipeline caused by waves. Its upper end is connected with the base platform, and the lower end is connected with the slurry pipeline (Fig. 11).

Water Surface Support (Layout and Recovery) System, Fig. 9 The power catwalk



Water Surface Support (Layout and Recovery) System, Fig. 10 The main software interface of the water surface collaborative control system

Placing Transfer Device in Underwater Relay Station

The layout and transfer device of underwater relay station mainly meets the requirements of loading

and transferring of underwater relay station in the process of laying and recycling, realizes the end clamping of hose, and provides operable space for manual docking between underwater relay station

and hose; it also takes into account the platform operation conditions such as umbilical cable connection of underwater relay station, signal beacon

of underwater relay station, heave compensation installation, and other platform operation conditions, which can facilitate other manual auxiliary operation process work around the moon pool (Fig. 12).

Hose Placement and Recovery System

The hose laying and recovery system mainly realizes the arrangement of the head end of the hose from the side of the ship and towing to the stern of the ship to connect with the mining vehicle. In the process of hose laying or recycling, the buoyancy block and beacon are bound, fixed or disassembled, and homing. The end of the towing hose is connected with the underwater relay station through the moon pool (Figs. 13 and 14).

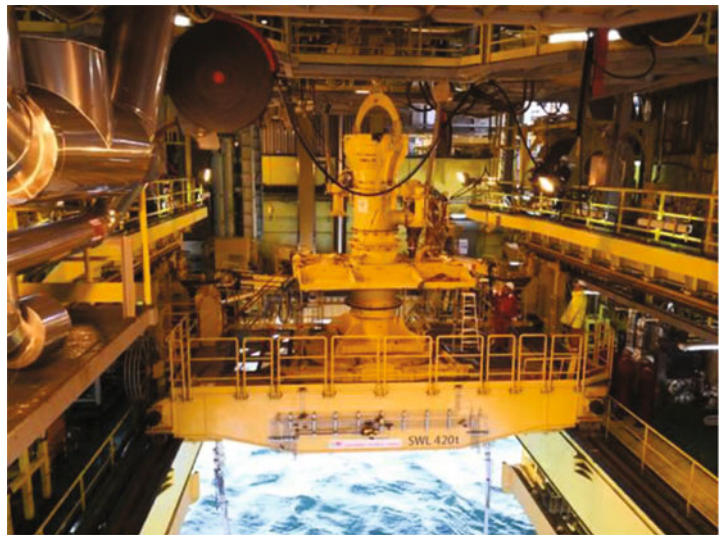
Pipe Clamping Device

As an important bearing part of the water surface support deployment and recovery system, the pipe clamping device is embedded on the working surface of the base platform. In the process of laying and recycling the underwater equipment, the pipe clamping device holds the pipeline to provide support and operation conditions for the pipeline docking and disassembly (Fig. 15).



Water Surface Support (Layout and Recovery) System, Fig. 11 Pipe ship coupling device

Water Surface Support (Layout and Recovery) System, Fig. 12 Placing transfer trolley



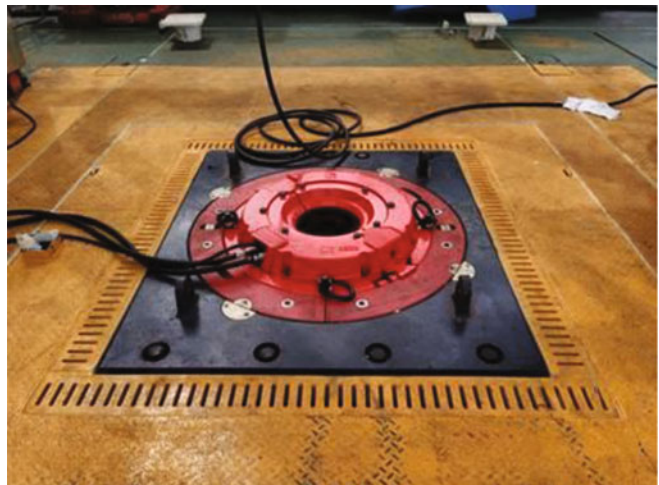
**Water Surface Support
(Layout and Recovery)
System, Fig. 13** The
traction guide device



**Water Surface Support
(Layout and Recovery)
System, Fig. 14** The hose
winch



**Water Surface Support
(Layout and Recovery)
System, Fig. 15** Pipe
clamping device



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Water Surface Support System

► Water Surface Support (Layout and Recovery) System

Wave Energy Converters

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Synonyms

WEC – Wave energy converter

Definition

Wave energy converter is a device or a plant which is set up to convert wave energy into useful energy, such as electricity, or provide power for other purposes, such as seawater desalination, etc.

Scientific Fundamentals

Generally, a wave energy converter consists of two or three main stages in converting wave energy into useful power. In the primary wave energy conversion, with the interaction of the wave and structure(s), the wave energy can be converted into mechanical/pneumatic/potential energy, depending on the types of wave energy

converter, while in the secondary energy conversion, the power take-offs (PTOs, see power take-off) are used to further convert the trapped energy into useful mechanical energy (and finally into electricity in the tertiary conversion stage using the generators).

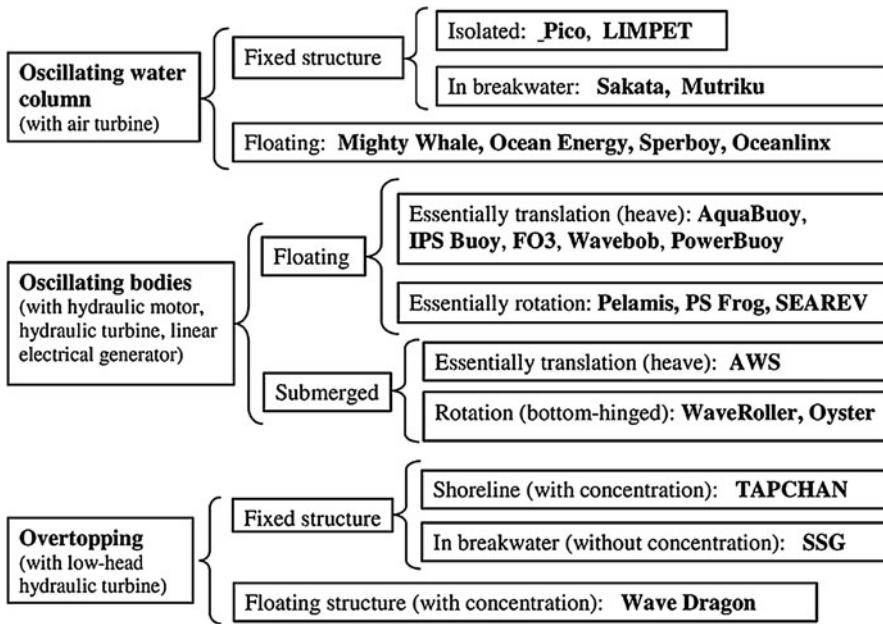
Technology Classification

It has been reported that up to date more than 1000 wave energy technologies have been patented (see Falcao 2010), but in reality only a handful wave energy converting technologies have been effectively proven using numerical and/or experimental studies, and some of them have been advanced to higher technology readiness levels (TRLs), and fewer even reach the pre-commercial sea trials in real seas or generate power to the grid.

Classifying all wave energy technologies has been attempted by different researchers, and several different classifications for wave energy technologies have been proposed, and some good examples can be found in (Falcao 2010; Drew et al. 2009; EMEC 2018). In a rather simple form, Falcao (2010) classifies the wave energy technologies into three main types, oscillating water columns, oscillating bodies, and overtopping devices (see Fig. 1), although this classification may not include all the proposed or developed wave energy technologies. However, it uses a rather simple form to cover the main-streams of the wave energy conversion technologies, mainly based on the types of four power take-offs (PTOs), including (1) air turbines for oscillating water columns; (2) hydraulic systems and (3) direct drives for oscillating bodies; and (4) low head water turbines for overtopping devices. It can be also seen that all these three types of wave energy technologies can be further classified as floating and fixed devices or, in more appropriate definitions, fix-referenced and self-referenced devices (Sheng and Lewis 2016).

Mathematical Modelling of Wave Energy Conversion

Generally, both frequency-domain and time-domain analysis methods of hydrodynamics of wave energy converters can be employed for studying the forces acting on and the motions



Wave Energy Converters, Fig. 1 Categories of wave energy converters (Falcao 2010)

of the wave energy devices in waves, and the modelling would be depending on the requirements or the practical problems. Frequency-domain analysis is fast, but it is only used for linear systems, while time-domain analysis is much more time-consuming and can be used in nonlinear systems. Here only the popular frequency-domain and time-domain methods are briefly discussed.

Frequency-Domain Analysis

Frequency-domain analysis is based on the assumptions of the Laplace equation and the linear boundary conditions in the potential flow. To solve such a linear equation, the superimposition principle can be employed which could significantly simplify the dynamic problems. For instance, for the wave-structure interaction problems, the total velocity potential function flow can be decomposed into diffracted and radiated velocity potentials, and these potential components can be solved independently based on the different hydrodynamic boundary conditions:

- Velocity potential for incoming waves and velocity potential for the scattered waves (due

to the existence of the structure in waves): diffraction problem

- Velocity potential for the radiated waves (due to the motions of the structure): radiation problem

The diffracted problem accounts for the interaction of the structure and wave under the assumption that the structure is fixed in the fluid. When the waves interact with the structure, the structure may reflect and refract the waves due to the existence of the structure. The total waves comprising the incoming waves and the reflected/deflected waves are called the diffracted waves. The solution for the diffracted problem is the excitation forces/moments acting on the structure due to the diffracted waves.

The radiated problem accounts for the case in which the structure moves in an oscillating manner in calm fluid. As such, the structure generates waves which radiate away in all directions (in 3D cases). The solution of the radiated problem is the added mass and potential damping forces and moments due to the radiated waves. The restoring forces and moments due to the motions of the structure will also be obtained.

These two independent problems (the diffracted problem and the radiated problem) form the full hydrodynamic problem for the potential flow thanks to the superimposition principle.

To solve the potential flow problems, the boundary element method (BEM) has been developed in the last century, with strict mathematical derivation (see details in books Newman 1977; Faltinsen 1990; Falnes 2002). Now some commercial and open-source BEM are available (WAMIT 2016; AQWA; Parisella and Gourlay 2016) for both research and industry applications.

In frequency domain, the dynamic equation of motions for the wave energy converter is very similar to that of the conventional floating structures in waves but with an additional force from the power take-off system for wave energy conversion. The generalized frequency-domain dynamic equation for the wave energy converters is given as

$$\{-\omega^2[\mathbf{M} + \mathbf{a}(\omega)] + i\omega \mathbf{b}(\omega) + \mathbf{C}\} \mathbf{x}(\omega) = \mathbf{f}_{\text{ex}}(\omega) + \mathbf{f}_{\text{pto}} + \mathbf{f}_{\text{mooring}} \quad (1)$$

where $\mathbf{M}$ is the mass matrix of the wave energy converter; $\mathbf{C}$ the restoring force coefficient matrix; $\mathbf{a}$ and $\mathbf{b}$ the added mass and the damping coefficient matrix (from the solution of the radiated problem); $\mathbf{x}$ the complex amplitude vector of the motions; $\mathbf{f}_{\text{ex}}$ the excitation vector; $\mathbf{f}_{\text{pto}}$ the power take-off force vector; and $\mathbf{f}_{\text{mooring}}$ the linear or linearized mooring force vector.

The average converted power is calculated as

$$\bar{P}_{\text{reg}} = \frac{1}{2} \omega \text{Re} (\mathbf{f}_{\text{pto}} \cdot \mathbf{x}^*) \quad (2)$$

where the superscript $*$ means a conjugate of the complex vector.

When the motion in Eq. (2) is given as the response (the motion in the wave of a unit amplitude) as the conventional outputs from the BEM software, $\bar{P}_{\text{reg}}$ can be considered as the power response curve (the wave energy conversion in the wave of unit amplitude). Then in irregular waves, the capture power for a given spectrum $S(\omega)$ can be calculated as

$$\bar{P}_{\text{irr}} = 2 \int_0^\infty \bar{P}_{\text{reg}} S(\omega) d\omega \quad (3)$$

Time-Domain Analysis

For practical problems for wave energy converters, nonlinearities may come from very different sources: nonlinear wave excitation loads; nonlinear power take-off system (i.e., the nonlinear PTO); nonlinear mooring force (very common if the motions are large!); control system for improving wave energy conversion (when an active control is applied in wave energy conversion, the dynamics is invariantly nonlinear!); or nonlinear restoring force (due to the changes of the area of the water plane) and other nonlinear forces.

A well-accepted time-domain analysis is based on Cummins equation (Cummins 1962) by including the fluid memory effect, and with the Ogilvie relation (Ogilvie 1964), the frequency-domain results can be transformed into the parameters in the Cummins time-domain equation. Such an approach is often termed as a hybrid frequency-time-domain analysis (Taghipour et al. 2008). A modified Cummins time-domain equation for accommodating wave energy conversion can be given as

$$\begin{aligned} [\mathbf{M} + \mathbf{A}] \cdot \ddot{\mathbf{X}}(t) + \int_{-\infty}^t \mathbf{K}(t - \tau) \cdot \dot{\mathbf{X}}(\tau) d\tau \\ + \mathbf{C} \cdot \mathbf{X}(t) \\ = \mathbf{F}_{\text{ex}}(t) + \mathbf{F}_{\text{pto}}(t) + \mathbf{F}_{\text{mooring}}(t) \\ + \mathbf{F}_{\text{control}}(t) \end{aligned} \quad (4)$$

where $\mathbf{A}$ is the frequency-independent added mass matrix (often given as the added mass at infinite frequency); $\mathbf{K}(t)$ the impulse function matrix due to the radiation; $\mathbf{X}(t)$ the motion vector in time domain; $\mathbf{F}_{\text{ex}}$, $\mathbf{F}_{\text{pto}}$, $\mathbf{F}_{\text{mooring}}$, and $\mathbf{F}_{\text{control}}$ the time-dependent vectors for excitation, the PTO force, the mooring force, and the control force.

The transforms between the frequency-domain results and the time-domain parameters are

$$\begin{cases} \mathbf{K}(t) = \frac{2}{\pi} \int_0^\infty \mathbf{b}(\omega) \cos(\omega t) d\omega \\ \mathbf{A} = \mathbf{a}(\omega) + \frac{1}{\omega} \int_0^\infty \mathbf{K}(\tau) \sin(\omega \tau) d\tau \end{cases} \quad (5)$$

The constant added mass $\mathbf{A}$ is often given as the added mass at infinite frequency:

$$\mathbf{A} = \mathbf{a}(\infty) \quad (6)$$

After solving the time-domain equation, the instantaneous power is calculated as

$$P(t) = \mathbf{F}_{pto}(t) \cdot \dot{\mathbf{X}}(t) \quad (7)$$

And the average power conversion over a period of T is given by

$$\bar{P} = \frac{1}{T} \int_0^T P(t) dt \quad (8)$$

Key Applications

Oscillating Water Column (OWC)

An OWC is a specially designed hollow structure (fixed or floating) with an opening in water and an opening to the air. The partially submerged hollow structure forms a water column and an air chamber, separated by a free surface. When a wave passes the hollow structure, the free surface in the hollow structure moves up and down, such that it can compress or decompress the air in the air chamber if the opening to the air is equipped with a power take-off (PTO) system. In a practical OWC, an air turbine can be used to damp the airflow through the upper opening to create the compressed and decompressed air in the air chamber in such a manner that the wave energy is converted into pneumatic energy. The changing pressure in the air chamber could exhale or inhale an airflow through the air turbine PTO and drive the air turbine PTO in a relatively high rotational speed to convert the pneumatic energy into mechanical energy.

As a unique and maybe the largest advantage for the OWC devices as pointed out by Evans and Porter (1995), they may be the only wave energy converters which can effectively overcome the challenges for converting the low-frequency motions in waves (~ 0.1 Hz) into electricity of 50 or 60 Hz. This unique feature allows the PTOs of a high rotational speed to be connected to the generators directly to generate electricity.

Different OWC energy converters have been proposed and studied after the successful OWC navigation buoys with an aim to increase the OWC energy conversion efficiency and capacity for possibly massive wave energy production. These include “Kaimei” (Masuda 1979), Backward Bent Duct Buoy (BBDB) (Masuda et al. 1988) (which is being developed by OceanEnergy Ltd (2017) and currently is in preparation for pre-commercial open sea trials in Hawaii), Spar OWC (Falcao et al. 2012), sloped OWC (Payne et al. 2008), enclosed OWC (Fonseca and Pessoa 2013), as well as the shoreline OWC plants: LIMPET (in decommissioning after generating power to grid for more than 10 years) (Folley et al. 2006), PICO (in decommissioning) (Le Crom et al. 2009; Pecher et al. 2011), and Mutriku (still operational) (EVE 2013). As another successful story in wave energy utilization/production, the LIMPET OWC plant (Scotland, UK), a shoreline OWC wave energy plant, has generated wave power to grid for more than 60,000 h in 2000–2011 (Heath 2012), in a period of slightly over 10 years, and in the last 4 years in its service, the power generation availability reaches 90% (EC 2017). As a pioneer (experimental) wave energy plant, this is a great achievement for the special wave energy technology.

Oscillating Bodies

This category covers a large range of wave energy conversion technologies, including the conventional point absorbers, attenuators, oscillating surging converters, and many other wave energy converters if their body motions are used for wave energy conversion.

Point Absorbers

Point absorbers seem the simplest wave energy converters, which have attracted a lot of attention from researchers and developers, as well as outsiders. The principle of wave energy conversion for point absorbers is very simple and very easy to understand even for people who have no much knowledge about wave energy conversion when he/she watches the up-and-down movements of

the navigation buoys under the wave action in harbors.

Generally, either hydraulic or direct drive power take-offs or other mechanical PTOs can be used to convert the kinetic energy of the device body (bodies) of point absorbers. The well-advanced point absorber technologies include CorPower (fix-referenced point absorber, with built-in control system in the device; see Fig. 2, (CorPower 2017)); CETO (fix-referenced fully submerged point absorber (Rafiee and Fievez 2015)); Wavebob (self-referenced floating point absorber (WaveBob 2011)); OPT (self-referenced floating point absorber (OPT 2015)) and its relevant reference model 3 (RM3 (Yu et al. 2015)); etc.

Attenuators

Pelamis (Yemm et al. 2012), a famous attenuator wave energy converter, may be regarded as the first wave energy converter to reach the pre-commercial level of development, and even three commercial prototypes were manufactured and deployed in the Portuguese coast in 2009 (but no power has been generated yet) based on the report (EC 2017). The later version (P2) has been

tested in EMEC with the financial support from Eon and Scottish Renewables. The company had finally run into administration in 2014, and Wave Energy Scotland (WES) has bought all IPs from the company (EC 2017).

A slightly different form from Pelamis, the raft-type attenuator wave energy converters are now being developed by some organizations, including M4M developed at the University of Manchester, UK, (M4M 2015) and SeaPower in Ireland (SeaPower 2015). Numerical modelling technologies have been widely applied for the attenuator wave energy converters (Zheng et al. 2015) (Fig. 3).

Oscillating Surging Services

The Oyster device is an oscillating surging device, developed by Aquamarine Power (Whittaker and Folley 2012). Such an oscillating surging device is reported to have a high energy conversion efficiency (Babarit et al 2012), and an additional benefit could be a possible low initial building cost of the device since it is basically regarded in a 2D extension, rather than a 3D extension like many other devices if a larger device is expected. A significant challenge is the structural reliability



Wave Energy Converters, Fig. 2 CorPower point absorber wave energy converters (CorPower 2017)



Wave Energy Converters, Fig. 3 Pelamis 2 in waves. (From EMEC website: <http://www.emec.org.uk/about-us/wave-clients/pelamis-wave-power/>)

Wave Energy Converters,

Fig. 4 Oyster device in waves. (Adopted from website: <https://oregonshores.org/article/rules-ocean-energy-siting-are-horizon>)



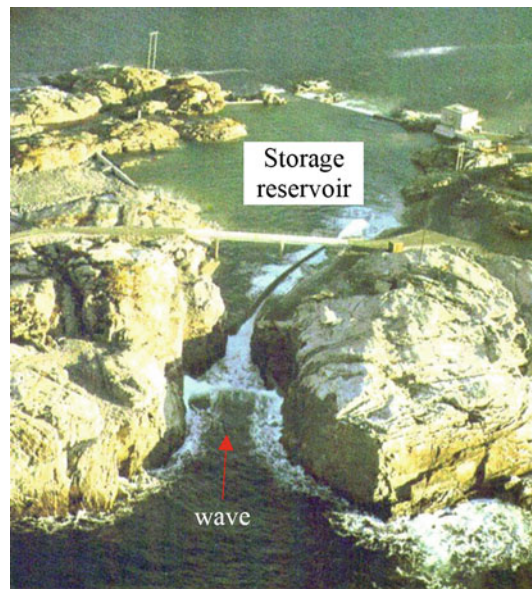
of such a device, since it is found that the device is frequently plunged into water ahead the wave action (see Fig. 4), indicating large wave slamming forces on the device.

Overtopping Devices

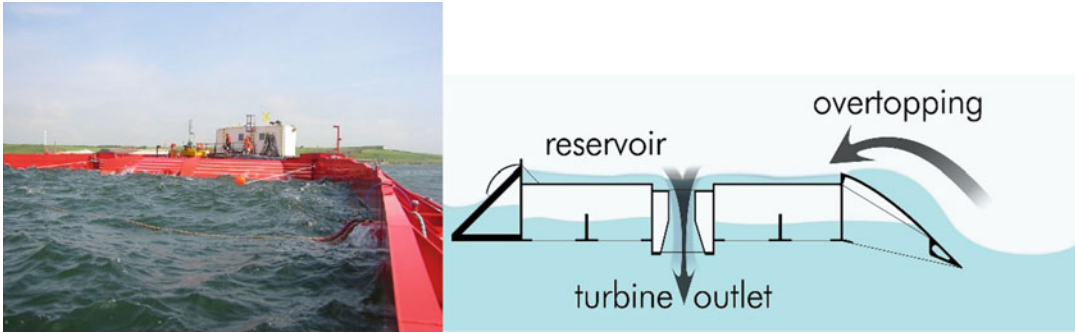
Tapchan

Tapchan is a fixed overtopping wave energy plant, one of the two pioneer wave energy plants in Norway in the 1980s (in the same period, a fixed OWC plant was built a few hundred meters away, both on the west coast of Norway). Accordingly, this prototype wave energy converter has the following features:

- The wave energy is converted to potential energy in an onshore water reservoir.
- The generation of electricity is carried out by a standard hydroelectric power plant technology.



Wave Energy Converters, Fig. 5 Tapchan wave energy plant. (Adopted from (Falcao 2014))



Wave Energy Converters, Fig. 6 Wave Dragon 1:4.5 model tested in sea. (Adopted from the Internet)

- The conversion device is entirely passive and has no moving parts (except the water turbine).
- High energy conversion efficiency can be maintained over a broad range of wave heights, frequencies, and directions.

The plant was destroyed by a winter storm in 1988 (see https://www.youtube.com/watch?v=vG6R_R2YyAo) (Figs. 5 and 6).

Floating Overtopping WECs

Wave Dragon is a floating overtopping wave energy converter. From the literature, this device may have so far the highest power rating in all single wave energy converters: 12MW (<http://www.wavedragon.net/>). A proposed full-scale Wave Dragon with a 7MW rated power is a huge device, with an overall length of 350m between two arms and a displacement of 40,000 m³. In the development, the concept has finished a 1:4.5 scale sea trial in the test site in Nissum Bredning, Denmark (Kofoed et al. 2006).

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Wave Energy Utilization Buoy

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Synonyms

Wave energy converters; WEC

Definition

Wave energy utilization buoy is a floating power station for generating electrical energy from wave power. The buoy devices being developed include: Powerbuoy, OE buoy, RM3, IPS buoy, and so on. (The examples shown here are not intended to form an exhaustive list and were chosen among the projects that reached the prototype

stage or at least were object of extensive development effort.)

Scientific Fundamentals

A Brief Overview of Buoy Wave Energy Device

Wave energy is a type of well-concentrated renewable energy when compared to other types of renewable energy (such as solar, wind, etc.). The wave energy resources are huge, utilizing wave energy is desirable in many countries around the world, especially in the countries with long coastal lines and rich wave energy resources.

Wave energy researchers and developers have been working in solving the practical problems in wave energy extraction for many years. Though some early optimistic predictions in capturing large wave energy from seas at a competitive rate (for example, 2GW wave power plant producing power at a rate of 1.3 p/kWh has been described in Whittaker et al. 1985) have not been reached, the research and development of wave energy conversion have made a great deal of progress, and some of the wave energy devices have reached the stages of full-scale pre-commercial or large-scale sea trials. For instance, in 2009, a 1:2.5 scale OE Buoy device (an oscillating water column (OWC) device) finished a sea trial for a period of over two and half years in Galway Bay. (Ocean Energy Ltd. 2018) Recently, OceanEnergy announced its pioneering wave energy convertor OE Buoy will be built by Oregon-based Vigor and deployed at the US Navy's Wave Energy Test Site on the windward coast of the Hawaiian Island of O'ahu in the fall of 2018, the 826-ton OE Buoy measures 125×59 feet with a draft of 31 feet and has a potential rated capacity of up to 1.25 MW in electrical power production (Wave Energy Conversion Pioneer 2018). More wave energy devices and their developments can be found in Refs. (Falcão 2010; ISSC Specialist Committee 2009)

In the development of wave energy, nearshore or shallow water regions are frequently considered for developing wave energy converters/farms due to the closeness to the shore and the easy

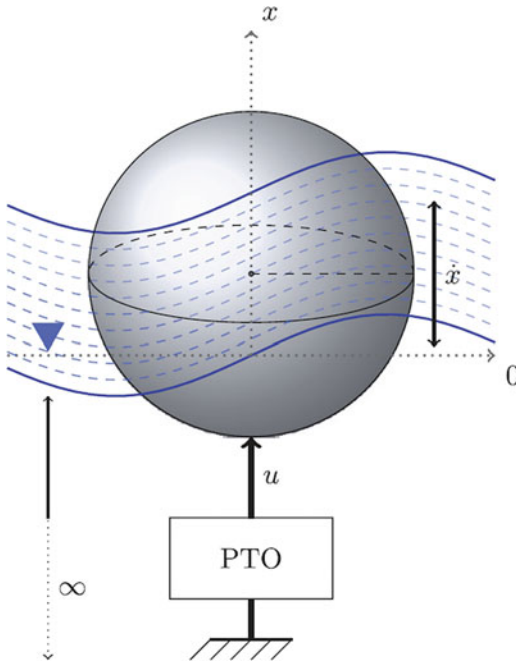
infrastructure, for instance, the cable connection and the availability for the operation and maintenance. One important aspect of these developments is the availability of wave energy resources in shallow water regions. Folley and Whittaker (2009) have shown that the wave resources reduce by less than 10% when traveling from a water depth of 50 m to 10 m. Hence the development of wave energy in shallow water regions has been preferred. So far most of the installed wave energy converters are deployed in the water depths of less than 50 m, and most of the proposed wave farms are deployed in the water depths of less than 100 m. Similarly, Johanning et al. (2007) concluded that wave energy converters (WECs) would be very likely installed in the shallow to intermediate depths, typically at around 50 m contour in the open areas for wave energy production.

Point absorbers may be the simplest floating buoy wave energy converters, and the main motion mode for power conversation is heave, though other motion modes may contribute to small part of the power conversation. The horizontal dimensions of point absorbers are much smaller than the wavelength, the known devices are Power Buoy, (OPT 2018a) the US reference model (RM3, Yu et al. 2015; Prendergast et al. 2018) and RM6, (Bull et al. 2014; Sheng 2019a), and so on. The main motions of point absorbers are the relative heave motions between buoy and referenced damping panel (or referenced to the sea bed). It is quite straightforward for the cases with only considering the relative heave motions (Falnes 1999a and Sheng and Lewis 2016a).

Scientific Fundamentals

The study of the hydrodynamics of floating wave energy converters could benefit from previous studies on the, largely similar, dynamics of ships in wavy seas, that took place in the decades preceding the mid-1970s. The presence of a power take-off mechanism (PTO) and the requirement of maximizing the extracted energy introduced additional issues.

In this section, we will take the single buoy wave energy converter as an example to illustrate how the device extracts wave energy from regular



Wave Energy Utilization Buoy, Fig. 1 Wave energy converter. Figure from (Faedo et al. 2017)

(sinusoidal) waves and converts it into electrical energy. Detailed information on wave energy utilization can be found in the literature (Falcao 2010). The oscillation system consists of a half-immersed sphere and a linear PTO with one end anchored in the sea bed, as shown in Fig. 1.

Theoretical and Numerical Modeling

We assume the waves with small amplitude, which allowed the linearization of the governing equations and the use of the frequency-domain analysis. The hydrodynamic forces on the wetted surface of the body were decomposed into excitation forces (due to the incident waves), radiation forces (due to body motion), and hydrostatic forces (connected with the instantaneous position of the floating body with respect to the undisturbed free surface). As shown in Fig. 1, the body position is defined by the vertical coordinate x (with $x = 0$ in calm water), the equation of motion is

$$(m + A)\ddot{x} = f_d - B\dot{x} - \rho g S x + f_{PTO} \quad (1)$$

where, $f_d(t)$ is the vertical component of the excitation force (acting on the assumedly fixed body,

$f_d = 0$ in calm water), $f_{PTO}(t)$ is the vertical force due to the PTO mechanism, $A(\omega)$ is the hydrodynamic coefficient of added mass (accounting for the inertia of the water surrounding the body), $B(\omega)$ is the radiation damping coefficient (accounting for the damping on the body due to energy transfer to waves radiated away), and S is the cross-sectional area of the body by the unperturbed free surface plane ($\rho g S x$ is the hydrostatic restoring force).

We assume the PTO force to consist of a linear damper (coefficient C) and a linear spring (stiffness K):

$$f_{PTO} = -C\dot{x} - Kx \quad (2)$$

Then the whole system becomes fully linear.

In regular waves of amplitude A_ω and frequency ω , we write $\{x, f_d\} = \text{Re}(\{X, F_d\}e^{i\omega t})$, where X and F_d are complex amplitudes and $\text{Re}()$ means real part of. Then, from eq. 1, we obtain

$$X = \frac{F_d}{-\omega^2(m + A) + i\omega(B + C) + \rho g S + K} \quad (3)$$

Since the system is linear, the excitation force is proportional to wave amplitude, i.e., $|F_d| = \Gamma A_\omega$, where $\Gamma(\omega)$ is a hydrodynamic coefficient of diffraction force. The time-averaged absorbed power is $\bar{P} = \overline{f_d \dot{x}} = C\omega^2 |X|^2 / 2$, which can be written as

$$\bar{P} = \frac{1}{8B} |F_d|^2 - \frac{B}{2} \left| U - \frac{F_d}{2B} \right|^2 \quad (4)$$

where $U = i\omega X$ is the complex amplitude of the velocity $\dot{x}$.

For a given body and given incident regular wave, B and F_d are fixed. Then the absorbed power $\bar{P}$ depends on X , i.e., on the PTO damping and spring coefficients C and K . The maximum value equal to

$$\bar{P}_{max} = \frac{1}{8B} |F_d|^2 \quad (5)$$

occurs for $U = F_d/2B$, then combined Eq. 3 we obtain the two optimal conditions,

$$\omega = \left(\frac{\rho g S + K}{m + A(\omega)} \right)^{1/2} \quad (6)$$

$$C = B(\omega) \quad (7)$$

Equation 6 is a resonance condition: its right-hand side is the frequency of free oscillations of an undamped mechanical oscillator of mass $m + A$ acted upon by a spring of stiffness $\rho g S + K$. Equation 7 shows that the optimal PTO damping should equal the radiation damping.

In regular waves, the averaged wave energy density, P_w (per unit surface area) over a wave period is given by

$$P_w = \frac{1}{2} \rho g A^2 \quad (8)$$

where A is the amplitude of regular waves. We define the capture width $l = \bar{P}/P_w$, for a body with a vertical axis of symmetry (but otherwise arbitrary geometry) oscillating in heave, it can be shown that

$$l_{max} = \frac{\bar{P}_{max}}{P_w} = \frac{\lambda}{2\pi} \quad (9)$$

where λ is the wavelength. If the body oscillates in sway $l_{max} = \frac{\lambda}{\pi}$ equation 9 is an important theoretical result, obtained independently in 1975–1976 by Budal and Falnes, Evans, Newman, and Mei, (Falcao 2010) on the maximum power that can be absorbed from the waves, as is the well-known Betz limit for the power coefficient of wind turbines.

Maximum Wave-Power Absorption Under Motion Constraints

In practice, most wave-energy devices will have physical limitations placed upon their excursions due to restraints such as mooring lines or piston stroke. Whereas there exists a significant band of wave frequencies and lengths for which the device does not reach these limits, in long waves a relatively small device would, ideally, need to make large excursions to achieve optimum power absorption. Again, larger amplitude waves require larger device excursions for optimum absorption. Evans D V (1981) proposed a simple expression

for the maximum efficiency of power absorption of a buoy when its velocity and hence amplitude is constrained such that its magnitude never exceeds a given multiple of the incident wave amplitude and also for a system of absorbing bodies under global velocity constraints. Here we will briefly demonstrate his conclusion.

We consider a regular incident wave condition in a frequency of ω , N independently oscillating buoys making simple harmonic motions response to the wave also in the frequency of ω , and each buoy can absorb energy from the incident wave field. Let $\mathbf{U}$ be the complex N -vector denoting the amplitude and phase of the velocity of each buoy, and $\mathbf{B}$ the $N \times N$ real, symmetric radiation damping matrix. Let $\mathbf{F}$ be the complex N -vector denoting the amplitude and phase of the exciting force on each buoy in its direction of subsequent motion, when all buoys are held fixed. Then the optimized energy absorption can be shown as:

$$\begin{aligned} P_{max}(\mathbf{U}) &= \frac{1}{2} \mathbf{U}^* (\mathbf{B} + 2\mu \mathbf{I}) \mathbf{U} \\ &= \frac{1}{8} \mathbf{F}_s^* \left[\mathbf{B}^{-1} - \mathbf{B}^{-1} (\mathbf{B} + \mu \mathbf{I})^{-2} \mu^2 \right] \mathbf{F}_s \end{aligned} \quad (10)$$

for

$$\beta \leq \frac{1}{2} \|\mathbf{B}^{-1} \mathbf{F}_s\| = \|\mathbf{Y}\| \quad (11)$$

where $*$ denotes conjugate transpose, β is the constraints of the buoys, μ determined by $\mathbf{U}^* \mathbf{U} = \beta^2$ and $\mathbf{U} = (\mathbf{B} + \mu \mathbf{I})^{-2} \mathbf{B} \mathbf{Y}$, $\mathbf{Y} = \frac{1}{2} \mathbf{B}^{-1} \mathbf{F}$.

We assume each of the N buoys have constraints on their amplitudes and velocities which may be of the form:

$$|U_i| \leq \omega A \alpha_i = \beta_i \quad (i = 1, 2, \dots, N) \quad (12)$$

where A is the incident wave amplitude and α_i are constants governing the permissible buoy amplitude compared to the incident wave amplitude. In normal operating conditions $\alpha_i = 0(1)$ but in larger waves α_i may be small.

We define the capture width $l = P/P_w$, for a vertically oscillating buoy with a vertical axis of symmetry,

$$l_{max} \equiv \frac{P_{max}}{P_w} = \frac{\lambda}{2\pi} \left\{ 1 - H(1 - \delta)(1 - \delta)^2 \right\} \quad (13)$$

where

$$\delta = \frac{2\omega A \alpha B}{|F|} = \alpha \left(\frac{2M}{\rho a^3} \right)^{\frac{1}{2}} (Ka)^{\frac{3}{2}} \lambda_{33}^{\frac{1}{2}} \quad (14)$$

where K is the wave number, ρ is the density of sea water, a is the radius of the buoy, and λ is the damping coefficient in heave, nondimensionalized with respect to a typical mass, M .

Figure 2 Evans 1981 shows the variation of $l_{max}/2a$ with Ka for different α_i of a half-immersed sphere. It can be seen that, if the amplitude of motion of the sphere is not allowed to exceed the wave amplitude the capture width never exceeds 70% of the sphere diameter in contrast to the result from unconstrained motions where l_{max} increases indefinitely as $Ka \rightarrow 0$. For sphere amplitudes up to twice the wave amplitude ($\alpha = 2$), l_{max} peaks at a value 108% of the sphere diameter showing that energy is being absorbed from outside the region

occupied by the sphere. In steeper waves where the ratio may have to be as small as $\alpha = 0.5$ the captured width never exceeds 40% of the sphere diameter.

Open Sources for WEC Simulation

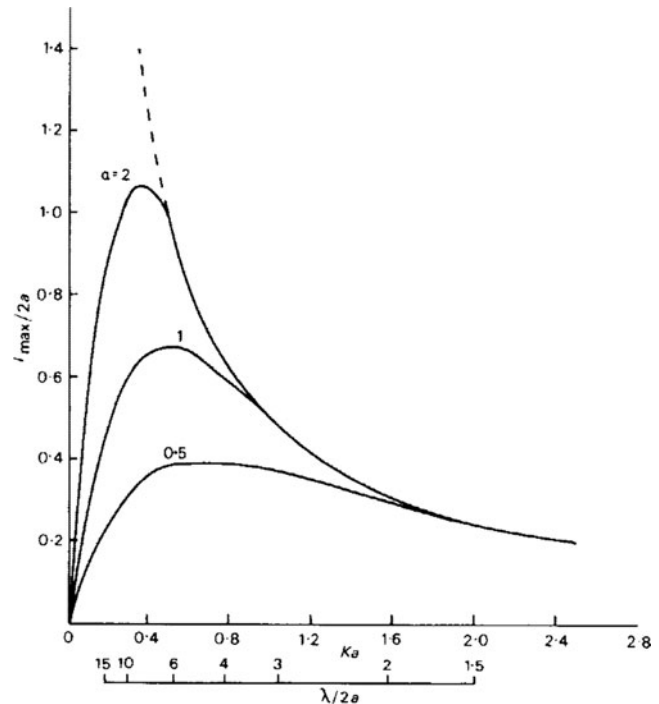
WEC-Sim (Wave Energy Converter SIMulator) is an open-source wave energy converter simulation tool. The code is developed in MATLAB/SIMULINK using the multibody dynamics solver Simscape Multibody. WEC-Sim has the ability to model devices that are comprised of rigid bodies, power-take-off systems, and mooring systems. Simulations are performed in the time-domain by solving the governing WEC equations of motion in 6 degrees of freedom.

WEC-Sim is a collaboration between the National Renewable Energy Laboratory (NREL) and Sandia National Laboratories (Sandia), funded by the US Department of Energy's Water Power Technologies Office. Due to the open-source nature of the code, WEC-Sim has also had many contributions from external

Wave Energy

Utilization Buoy,

Fig. 2 Variation of maximum capture width/diameter ratio with wave length for half immersed energy absorbing sphere for different motion constraints. The dashed line denotes unconstrained sphere (Evans 1981)



collaborators. For more details see (Wave Energy Converter SIMulator 2018).

Key Applications

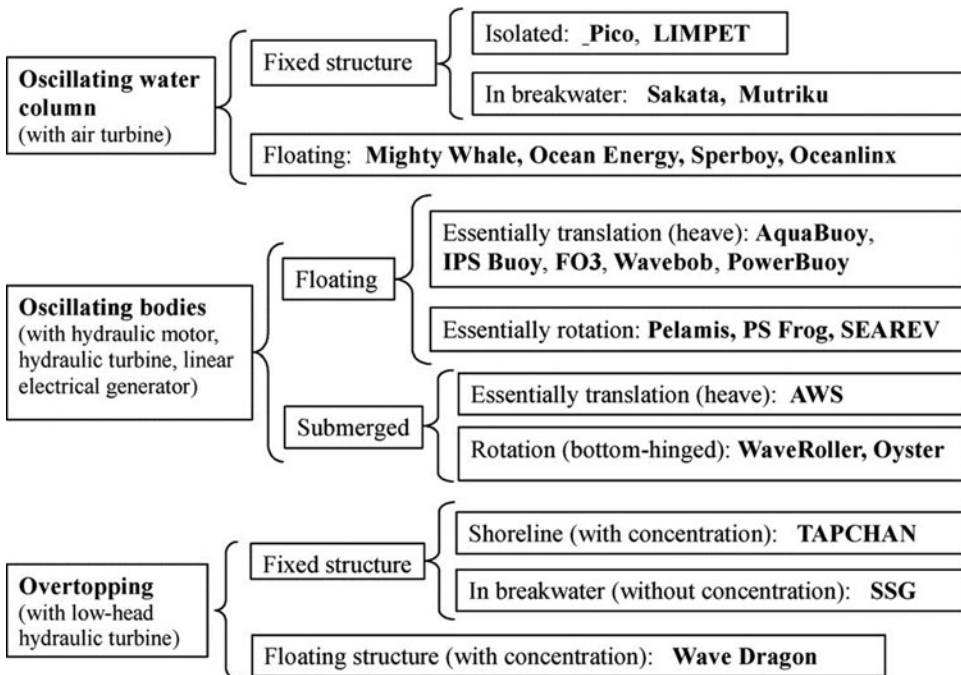
There is a wide variety of wave energy utilization technologies, resulting from the different ways in which energy can be absorbed from the waves, and also depending on the water depth and on the location (shoreline, nearshore, offshore). Several methods have been proposed to classify wave energy systems, according to location, to working principle, and to size (“point absorbers” versus “large” absorbers).

The classification in Fig. 3 (Falcao 2010) is based mostly on working principle. The floating devices can be classified as wave energy utilization buoy. The examples shown are not intended to form an exhaustive list and were chosen among the projects that reached the prototype stage or at least were object of extensive development effort.

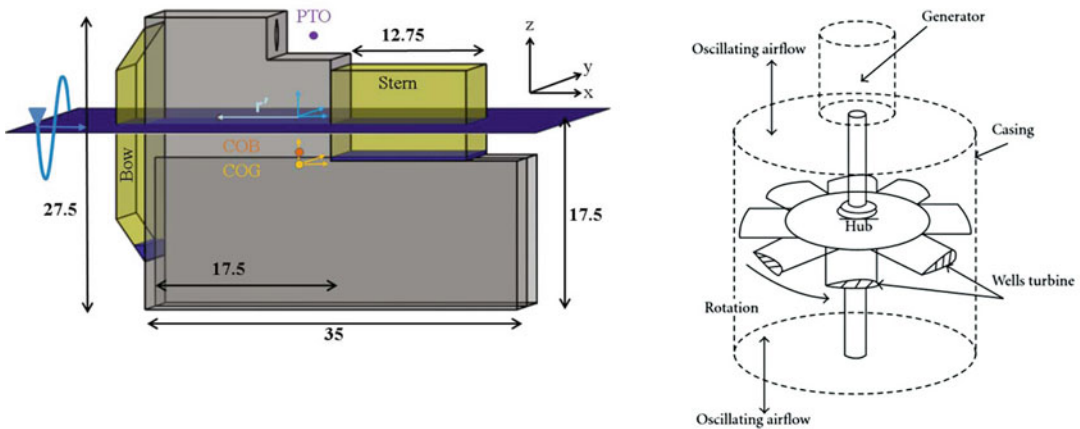
A few devices are briefly described in the following section.

OE Buoy (BBDB OWC, RM6)

OE Buoy is an Oscillating Water Column wave energy converter deployed by the HMRC (The Hydraulics and Maritime Research Centre) in Cork, Ireland, and is now owned and developed by the company OceanEnergy. The OE Buoy is a version of a device known as the Backward Bent Duct Buoy (BBDB) which was invented in 1986 by wave energy pioneer and Japanese naval commander Yoshio Masuda et al. 1988. It was deployed in half-scale test mode in Spiddal near Galway in Ireland for over 2 years between 2007 and 2009. As of March 5, 2011, the model has been redeployed at the same site, primarily as a data collector for the EU-funded Cores Project. The full-scale pioneering OE buoy developed by OceanEnergy will be built by Oregon-based Vigor and deployed at the US Navy’s Wave Energy Test Site on the windward coast of the Hawaiian Island



Wave Energy Utilization Buoy, Fig. 3 The various wave energy technologies (Falcao 2010). The floating devices can be classified as wave energy utilization buoy



Wave Energy Utilization Buoy, Fig. 4 RM6 BBDB device design and dimensions and wells turbine schematic. The width of the device (not shown) is 27 m (Bull et al. 2014)

of O'ahu in the fall of 2018 (Wave Energy Conversion Pioneer 2018).

RM6 reference model (see Fig. 4) is an oscillating water column wave energy converter developed by Sandia National Laboratories using a combination of numerical modeling tools and scaled physical models (Bull et al. 2014). Bull et al. (2015) studied the US RM6 BBDB oscillating water column wave energy converter, providing some insights on the BBDB wave energy converters on the optimization of the air turbine PTO, as well as the electricity generation. The systematic study of the power performance of BBDB OWC devices can be found in the literature (Sheng and Lewis 2018, 2016b; Sheng 2019b).

Reference Model 3 RM3 is essentially a two-body floating (self-referenced) point absorber, it can also be considered as a fix-referenced PA when the damping panel is fixed (no motion is allowed for it). The following section will take RM3 as an example to illustrate the basic principle of wave energy device conversion of wave energy.

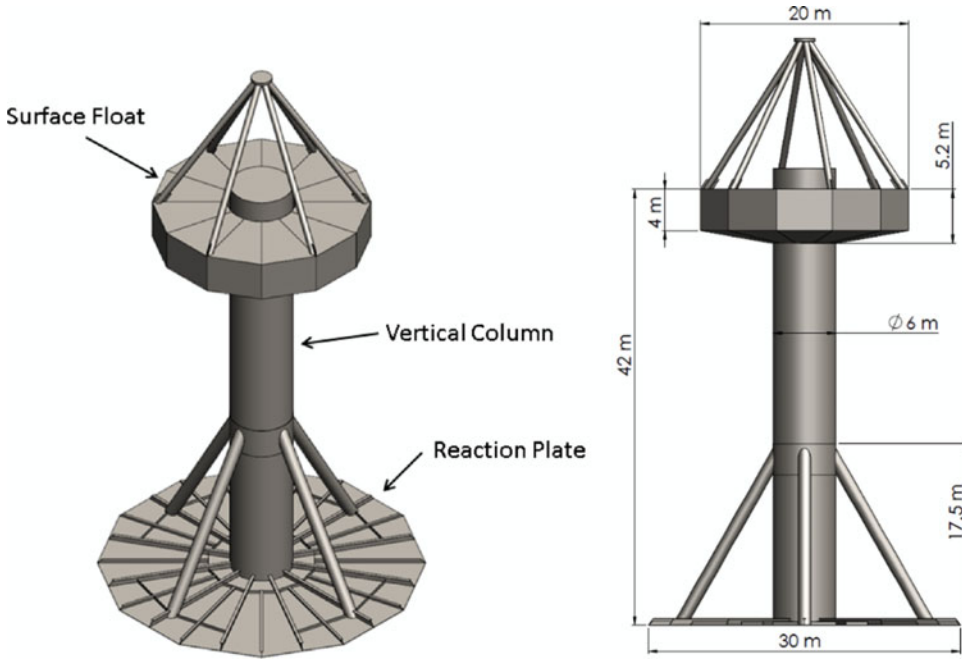
The original design of RM3 is shown in Fig. 5. This point absorber consists of two main moving bodies: the surface float of diameter 20 m with a small draft, and the vertical column which is connected to a large reaction plate (30 m in diameter). Overall, the point absorber has a draft of

30 m, and the displacements 848.2 m^3 and 680.8 m^3 for the vertical column (the cylinder and the reaction plate has been simplified as zero thickness) and for the surface float, respectively.

The surface float moves up and down the vertical column in response to the motion of the waves. The reaction plate maintains the vertical column in a relatively stationary position. The relative motion of the surface float with respect to the vertical column drives a mechanical system contained in the vertical column that converts the linear motion of the surface float into a rotary one. The rotary motion drives electrical generators that produce electricity for the payload or for export to nearby marine applications using a submarine electrical cable. This high-performance wave energy conversion system generates power even in moderate wave environments.

In converting wave energy, the surface float will very much move in-phase, while the vertical column moves out-of-phase with the passing waves. These in-phase and out-of-phase motions could generate a large relative motion between these two bodies, thus providing good motions for converting wave energy. In the case of a fix-referenced RM3, the vertical column can be taken as a fixed structure.

It will be seen that for a floating PA with its relative motions between the moving components being used, it could be beneficial for improving its wave energy conversion efficiency, as well as its



Wave Energy Utilization Buoy, Fig. 5 The RM3 point absorber wave energy converter (Yu et al. 2015).

compliance thus its survivability in the extreme waves (with long wave periods).

Following (Sheng and Lewis 2016a), for such floating/self-referenced WECs, the heave motions of the two-body systems are normally

independent of the other motion modes, especially when the motions are small. For power conversion, the relative heave motions of the two bodies are taken, and the corresponding dynamic equation can be expressed as:

$$\begin{cases} [i\omega Q_{33} + b_{33} + B_{33}]v_3 + [i\omega Q_{39} + b_{39}]v_9 = f_3 - B_{pto}(v_3 - v_9) \\ [i\omega Q_{93} + b_{93}]v_3 + [i\omega Q_{99} + b_{99} + B_{99}]v_9 = f_9 + B_{pto}(v_3 - v_9) \end{cases} \quad (15)$$

with

$$\begin{cases} Q_{33} = (m_{33} + a_{33}) - \frac{c_{33}}{\omega^2}; Q_{39} = a_{39} - \frac{c_{39}}{\omega^2} \\ Q_{93} = a_{93} - \frac{c_{93}}{\omega^2}; Q_{99} = (m_{99} + a_{99}) - \frac{c_{99}}{\omega^2} \end{cases} \quad (16)$$

where B_{pto} is the damping coefficient of the PTO for the RM3 device; b_{33} , b_{39} , b_{93} , and b_{99} are the radiation damping coefficients due to the body motions; B_{33} and B_{99} the additional linear damping coefficients caused by the heave motion

of the bodies; f_3 and f_9 the complex excitations for the heave motions of both moving bodies of the device; v_3 and v_9 the velocities of the heave motions; m_{33} and m_{99} the mass of each body; a_{33} , a_{39} , a_{93} , and a_{99} the self- and cross-term added masses related to the heave motion of two bodies; and c_{33} , c_{39} , c_{93} , and c_{99} the corresponding restoring force coefficients for the device.

The average power conversion is then given by Eq. 17 below,

$$\overline{P} = \frac{1}{2} B_{pto} |v_9 - v_3|^2 \quad (17)$$

with the relative velocity solved from Eq. 15, the average power is,

$$\overline{P} = \frac{1}{2} B_{pto} \frac{Z_1^2 + Z_2^2}{(Y_1 + Y_2 B_{pto})^2 + \omega^2 (Y_3 + Y_4 B_{pto})^2} \quad (18)$$

In the above equation Y_1 , Y_2 , Y_3 , and Y_4 are

$$\begin{cases} Z_1 = f_{3r}(b_{99} + B_{99} + b_{93}) - f_{9r}(b_{33} + B_{33} + b_{39}) - \omega f_{3i}(Q_{99} + Q_{93}) + \omega f_{9i}(Q_{33} + Q_{39}) \\ Z_2 = f_{3i}(b_{99} + B_{99} + b_{93}) - f_{9i}(b_{33} + B_{33} + b_{39}) + \omega f_{3r}(Q_{99} + Q_{93}) - \omega f_{9r}(Q_{33} + Q_{39}) \\ Y_1 = (b_{33} + B_{33})(b_{99} + B_{99}) - b_{93}b_{39} - \omega^2(Q_{33}Q_{99} - Q_{93}Q_{39}) \\ Y_2 = b_{33} + B_{33} + b_{99} + B_{99} + b_{39} + b_{93} \\ Y_3 = (b_{33} + B_{33})Q_{99} + (b_{99} + B_{99})Q_{33} - b_{93}Q_{39} - b_{39}Q_{93} \\ Y_4 = Q_{33} + Q_{99} + Q_{39} + Q_{93} \end{cases} \quad (19)$$

where the subscripts r and i denote the real and imaginary parts of the excitation forces f_3 and f_9 .

Based on the Eq. 18, an optimized PTO damping for maximizing the wave energy conversion for the wave of a given frequency can be obtained in Eq. 20 and the corresponding maximized power conversion in Eq. 21.

$$B_{pto} = \sqrt{\frac{X_1^2 + \omega^2 Y_1^2}{X_2^2 + \omega^2 Y_2^2}} \quad (20)$$

$$\overline{P}_{max} = \frac{1}{4} \frac{Z_1^2 + Z_2^2}{X_1 X_2 + \omega^2 Y_1 Y_2 + \sqrt{(X_1^2 + \omega^2 Y_1^2)(X_2^2 + \omega^2 Y_2^2)}} \quad (21)$$

The optimized PTO damping and the maximal converted wave energy both vary depending on frequency; hence this optimized condition may only be useful in the regular waves, where the wave frequencies are constant for a given wave. Figure 6 shows the comparison of the maximal capture powers for the fix-referenced and self-referenced PAs (“fix-referenced” and “self-referenced” in the legend) and the self-referenced devices are with two different reaction plates ($D = 25$ m and $D = 30$ m), more details can be found in Sheng and Lewis (2016a).

PowerBuoy

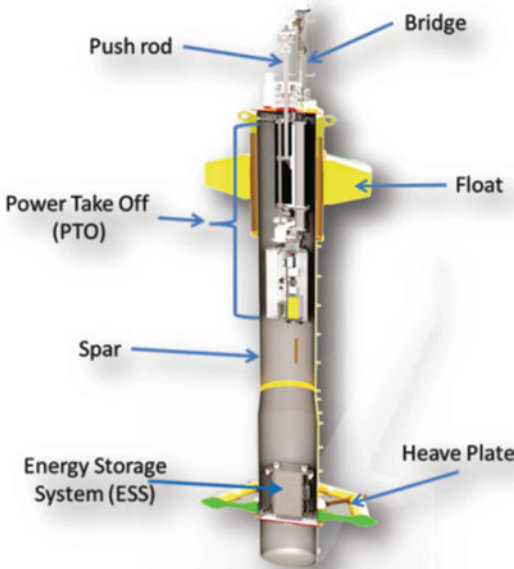
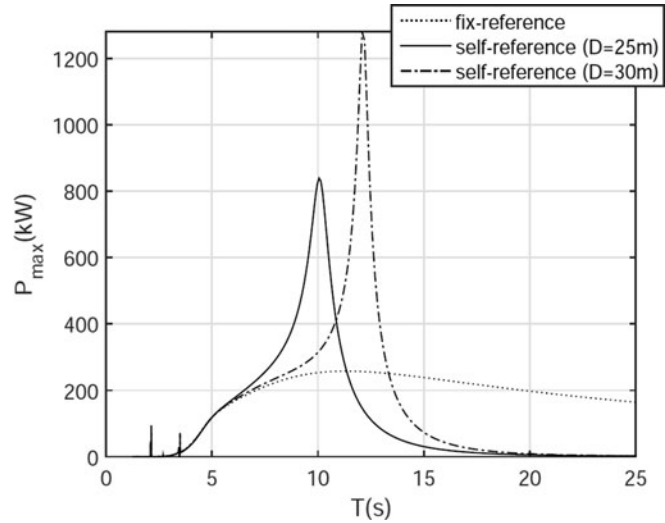
PowerBuoys are manufactured by Ocean Power Technologies (OPT) in Pennington, New Jersey

reactance-dependent damping coefficients and Z_1 and Z_2 are radiation damping and excitation-based values that are calculated using Eq. 19 as:

(OPT 2018b) (see Figs. 7 and 8). The outline of the PowerBuoy design process is illustrated by examples from recent development efforts in (Edwards and Mekhiche 2014), which attempts to address the complex multidiscipline and concurrent engineering process required to successfully design, develop, test, and validate of OPT’s PowerBuoy. OPT’s PowerBuoy portfolio includes two power output ranges: up to 3 kW and up to 15 kW, the average output power is deployment-site dependent. OPT’s products serving the off-shore power needs are the PB3 (commercial ready) and the PB15 (under development). Product specifications for the PB3 are shown in Fig. 9. OPT’s PB3 was designed acting as an uninterruptable power supply (UPS) which

Wave Energy Utilization Buoy,

Fig. 6 RM3 maximal capture powers in regular waves ($D = 25$ m and $D = 30$ m) (Bull et al. 2014).



Wave Energy Utilization Buoy, Fig. 7 PowerBuoy cross sectional view (Parsa et al. 2017)

constantly recharges itself by harvesting energy from the waves. It was developed for offshore remote autonomous applications such as the ones found in the oil and gas industry. It was ocean-deployed, moored, and floats over the point of use and can operate in any ocean depth over 20 meters and up to 1000 meters. In July 2016, OPT deployed its first commercial PB3 PowerBuoy off the coast of New Jersey (Ocean Power Technologies 2018; Parsa et al. 2017)

Considering the levels of validation and maturity that have been achieved through full-scale ocean deployments and accelerated life testing, and based on the American Petroleum Institute (API) Technology Readiness Scale (TRL) of 7 (API 17 N), the PB3 was estimated to have achieved a TRL6.

IPS Buoy

The IPS Buoy is a system for generating electricity from ocean waves designed by two Swedish companies, Interproject Service AB (IPS) and Technocean (TO).

Figure 10 shows the working principle of the IPS OWEC Buoy. As the buoy A moves up and down with waves, the acceleration tube B, which is open in both ends, will move vertically relative to the water column in it. The water column will “trap” the piston C, forcing it to move relative to the acceleration tube. This pumping motion is mechanically or hydraulically transformed to a rotary motion driving a generator D.

A typical IPS OWEC Buoy consists of a round 6–8 meter buoy floating, a 20 meter underwater acceleration tube with a piston, and an energy conversion machinery including an electric generator. The buoy is typically located in 50–100 meters deep water and anchored via a slack mooring arrangement. One such buoy built for 120 kW mean output may produce up to 1.0 GWh



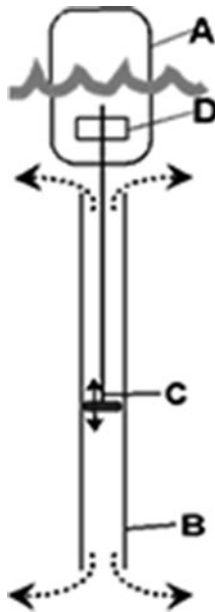
Wave Energy Utilization Buoy, Fig. 8 Two PB3 as deployed in the ocean in surface expression (Parsa et al. 2017)

DIMENSIONS	ELECTRICAL
Height: 13.3 m	Continuous Average Power based on 8.4 kWh/Day (typical) (Annual Average: Site Dependent)
Draft: 9.28 m	Payload Peak Power: up to 3 kW peak power to load: 7.5 kW (custom)
Spar Diameter 1.0 m Float Diameter: 2.65 m	Nominal Battery Capacity (ESS): 50 kWh (approx.); Modular and Scalable to 100, 150 kWh (appx.)
Weight: 8,300 kg	Zero wave day capacity: 100 W load for 1+ weeks (50 kWh ESS)
MOORING	DC Output: 24 V and 300 V (standard) 5 V to 600 V (custom)
Type: Single point or 3 point	Power Generation Sea States: 1 - 5
Point: Anchor or shackle	
Min Depth: 25 m	
Max Depth w/ standard design: 1,000 m*	*Other mooring designs available for deeper deployments

Wave Energy Utilization Buoy, Fig. 9 Product specifications for the PB3 (OPT 2018c)

annually. In the waters west of Scotland and Ireland where the power “content” in the waves is in the order of 50 ~ 70 kW/m wavefront, a 10 m IPS

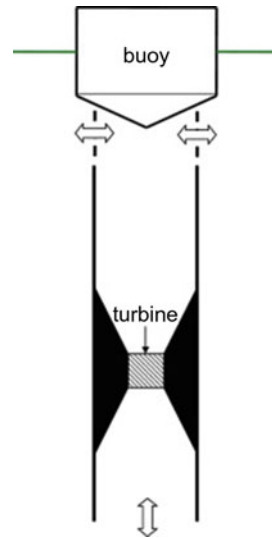
OWEC Buoy will reach a power of 150–250 kW and produce more than 1.4 GWh of electric energy per year The IPS OWEC Buoy (2018).



Wave Energy Utilization Buoy, Fig. 10 Working principle of the IPS OWEBC Buoy

The piston slides along the central part of the tube that needs to limit the stroke of the piston at both ends. Falcão A F O et.al (2012) proposed using a hydraulic turbine inside the tube to solve the problem of the end-stops, as shown in Fig. 11. In such situations, the dynamics of the IPS buoy can no longer be theoretically modeled as Falnes (1999b) did for a two-body heaving system or as done in (Falcão AF de et al. 2008) for the IPS buoy, for the cross-section of the acceleration tube is nonuniform, the flow inside the tube is also nonuniform, the inertia of the enclosed water cannot be represented by that of a solid body. Besides, apart from the axial force on the piston (or on the hydraulic turbine), the extra-axial force on the noncylindrical inner walls of the tube must be accounted for.(Falcão et al. 2012)

In regular waves, for a given buoy and given tube diameter and diameter ratio, maximum wave energy absorption is attained by an infinite number of tube combination mass (tube mass plus tube added mass) tube length and PTO damping coefficient. While in irregular waves, the performance was found to be much poorer (no phase control is considered), even if the tube combination mass,

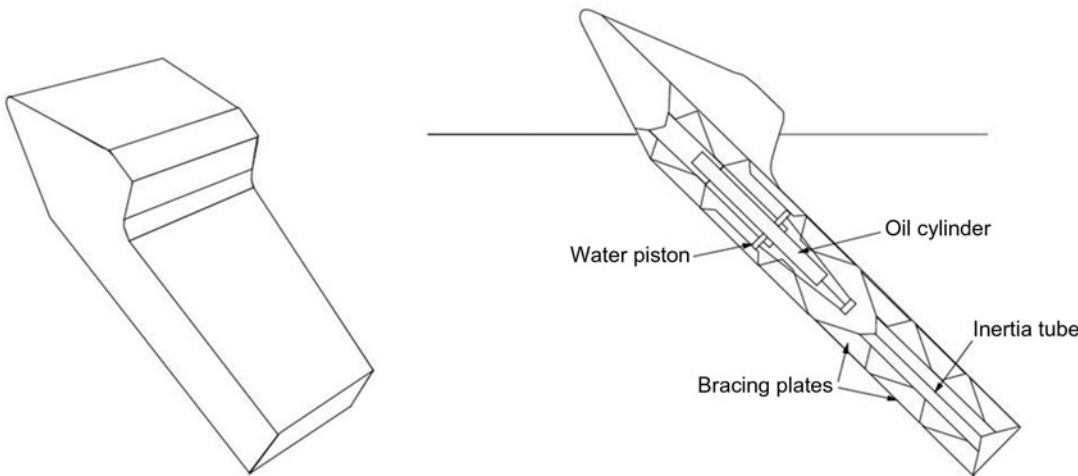


Wave Energy Utilization Buoy, Fig. 11 IPS buoy with the piston replaced by a hydraulic turbine. (Falcão A F O et al. 2012)

the tube length, and the PTO damping coefficient are optimized. The use of a hydraulic turbine in the tube instead of a piston was found in general to result in poorer wave energy absorption and to require a longer tube (Falcão et al. 2012).

The Sloped IPS Buoy

The sloped IPS buoy concept was developed by Edinburgh University as a device that would replace the Solo Duck(Salter 1974) as a confidence-building step in wave energy technology (Salter and Lin 1995). It is based on the axisymmetrical device designed by Interproject Services (IPS). The device is an inclined flat plate with a curved head (approximately 30 m wide and 6 m from front to back), which is inclined at an angle to the vertical (see Fig. 12 and Table 1, Thorpe 1999). The long tail is open to the sea at the bottom and is mainly empty, except for bracing plates. This adds a large inertia to movement in all directions, except for “slope” (i.e., back and forth in the direction of the tail). In this mode, the plates are edged on to the movement and so offer minimal resistance. An important feature of the buoy is the flared ends of the tube containing the water piston. When the movement of the piston exceeds the length of the



Wave Energy Utilization Buoy, Fig. 12 Shape and side view of the sloped-IPS Buoy (Thorpe 1999)

Wave Energy Utilization Buoy, Table 1 Summary of the characteristics of the sloped-IPS Buoy (Thorpe 1999)

Device width	30 m
Directionality factor	0.94
Mean power	1495 kW
Capture efficiency	81%
Power captured	1212 kW
Conversion efficiency	92%
Generator efficiency	88%
Availability	90%
Average power out	985 kW
Annual output	7.77 GWh

narrower part of the tube, it will enter the wider section allowing water to flow past it more easily. This flow effectively decouples the piston from the large inertia of the water mass in the tail. This arrangement avoids the shock loading that “end stops” can cause.

Salter and Rampen (1993) designed the “wedding cake” multieccentric radial piston hydraulic machine for the sloped IPS buoy, but the piston will be operating in seawater so the durability of the device needs to be assessed specially. Taylor J R M and Mackay I (2001) designed an eddy current dynamometer for the Swedish free-floating sloped-IPS buoy and optimized the sloped-IPS buoy models so that the detailed

response data can be collected for a very wide range of sea conditions.

More details of numerical simulation about sloped-IPS Buoy can be found in (Payne et al. 2006, 2015).

Summary

The ocean has a rich reserve of wave energy. So far, thousands of WEC patents have been proposed or put into practice, the number does not seem to be decreasing: new concepts and technologies replace or outnumber those that are being abandoned. All forms of devices should focus on cost, availability, and survivability to improve the industrial readiness level for large-scale commercial applications.

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Wave Measurement Buoy

Wanan Sheng
SW MARE Marine Technology and Consultation,
Cork, Ireland

Synonyms

Navy oceanographic meteorological automatic device; NOMAD; Ocean data acquisition system; ODAS; Waverider buoy; Wavescan buoy; Weather buoy

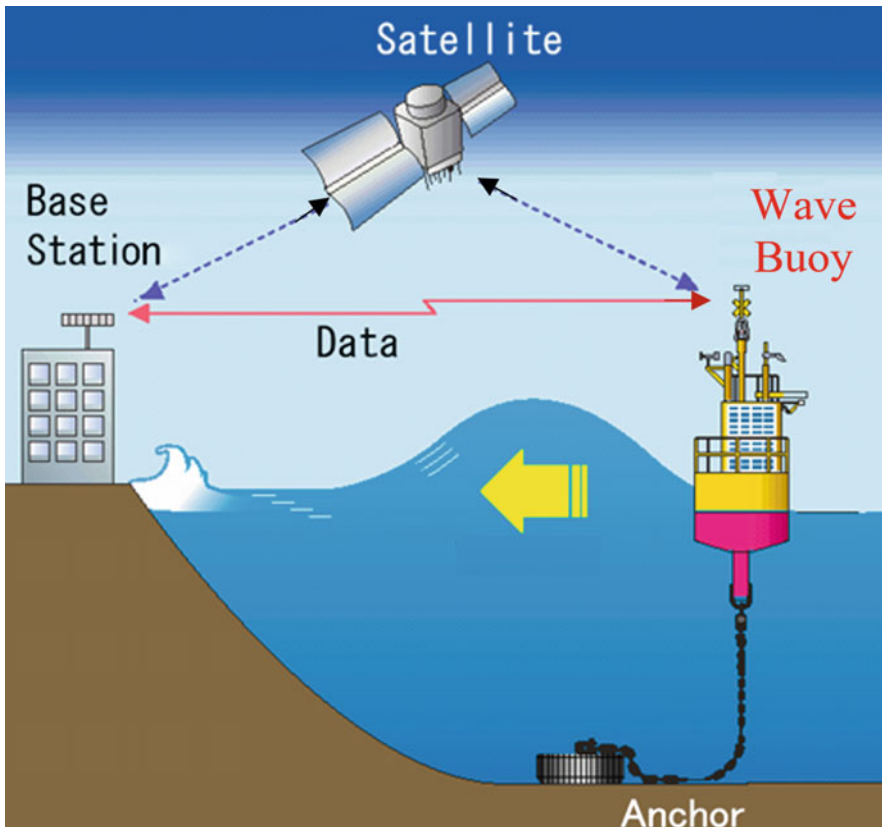
Definition

Wave measurement buoys are the special buoys which are commissioned to measure the ocean

waves (the modern wave measurement buoys may also collect the meteorological and oceanographic data, depending on the purposes of the wave measurement buoys). In most cases, the wave measurement buoys are anchored at seabed to allow some motions so to reduce the environmental loads acting on the buoy (see Fig. 1).

Scientific Fundamentals

Nowadays, wave measuring buoys are essential devices in many applications. Many national and international meteorological and oceanographic services monitor ocean wave characteristics nearly continuously over great expanses of the world's coastlines and in deep waters. These provide important information for mariners, fishermen, offshore oil workers, weather forecasters,



Wave Measurement Buoy, Fig. 1 Wave measurement and communications. (Adopted from website: <https://www.bosai-jp.org/en/solution/detail/29/category>)

ocean engineers, harbor authorities, coastal managers, marine scientists, and the public-at-large planning, coordinating, and conducting a variety of maritime activities.

The wave measurement buoys are normally anchored to the seabed using a single point mooring system (Fig. 1), which could provide enough flexibility for the motion of the buoy, allowing the buoy following the water particle motions, and therefore, the buoy's heave motion can be assumed to be the wave elevation (details can be seen in the following section).

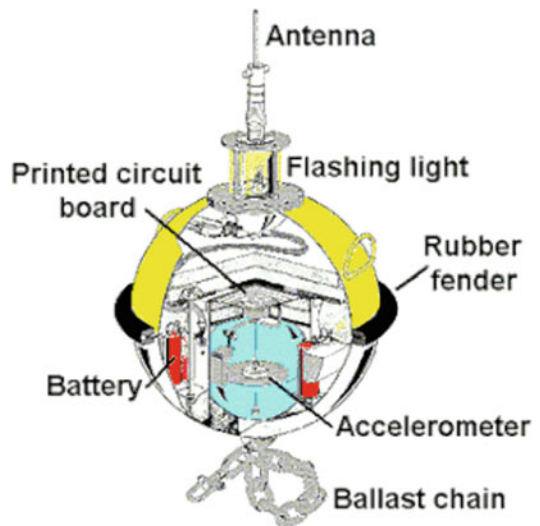
It should be noted that in many cases, the wave measurement buoys may have equipped with many sensors to measure other environmental and oceanographic parameters for providing more information to different users as mentioned above.

One of the advantages for the wave measurement buoys for measuring waves is their capability in long-term measurement: the measured data can be transformed using the direct radio communication or via a satellite (see Fig. 1); and the wave measurement is subject to battery life, and that is why the wave measurement buoys are equipped with solar panels.

Principle of Wave Measurement Using Wave Buoy

The wave measurement buoys measure the ocean waves on the assumption that the buoys follow the water particles traveling on a circular orbit (in deep water conditions). A good example is shown in Fig. 2 (WaveRider wave measurement buoy). The main measurement unit is the accelerometer which is located on a stabilized platform, so it could measure the up-and-down motion of the buoy (in a nautical term as heave motion), rather than the motions of all 6 degrees of freedom (6-DOFs). The accelerometer measures the heave acceleration, and the double integration of it would lead to the motion of the buoy.

It should be noted that the wave measurement buoys follow the water particle motion only for certain wave lengths/periods. The RAO (response amplitude operator) of the heave motion for the 0.9 m WaveRider is shown in Fig. 3. It can be seen that when the wave period is larger than 2.5 s, the



Wave Measurement Buoy, Fig. 2 Setup of the WaveRider buoy (website figure)

RAO is very close to 1.0 (equal to 1.0 when the wave period is larger than 3.0 s). The unit RAO means that the heave motion of the WaveRider buoy is same as that of the wave elevation.

When wave period is smaller than 3.0 s, an appropriate correction must be applied to get the correct relation between the buoy's heave motion and the wave elevation.

Another important factor is the buoy's phase response ("red line" in Fig. 3). When the wave period is larger than 1.7 s, the buoy's heave motion is in phase with the wave elevation, and for the shorter waves, there will be a large response phase between the heave motion and wave, for which the correction would be much more difficult and the measurement accuracy may be limited (it is also subject to the damping level of the buoy in waves).

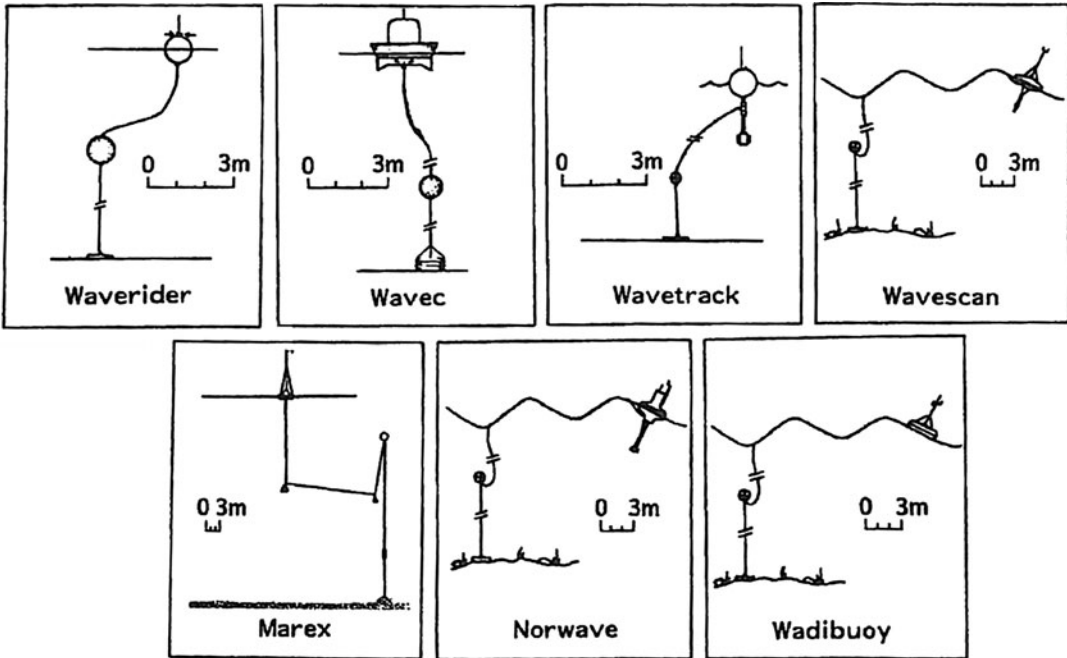
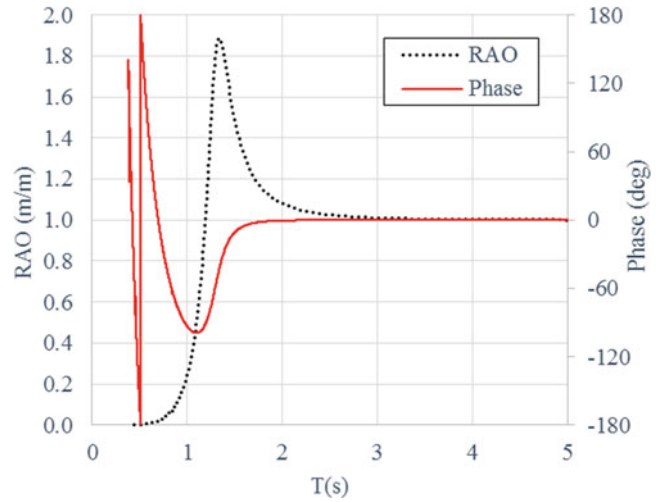
Comparison of Wave Buoys and Other Wave Measurements

One question may be asked that whether the wave measurement buoys can measure the wave accurately. For instance, whether the wave measurement buoys could follow the water particles or whether the buoy could avoid the large wave peaks (3-D waves). To answer this question there

Wave

Measurement Buoy,

Fig. 3 WaveRider buoy response to wave excitation (WaveRider diameter: 0.9 m)



Wave Measurement Buoy, Fig. 4 Wave measurements buoys used in WADIC project

are many comparisons of the wave buoys to other measurement methods.

In the WADIC project (Allender et al. 1989), seven different wave measurement buoys (Fig. 4) are used for the comparisons of wave measurements, including wavestaff, wave radar, and laser measurement. The conclusion is that all the

measurements are reassuringly good, and the large differences are found mostly for swells, high frequency waves (0.2–0.5 Hz), and the extreme sea states.

In another research work, Liu et al. (2015) studied the wave measurements using wave measurements in laboratory condition. Two wave

buoys are, namely WaveScan (1:8 & 1:16) and ODAS (1:11.25), used for comparisons with the wave measurements of wave probes in the wave tank. The data analysis has shown that the wave buoys measure waves in a very good agreement with the measurements of wave probes.

In a recent survey in 2010, Mettlach and Teng (2010) examined the ocean wave measurements around the world, and it is found that the nearly all wave buoys use onboard accelerometers for measuring waves; however, Japan, the Netherlands, South Africa, and the USA have experimented with GPS and D-GPS to produce wave measurements.

Key Applications

WaveRider Buoy

The Waverider buoys, developed by Datawell BV (<http://www.datawell.nl/products/buoys.aspx>), use the same sensors in the well-known Directional Waverider MkI, II, III, and 4. For the directional WaveRider 4 (Fig. 5), its sample frequency of the data processing is doubled to 2.56 Hz. The high frequency limit of the heave and direction signals is shifted from 0.58 to 1.0 Hz. With this choice, the high frequency limit of the wave buoy is determined by the hydrodynamic response of the hull, not by the onboard instrumentation. In addition, the DWR4 transmission protocol allows for a superior heave and horizontal displacement resolution. Following is the list that displays the features of Waverider buoys.

- Wave motion sensor based on a stabilized platform, accelerometers, and magnetic compass
- Measures wave height for wave periods of 1–30 s, accuracy 0.5% of measured value
- Measures wave direction
- Measures surface current
- Measures water temperature
- GPS for buoy monitoring and tracking through HF link
- Internal logger
- Power switch
- LED flash antenna

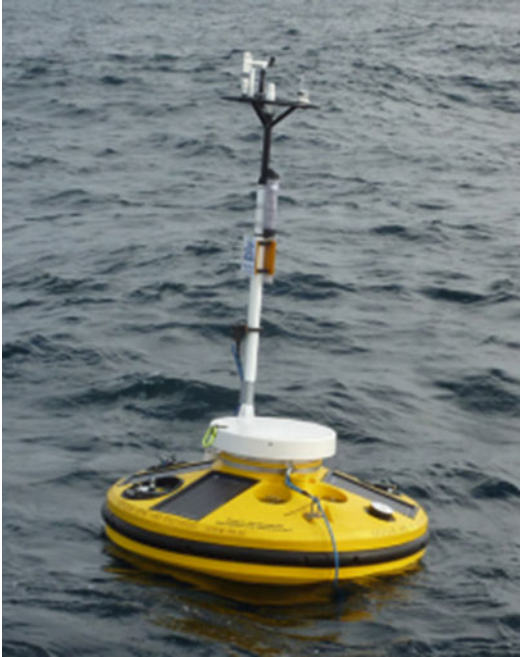


Wave Measurement Buoy, Fig. 5 WaveRider buoy

- 0.9 m (0.7 m) diameter spherical hull of AISI 316 including eccentric mooring eye
- Optional Cunifer hull, warranted not to corrode
- 1.6 years (10 months) battery life
- Optional HF transmitter range 50 km over sea
- Optional Iridium SBD module for ocean wide coverage and unlimited range
- Optional Iridium Internet module
- Optional Argos module for ocean wide coverage and unlimited range
- Optional GSM internet module for data transmission via the GSM network
- Optional Solar Panel system
- Optional hull painting

WaveScan Buoy

The SEAWATCH WAVESCAN buoy (usually referred to as the WAVESCAN buoy, Fig. 6) is a versatile instrumentation platform that may be used in all applications. It has been designed to



Wave Measurement Buoy, Fig. 6 WaveScan buoy

provide less drag and large buoyancy, making it well suited for deep offshore locations or areas of strong current forces. The buoy's hull design is based on the dynamic response and stability requirements from comprehensive wave tank testing.

The WAVESCAN buoy is a multiparameter buoy that can be used to collect directional wave data as well as meteorological, oceanographic, and water quality parameters. The buoy also has innovative “wells” for sensor mounting, easing service, and maintenance.

Features

- The ideal buoy for deep water and severe current conditions
- A unique design optimizes wave direction measurements
- Multiparameter platform for client's chosen selection of sensors
- Modular shaped hull for easier transport and local assembly
- Real-time data transfer and presentation
- Full onboard processing of all measured data



Wave Measurement Buoy, Fig. 7 ODAS buoy



Wave Measurement Buoy, Fig. 8 Triaxys Directional Wave Buoy

- Two-way communication link for data transfer and control of a number of buoys
- Special mooring design minimizes mooring influence on buoy motions
- Successful track record worldwide since 198.

ODAS Buoy

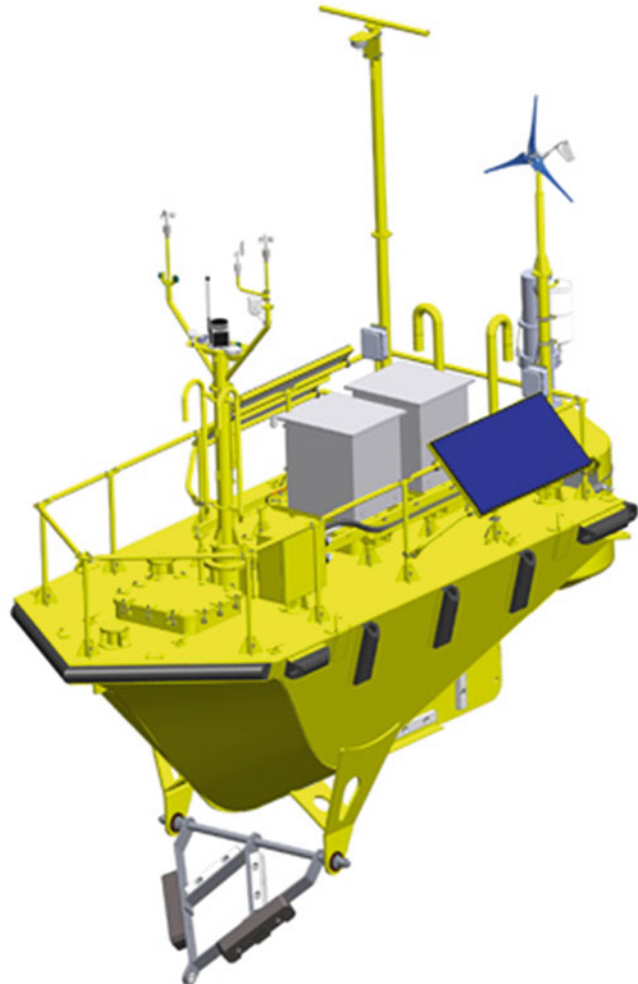
An **ocean data acquisition system (ODAS, Fig. 7)** is a set of instruments deployed at sea to collect as much meteorological and oceanographic data as possible. With their sensors, these systems deliver data both on the state of the ocean itself and the surrounding lower atmosphere. The use of microelectronics and technologies with efficient energy consumption allows to increase the types and numbers of sensor deployed on a single device.

Triaxys Directional Wave Buoy

The TRIAXYS™ Directional Wave Buoy (AXYS Technologies, http://axystechnologies.com/products/triaxys-directional-wave-buoy/?open_cat=40, Fig. 8) is a precision instrument incorporating advanced technologies that make it an easy-to-use, reliable, and rugged buoy for accurate measurement of directional waves.

- The TRIAXYS™ sensor unit is comprised of three accelerometers, three rate gyros, a fluxgate compass, and the proprietary TRIAXYS™ Processor.
- The TRIAXYS™ provides continuous wave sampling, support for any telemetry, up to 10 Hz motion sampling and up to 32 GB (>5 years) of data logging capacity.
- The TRIAXYS™ sensor also uses the WatchMan500™ controller, which is the core

Wave Measurement Buoy,
Fig. 9 A NOMAD buoy



Wave Measurement Buoy, Table 1 A list of organizations as of July 2010 that have deployed moored wave buoys (Mettlach and Teng 2010)

Organization	Types of buoys	No.
Agency for Assessment and Application of Technology, Indonesia	Wavescan	6
Australian Bureau of Meteorology	0.9-m Waverider [®]	16
Caribbean Integrated Coastal Ocean Observing System, Puerto Rico	2.5-m ODAS (ocean data acquisition system)	1
Centre for Environment, Fisheries & Aquaculture, UK	Waverider	20
Chesapeake Bay Interpretive Buoy System Stations, USA	Small discus	8
Coastal Data Information Program, Scripps Institution of Oceanography, USA	Waverider	51
Coastal Ocean Monitoring Center, National Cheng Kung University, Central Weather Bureau, Taiwan	2.5-m discus	8
Delaware Coastal Management Program, USA	TRIAXYS [™]	2
Environment Canada	3-m discus, 6-m NOMAD (Navy Oceanographic and Meteorological Automatic Device), 1.7-m WatchKeeper [™] , WatchMate [™]	52
Environment Canterbury Regional Council, New Zealand	Waverider	3
Gulf of Maine Ocean Observing System, USA	2-m ODAS	12
Hellenic National Center for Marine Research, Greece	Wavescan	10
Hydrometeorological Service of Vietnam	Wavescan	4
Icelandic Maritime Administration	Wavescan	6
Japan Meteorological Agency	Small discus	1
King Abdullah University, Saudi Arabia	3-m discus	1
Korean Meteorological Agency	3-m, 6-m NOMAD	5
Manly Hydraulics Laboratory, Australia	Waverider	7
Marine Institute, Ireland	TRIAXYS	
Marine Institute, Peru	Wavescan	6
Met Eireann, Ireland	2.5-m ODAS, Wavescan	6
Météo-France	ODAS 3-m discus	9
Meteorological Department, States of Jersey, UK	TRIAXYS	1
Michigan Technological University, USA	Small discus	1
National Data Buoy Center, USA	1.8-m, 2.4-, 3-, 10-, 12-m discus, 6-m NOMAD	105
National Institute for Ocean Technology Stations, India	Wavescan	4
National Marine Service, Italy	1.7 m Seawatch	14
National Research Council of Thailand	Wavescan	8
Oceanographic Institute of the Navy, Ecuador	Wavescan	3
Peoples Republic of China	Small discus	3
Ports Authority of Spain	Wavescan, Waverider, TRIAXYS	14
South African Weather Service	Waverider	6
UK Met Office	3-m ODAS, TRIAXYS	9
University of Connecticut, USA	Small discus	2
University of Michigan, USA	Small discus	1
University of New Hampshire, USA	Waverider	1
U.S. Army Corps of Engineers, Duck NC, USA	Waverider	1
Woods Hole Oceanographic Institution, USA	2.4-m discus	2

technology used in other AXYS monitoring systems.

- The TRIAXYS™ Directional Wave Buoy is solar powered.
- The TRIAXYS™ is easy to handle during deployment and can be rolled off a ship deck without any concern for sensor damage due to spinning.
- The TRIAXYS™ can also withstand extreme temperatures as its sensor is not subject to freezing.
- The TRIAXYS wave buoy can be configured with AIS Aid to Navigation to broadcast buoy position and weather information to local vessel traffic.
- TRIAXYS™ buoys have been deployed in waters from 10 to 300 m.
- Standard mooring configurations specially designed for wave buoys are available for water depths up to 300 m.

NOMAD Buoy

The AXYS NOMAD (AXYS Technologies, <http://axystechnologies.com/products/nomad-buoy/>) is a unique aluminum environmental monitoring buoy designed for deployments in extreme conditions. The NOMAD (Navy Oceanographic Meteorological Automatic Device) is a modified version of the 6 m hull originally designed in the 1940s for the US Navy's offshore data collection program. It has been operating in Canada's Weather Buoy network for over 25 years and commonly experiences winter storms and hurricanes with wave heights approaching 20 m.

- Custom configurable with a wide range of sensors for monitoring meteorological, oceanographic, and water quality parameters.
- A watertight central compartment houses the electronics payload, and there are two masts for mounting meteorological sensors, telemetry hardware, solar panels, and obstruction lights.
- Uses the WatchMan500™ controller to integrate sensor systems and provide onboard data processing, data logging, telemetry, and diagnostic/setup routines.
- A variety of telemetry options are available, including UHF/VHF, Inmarsat IsatData Pro,

Iridium, GPRS/HSPA, and CDMA/EVDO (Fig. 9).

Summary

There are many different types of wave measurement buoys deployed for measuring waves in oceans, and the principle for measuring waves is very similar. The choice of the wave buoys may be depending on many different requirements. For instance, in many applications, wave measurement may be only one of many other measurements: Small buoys are preferred if the wave measurement is the primary parameter and the shorter waves are more important while the large buoys provide more space and buoyance for accommodating more sensors for other measurements. Following table is a summary of the main buoys deployed in 2010 for measuring waves around the world (see Table 1).

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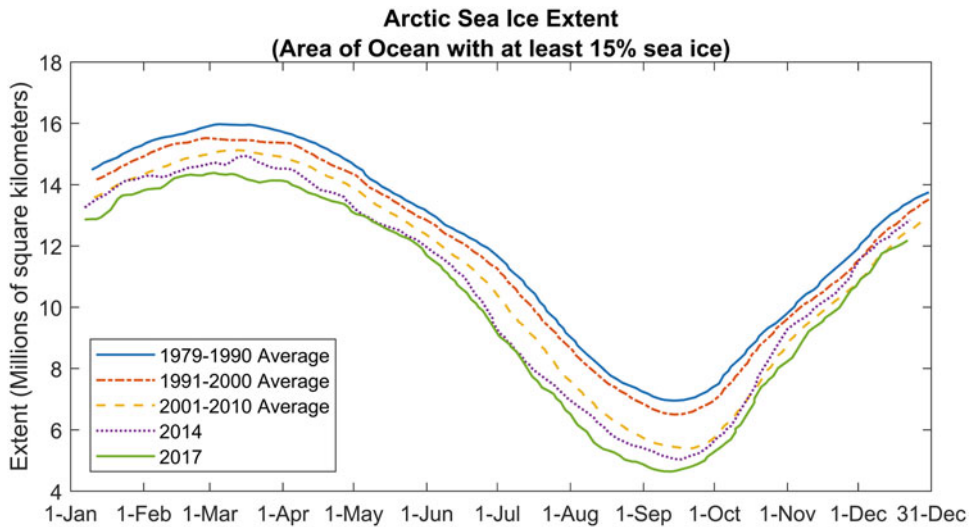
Wave-Ice Interactions

Hongtao Li

Offshore Engineering Technology Center of China Classification Society, Tianjin, China
Sustainable Arctic Marine and Coastal Technology (SAMCoT), Centre for Research-based Innovation (CRI), Norwegian University of Science and Technology, Trondheim, Norway

Introduction

As a result of global climate change, it has been observed shrinking of ice extent and increase of



Wave-Ice Interactions, Fig. 1 Extent of sea ice in Arctic Ocean (National Snow and Ice Data Center 2018b)

wave height in the Arctic Ocean (Thomson and Rogers 2014; Vizcarra 2018). Figure 1 demonstrates the evident diminishing trend of Arctic Sea ice extent. Ocean waves were recorded even in the order of 100 km into ice cover (Liu and Mollo-Christensen 1988; Meylan et al. 2014). Large wave itself poses as a threat to human activities, such as natural resource exploration, shipping, and tourism. Similarly, structural damages of sea-going vessels and offshore structures because of severe ice condition have been documented. In addition to low temperature, combination of severe wave and ice conditions in Arctic Ocean makes the safe operation in this region challenging. Currently, the stage of modeling wave-ice-structure interaction has not been reached yet.

Wave-ice interactions are highly nonlinear and coupled. Waves accelerate the formation of ice in winter season, break ice, mobilize ice around, and bring heat by convection to speed up the melting of ice in warm weather. At the same time, ice refracts, scatters, damps waves, and changes wavelength.

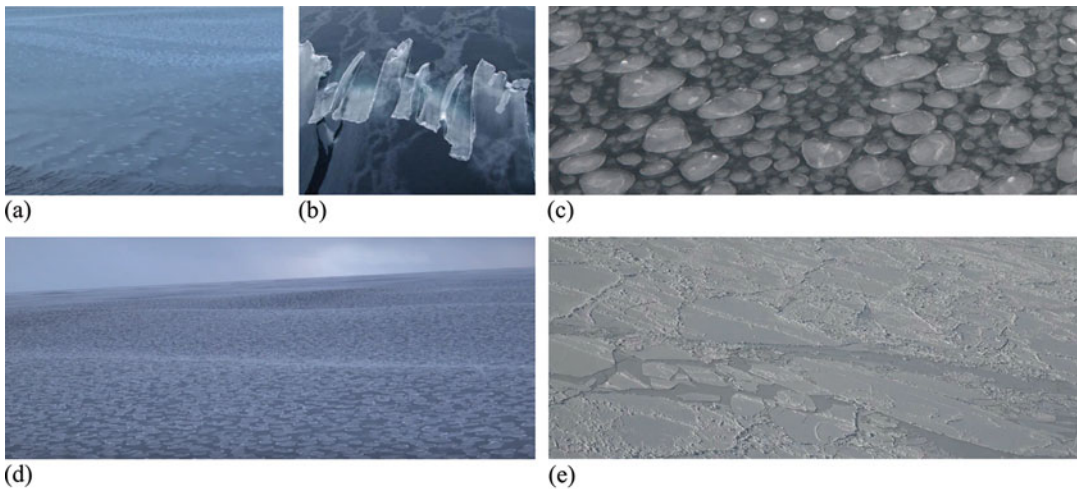
Herein, we focus on the effects of ice on waves. To start with, a brief account of sea ice types and forming process is provided. Then, effects of ice on waves and wave-ice interaction theories are

presented. Lastly, this entry concludes with summary.

Growth of Sea Ice

Sea water freezes and forms sea ice. This process is different from how iceberg is produced. Iceberg originates from glacier. It should be noted that iceberg is of fresh water ice.

In the supercooled sea water, frazil ice firstly forms, which has needle-like crystals form (Rogers et al. 2016). Under the action of current and waves, these fine spicules are swept together to become oil-slick and matte looking grease ice (see Fig. 2a). Then, the evolution phase differs due to different sea states. Under calm sea, grease ice turns into nilas ice. This kind of ice responds to waves like elastic thin sheet that easily bends (Carsey 1992). Nilas ice may interlock with each other (namely, finger rafting, see Fig. 2b), because of the actions from currents or winds. More specifically, nilas ice interchangeably rides over and under each other (National Snow and Ice Data Center 2018c). Thereafter, nilas ice may thicken through rafting and freezing to form smooth continuous ice cover under calm sea.



Wave-Ice Interactions, Fig. 2 (a) Grease ice and pancake ice in waves. (b) Nilas sea ice and finger rafting. (With the permission of Juan Botella. © Juan Botella for PolarTREC). (c) Close look of pancake ice. (d) Pancake ice in waves (see also Fig. 8 in Collins et al. (2017)). (e) Continuous ice cover. ((a) and (d) are reproduced with the

permission of Michael Collins. These images are from the video footage obtained by Michael Collins of the Naval Research Laboratory. © Michael Collins. (c) is reproduced with the courtesy of Junji Sawamura. © Junji Sawamura. (e) is from video footage that was obtained by Altan Turgut of the Naval Research Laboratory. © Altan Turgut)

Under slight to moderate sea, grease ice further aggregates to form pancake ice (see Fig. 2c, d), when the heat loss is sufficiently significant (Rogers et al. 2016; Wang and Shen 2010). The size of the pancake ice is governed by aggregation, bending, collision between ice and slope of sea surface (Shen et al. 2001). If the same environmental condition sustains, pancake ice cements between adjacent floes and thickens through rafting to form rough continuous ice cover (National Snow and Ice Data Center 2018a; Shen et al. 2001). If waves are strong, nilas ice may also evolve into rough continuous ice cover (Fig. 2e) by rafting and ridging. Sea ice growth process is summarized as shown in Fig. 3.

For a more detailed description of sea ice, readers are referred to Carsey (1992) for a summary of ice terminologies, Hobbs (2010) for physics of sea ice and Timco, and Weeks (2010) for engineering properties of sea ice. An old but still very valuable article is Weeks and Ackley (1986). Images of various ice types can be found in Fig. 8 of Lee et al. (2017), Antarctic Sea Ice Processes and Climate (2012), and PolarTREC (2018).

Wave-Ice Interactions

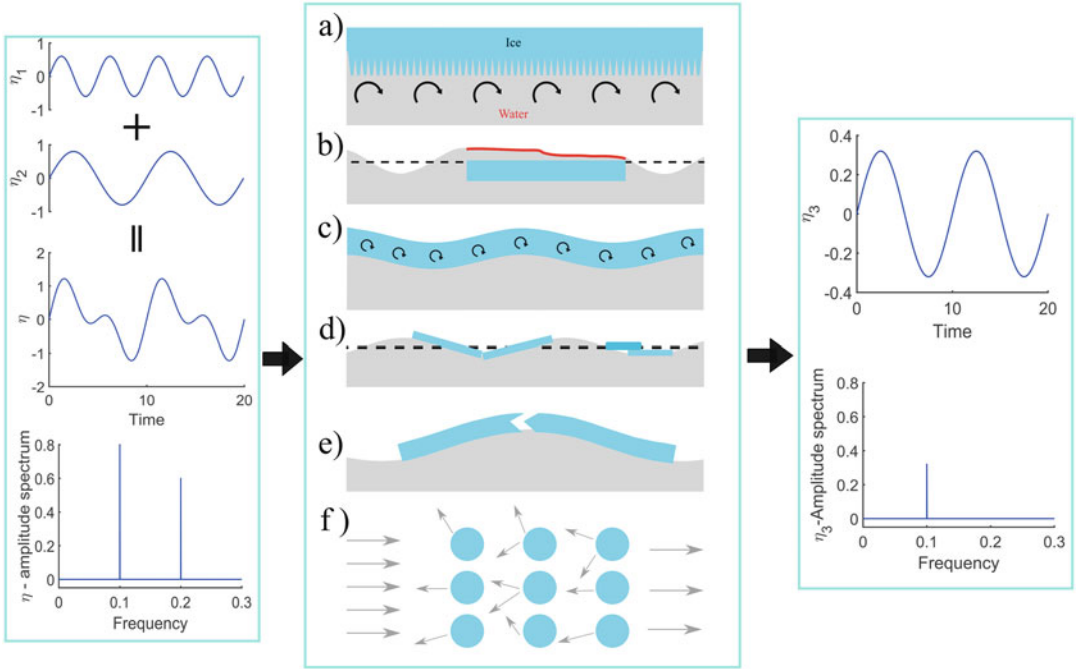
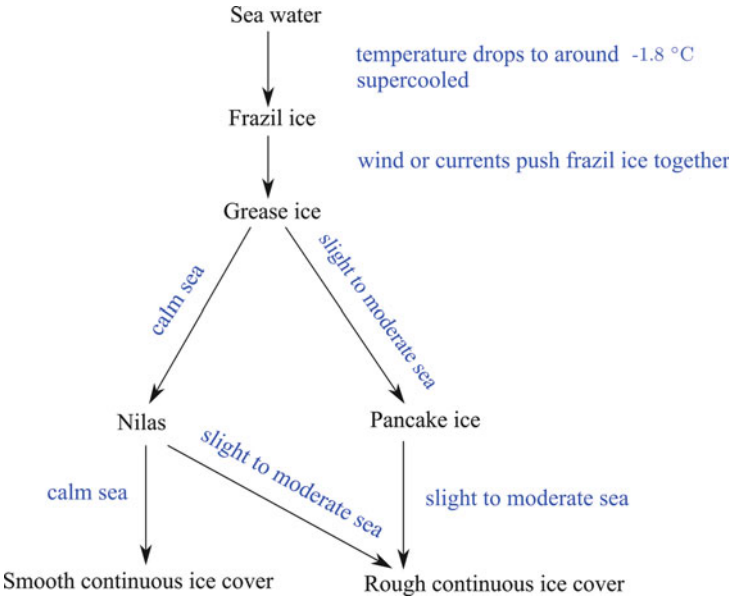
This section is divided into two subsections. Subsection “Effects of Ice on Waves” demonstrates various mechanisms to change properties of waves during wave-ice interactions. Subsection “Wave-Ice Interaction Models” gives an overview of different theories to model effects of ice on waves.

There have been several essential review literatures that summarize the knowledge in this field. Readers are advised to start with Li et al. (2018), Shen (2017), Zhao et al. (2015), and Collins et al. (2017). Hereafter, more comprehensive literature summaries, such as Squire et al. (1995) and Squire (2007), can be referred to.

Effects of Ice on Waves

To reveal how ice changes waves, Fig. 4 is provided to illustrate the various kinds of processes. This figure shows that a wave pattern, which consists of two harmonic waves (left panel of Fig. 4), attenuates when passing through an ice field. In this process, short waves are dissipated, and long waves with decreased amplitude retain (i.e., ice field acts as a low-pass filter, see the right

Wave-Ice Interactions, Fig. 3 Sea ice growth cycle

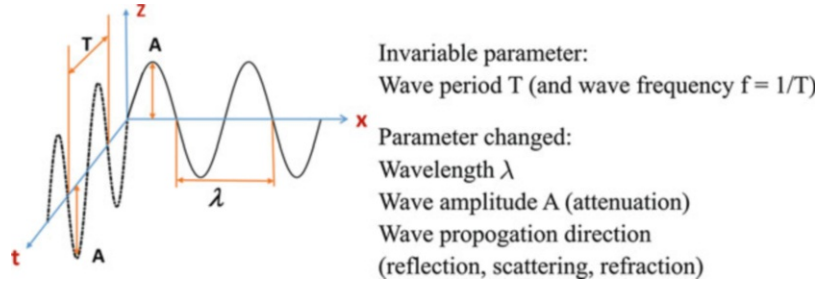


Wave-Ice Interactions, Fig. 4 Waves are attenuated due to (a) turbulence generated in boundary layer, (b) overwash, (c) vorticity of ice, (d) collision and rafting, (e) wave-induced fracture of ice, and (f) scattering. ((a) and (f) are adapted by permission from Springer Nature

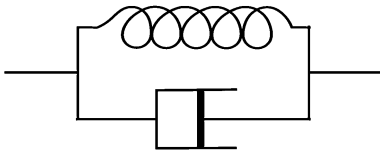
Customer Service Centre GmbH: Springer Nature (Zhao et al. 2015). © The Chinese Society of Theoretical and Applied Mechanics; Institute of Mechanics, Chinese Academy of Sciences and Springer-Verlag, Berlin/Heidelberg, 2015)

Wave-Ice Interactions,

Fig. 5 Change of wave parameters during wave-ice interaction



(a) Voigt model



(b) Maxwell model



Wave-Ice Interactions, Fig. 6 Two models to describe viscoelasticity of ice

panel of Fig. 4). Various attenuation mechanisms, as shown in the middle panel of Fig. 4, contribute to this phenomenon.

Additionally, wave propagation direction alters because of refraction, reflecting and scattering in ice fields (Fig. 4f). Wavelength is modified as well due to the presence of ice.

Figure 5 depicts what wave period, wave amplitude, and wavelength represent, which of them are changed and associates with what processes.

Wave-Ice Interaction Models

The development of wave-ice interaction theories can be traced back to nineteenth century. Nevertheless, most of the research work commenced after 1950s. In what follows, typical theoretical models are presented.

Figure 6 depicts two commonly used rheological models, i.e., Kelvin-Voigt model (henceforth denoted as Voigt model) and Maxwell model that consist of spring and dashpots. These two models are often employed to describe the constitutive

relation of ice, i.e., relation between strain, strain rate and stress.

Table 1 summarizes the wave-ice interaction theoretical models used to quantify the change of wavelength and amplitude of waves when waves traveling into ice fields. In Table 1, the second column lists abbreviations of each model, and corresponding assumptions for ice and water are tabulated in the two columns that follow. Last column gives the criteria to choose physically meaningful solution results.

The forthcoming parts are based on the work of Li et al. (2018) and references cited therein.

Two common assumptions of these models are (1) ice layer always stays in contact with water; (2) small-amplitude and small-slope waves (i.e., linear wave theory). Because of linear wave assumption, these models only produce exponential decaying behavior of waves when waves propagating in ice fields. On the contrary, large waves in some regions of Arctic and Antarctic exhibited a different attenuation trend.

Regarding VSK, VSDD, and VCWS, in which ice layer is modeled as fluid, immiscibility is assumed between water and ice.

According to the fundamentals of these models, they can be classified into two groups: (1) physics-based model and (2) phenomenological model. GW model falls within the first category. This model is applied for each single ice floe in ice field and includes as many physic processes (such as scattering, reflection, and refraction) involved as possible. Other models belong to second group, where all dissipation mechanisms that contribute to waves damping are considered by one parameter – effective viscosity, and concrete physical processes are neglected. Implicitly, the whole ice field is viewed as a heterogeneous media. Hence,

Wave-Ice Interactions, Table 1 Wave-ice interaction models (Li et al. 2018 and references therein, reproduced with permission of Tatiana Uvarova)

Wave-ice interaction model		Ice	Water	Damping of wave energy by	Selected wave number (criteria)
Mass loading model	WK	Discrete mass point	Inviscid	Scattering	Two real (Geophysics relevant)
Thin plate model	GW	Thin elastic plate	Inviscid	Scattering	Two real (Geophysics relevant)
Viscous layer model	VSK	Viscous fluid layer	Inviscid	Viscous damping in ice	One complex (least damped wave mode)
	VSLM	Thin elastic plate	Viscous	Viscous damping in water	One complex
	VSDD	Viscous fluid layer	Viscous	Viscous damping in ice and water	One complex (closest to open-gravity wave with least attenuation)
Viscoelastic model	VCWS	Viscoelastic fluid layer (Voigt model)	Inviscid	Viscous damping in ice (linear dashpot)	One complex (closest to open-gravity wave with least attenuation)
	VCFS	Viscoelastic solid beam (Voigt model)	Inviscid	Viscous damping in ice (linear dashpot)	Two complex (in first quadrant of complex plane)
	VCRP	Thin elastic beam	Inviscid	Damping forces due to vertical velocity of beam	Two complex (in first quadrant of complex plane)
	VCM	Viscoelastic solid (Maxwell model)	Inviscid	Viscous damping in ice (nonlinear dashpot)	One complex (close to wave number of gravity wave)

these models are also called as effective media continuum models.

Conceptually, WK model is suitable for sparsely scattered small ice floes. Intuitively, GW model works well for pack ice. According to previous studies, VSK and VSDD model are more applicable for grease ice. VSLM model fits to be used for high concentration ice field. In contrast, VCFS and VCRP model are more suitable for sea regions that are sparsely populated by ice. In principle, VCWS should be suited for all above mentioned ice types.

Summary

This entry gives a concise illustration of growth of sea ice under the influence of waves. Moreover, mechanisms to change waves during wave-ice interaction are listed. Lastly, different wave-ice interaction models are presented. This work

contributes to easing the efforts of readers when reading pertinent literatures in this field.

Cross-References

- [Ice Breaking Vessel](#)
- [Service Ships](#)
- [Special Marine Vehicle](#)

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Waverider Buoy

- [Wave Measurement Buoy](#)

Wavescan Buoy

- [Wave Measurement Buoy](#)

Weather Buoy

- [Wave Measurement Buoy](#)

WEC

- [Wave Energy Converters](#)

WEC – Wave Energy Converter

- [Power Take-Off System](#)

WEC: Wave Energy Converter

- [Wave Energy Utilization Buoy](#)

Weight

- [AUV/ROV/HOV Hydrostatics](#)

Welding Methodology

► Underwater Welding

Welding Technology

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Synonyms

Centimeter (cm); Kilojoule (kj); Melt inert gas
welding (MIG); Tungsten inert gas welding (TIG)

Definition

Welding is the process of forming a permanent connection between two or more materials (of the same or different species) through the combination and diffusion of atoms or molecules. Compared with riveting, welding can save about 15–20% of metal materials. Compared with casting, the production of welding structure does not require the production of wood mold and sand mold, nor special smelting and casting equipment. The process is simple and the production cycle is short.

Welding is an important link in the construction of ships and marine structures (Wang et al. 2017). The quality and efficiency of welding directly affect the construction cost and quality of ships and marine structures. The development of welding technology is the basis for improving the construction level of ships and marine structures. In terms of shipbuilding technology, before the 1930s, most ship structures were riveted or bolted. Around the Second World War, welding technology began to be widely used in ship structures, and higher requirements have also been put forward for the weldability and welding technology of hull steel (Ma et al. 2017).



Welding Technology, Fig. 1 Welding robot

The development and application of welding mechanization and automation technology in shipbuilding and marine structures are very rapid. The technology of submerged arc welding and gas-shielded welding began to be applied in the 1950s, and the method of CO₂ gas-shielded welding began to be applied in the 1970s. At the same time, various types of automatic welding and semiautomatic welding methods were developed. In the 1990s, welding robot technology becomes more and more perfect and practical. Plasma arc welding, laser welding, laser-arc composite heat source welding, friction stir welding, and diffusion welding have been applied in the construction of ships and marine structures, which have greatly improved the welding precision and quality (Fig. 1).

Introduction

Welding Method

Arc welding is mainly used in the construction of ships and marine structures. Arc welding includes electrode arc welding, submerged arc welding, gas tungsten arc welding, gas metal-arc welding,

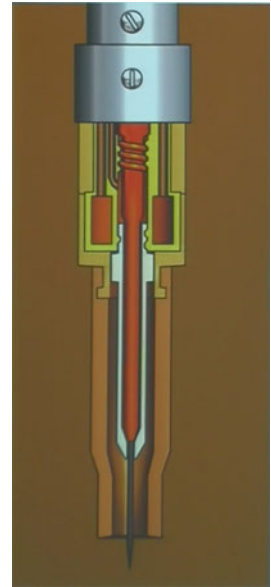
and plasma arc welding (Welding Society of Chinese Mechanical Engineering Society 2012). Their common characteristic is that the electric arc burning between the electrode and the workpiece serves as the heat source. When forming the joint, the filler metal can be used or not used.

Electrode arc welding is the earliest and still widely used arc welding method. It uses coated electrode as electrode and filler metal. The arc is burned at the end of the electrode and the surface of the welder. The coating can generate gas-shielded arc and weld under the action of arc heat, and the slag can be formed to cover the surface of the molten pool to prevent the interaction of molten metal from the surrounding gas. The more important function of slag is to produce physical chemical reaction with molten metal or add alloy elements to improve the performance of weld metal. Electrode arc welding equipment is simple, portable, and flexible. It can be used for repairing and welding short joints in assembly, and for welding hard-to-reach parts.

Submerged arc welding uses continuous feed wire as electrode and filler metal. When welding, a layer of granulated flux is covered over the weld zone, and the arc burns under the flux layer, melting the end part of the wire and the local base material to form the weld. Under the action of arc heat, part of the flux melts into slag and reacts with liquid metal. The slag floats on the surface of the metal pool. On the one hand, it can protect the weld metal, prevent air pollution, generate physical and chemical reaction with the molten metal, and improve the composition and performance of the weld metal. On the other hand, the weld metal can be cooled slowly. Submerged arc welding can use large welding current. Compared with electrode arc welding, its biggest advantage is good quality and high welding speed. Therefore, it is especially suitable for welding straight and circular joints of large workpieces, and most of them adopt mechanized welding.

Tungsten gas arc welding is a method of melting metal into weld by using the arc between tungsten electrode and workpiece. During the welding process, tungsten does not melt and only acts as an electrode. When welding, the inert gas argon or helium gas is used as the

Welding Technology,
Fig. 2 TIG welding gun



protective gas, and additional filler metal can be added as required, generally known as TIG welding. This method can control the heat input very well, so TIG welding is an excellent way to connect sheet metal and backing welding. The pulse form of welding current can better control the weld depth and improve the solidification characteristics of the weld pool. TIG welding can be used for almost all metal welding, including steel materials and nonferrous metals, and the weld quality is high, but compared with other arc welding methods, the welding speed is slow, and the thickness of the plates that can be welded is small (Fig. 2).

Gas Metal Arc Welding

In this method, the arc between the welding wire and the workpiece is used as the heat source, and the gas from the torch nozzle is used to protect the arc. Common shielding gases are argon, CO_2 , O_2 , or a mixture of these gases. When argon or helium gas is used as shielding gas, it is called metal inert gas arc welding (MIG welding). When CO_2 is used as a protective gas, it is called carbon dioxide gas-shielded welding. When mixed gas is used as a protective gas, it is called mixed gas arc welding.

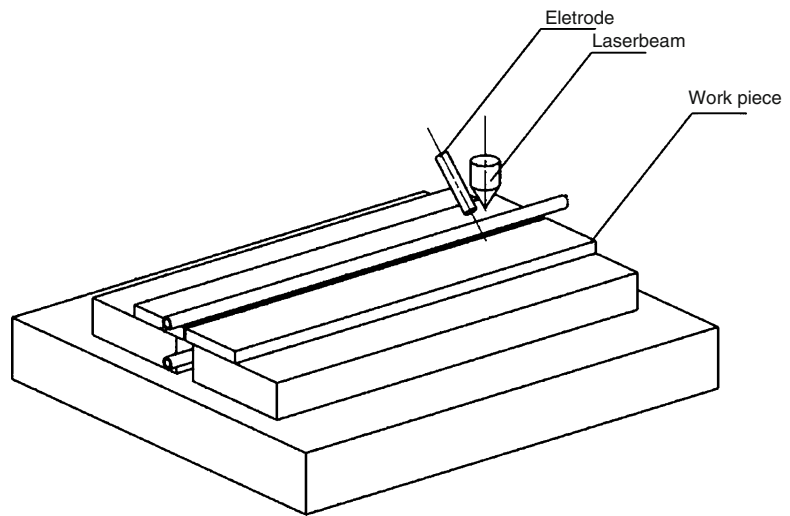
The main advantage of arc welding is that it can be easily welded in various positions, and it also has the advantages of fast welding speed and high deposition rate. It is suitable for welding of most metals and arc spot welding.

In recent years, the requirements for welding precision and quality have been continuously improved. High-precision welding methods such as laser welding, laser-arc composite welding, and friction stir welding have also been applied in the construction of ships and marine structures (Farajkhah and Liu 2016) (Figs. 3 and 4).

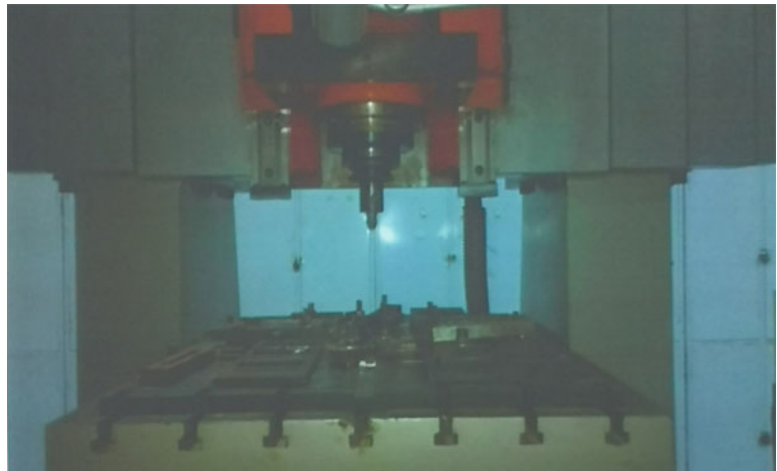
High Heat Input Welding of Steel

In the process of welding, the heat circulation of welding will lead to the thick tissue around the welding seam. These areas with large grain size are called the welding-heat-affected zone. Grain coarsening in the heat-affected zone may lead to the decrease of toughness of welded joints. Compared with the base metal, the toughness loss in the heat-affected zone is generally 20–30% and can even reach 70–80% when serious. Although the width of the weld-heat-affected zone is only a few millimeters in general, the serious decline in

Welding Technology,
Fig. 3 Laser welding



Welding Technology,
Fig. 4 Friction stir welding



the toughness of the coarse crystal region in the heat-affected zone is likely to be the crack source, and the safety of the engineering structure cannot be guaranteed (Hamada et al. 1995).

In order to reduce the construction cost and improve the productivity of shipbuilding, shipbuilding enterprises widely adopt 100–500 kJ/cm high heat input welding and develop a series of hull steel for high heat input welding, such as YP355 steel developed in Japan in the early 1980s, and YP390 steel and YP47 steel developed in the mid-1990s.

The energy of welding line refers to the heat input amount of welding seam per unit length of time. The expression is: $E=IU/V$ (kJ/cm), where I (A) is welding current, U (V) is arc voltage, and V (cm/s) is welding speed. Generally, the welding energy greater than 50 kJ/cm in the welding process is called high heat input welding, such as single-side arc welding, gas-electric vertical welding, electroslag welding, etc.

Under the welding of high heat input, the microstructure of heat-affected zone of traditional low-alloy high-strength steel will grow rapidly, and the strength and toughness of the welding seam will also be significantly reduced, which is easy to generate welding cold cracks, bringing difficulties to the manufacture of large steel structures. At the same time, with the improvement of steel plate strength, its impact toughness and welding performance are significantly decreased, and the susceptibility of welding crack is increased, which makes it difficult to manufacture large steel structures (Yan et al. 2006).

The research and development of low-alloy high-strength steel for high heat input welding must take into full consideration the formation process and control method of the structure in the heat-affected zone. A large number of studies and explorations have been conducted on the characteristics of microstructure and properties of welded joints, microstructure transformation behavior, and the dynamics of austenite grain growth of low alloy and high strength steel, especially for the welding heat-affected zone of the acicular ferrite formation as well as the use of oxide metallurgy welding coarse-grained methods, to improve the toughness of a large

amount of basic research and applied research. The results show that there are many factors influencing austenite grain coarsening and toughness loss in the weld coarse crystal area. The alloy design, purity, metallurgical production process, precipitated phase type, and its composition, shape, size, and distribution of materials all have different effects. According to the above research results, some methods to improve the toughness of steel-heat-affected zone for large wire energy welding are put forward, which are mainly to reduce the carbon content, refine austenite grain by second phase particle precipitation, and promote the formation of acicular ferrite by using the principle of oxide metallurgy.

In order to improve the toughness of the heat-affected zone, a lot of research work has been carried out successively. The two main methods are reasonable composition design and oxide metallurgy (Wang et al. 2012).

Composition Design of Steel for High Heat Input Welding

At present, the basic idea of steel composition design for high heat input welding is to reduce the carbon content to ensure the weldability, increase the content of manganese and microalloyed elements, increase the strength, and control the content of impurities such as phosphorus and sulfur. The oxide metallurgy method is adopted to form small dispersed fine inclusions, which can inhibit the austenite body length and control tissue transformation in the welding process, and keep the toughness of the steel coarse crystal area for high heat input welding at a good level.

Oxide Metallurgy Technology

Grain refinement is an effective way to improve the toughness. The smaller the grain size of the heat-affected zone, the better the toughness. In 1990, researchers at Nippon steel, Japan, first proposed the concept of oxide metallurgy, which controls the composition of oxides and makes them small and dispersed. The size, shape, and distribution of sulfide, carbonitride, and other nonmetallic inclusions are controlled by controlling the small, dispersed, high solubility

microinclusion material points in the molten steel. On the one hand, these inclusions prevent austenite grains from growing in the heat cycle of welding. On the other hand, they promote the nucleation of acicular ferrite, so as to refine the fine inclusions in the high heat input welding effectively and improve the toughness of the heat-affected zone. A large number of studies have shown that it is a very effective way to improve the toughness of heat-affected zone by utilizing the fine inclusions in these steels. Oxide metallurgy technology provides a new way of thinking for the development of high heat input welding steel.

Welding Deformation and Residual Stress

Welding deformation and residual stress are common problems in the welding of ships and marine structures, which have important influence on welding precision and welding quality.

The fundamental reason of stress and deformation during welding is that the welding parts are heated unevenly, and the thermal deformation and tissue deformation caused by heating are restrained by the rigidity of the welding parts themselves (Fang 2017). The stress and deformation occurring in the process of welding is called transient stress deformation, while the residual stress and deformation after the completion of welding and the complete cooling of components are called residual stress and deformation. The residual stress and deformation of welding seriously affect the bearing capacity and service life of welding structures (Chen et al. 2015).

During welding, the weldment is heated unevenly and the weld zone melts, while the thermal expansion of the high-temperature zone material adjacent to the welding pool is restricted by the surrounding cold state material, resulting in uneven compressive plastic deformation. During the cooling process, this part of the material (such as the two sides of the long weld) that has undergone compressive plastic deformation is also restricted by the surrounding metal and cannot freely contract, and to a certain extent, it is unloaded by stretching. At the same time, the weld pool solidifies and the weld metal cooling shrinkage is restricted, resulting in shrinkage

tensile stress and deformation. Thus, a shortened uncoordinated strain, called residual strain, or initial strain or inherent strain, is generated in the welded joint area.

Welding stress and deformation are caused by the interaction of many factors. The local uneven heat input during welding is the decisive factor for the formation of welding stress and deformation, and the heat input influences the metal movement around the heat source through the internal and external constraints formed by material factors, manufacturing factors, and structural factors, and finally forms the welding stress and deformation. The internal constraint of metal motion around the heat source mainly depends on the thermal physical parameters and mechanical properties of the material, while the external constraint mainly depends on the manufacturing and structural factors.

The basic law of welding stress and deformation is consistent with that caused by deformation and uneven temperature, but the process is more complicated. The main performance is that the temperature change range is larger when welding, the highest temperature on the weld can reach the boiling point of the material, and the temperature drops sharply from the welding heat source to the room temperature.

The appearance of welding deformation will bring a series of problems (Wang et al. 2018). Once the welding structure is deformed, it often needs to be corrected and is time-consuming. Sometimes, the more complex deformation correction workload may be greater than the welding work, and sometimes, the deformation is too large, may not be able to correct, resulting in waste. For the workpiece that needs to be machined after welding, the deformation increases the machine load and material consumption. The appearance of welding deformation will also affect the beauty and dimension precision of components and may reduce the bearing capacity of structures and cause accidents.

The control methods of welding deformation mainly include rigid fixation method, reverse deformation method, reasonable welding sequence selection, and reserved shrinkage allowance method.

Rigid Fixation Method

The method of reducing welding deformation by using external rigid constraint is called rigid fixation method. When assembling the plate, fixing the spliced steel plate on the platform with positioning welding can reduce the wave deformation of the steel plate. To prevent corner deformation, two end plates can be welded at both ends of the joint. In the process of hull construction, several process reinforcing plates are often welded along the joint weld to prevent angular deformation. When applying rigid fixation method, it is necessary to pay attention that the binding effect of the fixture will inevitably increase the welding stress and the possibility of welding crack. Therefore, this method is often used in welded steel structures, but not for cast iron and some high strength steel.

Inverse Deformation Method

Before welding, the deformation size and direction of the structure are estimated in advance, and the component is installed into a deformation in the opposite direction to offset the deformation caused by the welding process. This method is called reverse deformation method. This method is often used in the construction of ships and marine structures.

Select Reasonable Welding Sequence

In the process of assembly and welding, the influence on deformation is complicated due to the large number of welding seams. In order to control the deformation, the structure is divided into several parts, assembled and welded respectively, and then the welded parts are assembled into a whole. This allows asymmetrical or larger welds to contract freely without affecting the overall structure.

Reserve Shrinkage Allowance Method

The shortening of the longitudinal (along the weld direction) and transverse (perpendicular to the weld direction) of the component after welding can be controlled by estimating the weld shrinkage and leaving the shrinkage allowance in advance during the material preparation process.

The characteristics of welding residual stress include (Gadallah et al. 2017):

1. Mutual balance. Internal stress, tensile stress, and compressive stress coexist.
2. Follow the principle of stress superposition. The working stress caused by the load can be superimposed with the welding residual stress when the unrelieved stress is used.
3. A phenomenon of stress redistribution. When the stress and residual stress caused by external factors of the welded parts are superimposed, the ductile deformation will occur in this area when the yield point of the base metal is exceeded.
4. Without the occurrence of welding deformation, there is no obvious appearance. After a large number of welding, the appearance may not change, but there is a great stress in the interior.

The existence of welding residual stress has the following effects on the safety and service performance of welded joints and even welded structures:

Effect on Structural Rigidity

When the stress generated by the external load and the residual stress in a certain area of the structure are superimposed to reach the yield point of the material, the material in this area will undergo plastic deformation and further lose the ability to withstand the external load, resulting in the reduction of the effective section area of the structure and the decrease of the stiffness of the structure. When the external load is in the same direction as the longitudinal residual stress, the deformation is greater than that without residual stress. When unloading, the strain relaxation is less than the deformation when loading, and the component cannot be restored to the original size. The larger the tensile stress area in the component, the greater the impact on the stiffness and the greater the residual deformation after unloading.

Effect on Stability of Compression Bar Parts

When the compressive stress caused by the external load and the compressive stress component in the residual stress are superimposed to reach the yield point of the material, this section loses the ability to bear the compressive stress load further.

In this way, the effective cross-sectional area distribution of the bar is weakened, and the stability of the bar is changed.

Effect on Static Load Strength and Brittle Fracture

When there is a serious stress concentration and a high residual stress in the actual component, and if the working temperature is lower than the critical temperature of brittleness, under the combined action of tensile stress and working stress, the static load strength of the structure will be reduced and brittle fracture will occur under the action of the external load stress far below the yield point.

Effect on Fatigue Strength

The existence of residual stress makes the stress cycling average of cyclic load deviate, but does not affect its amplitude. As the mean value of the stress cycle increases, its limit decreases, otherwise it increases. Therefore, if there is stress concentration and residual stress, the fatigue strength will decrease. If the residual stress is eliminated under the external load, the residual stress will be weakened. The higher the stress concentration coefficient, more obvious the influence of residual stress.

Effect on Stress Corrosion Cracking

Stress corrosion cracking is a phenomenon of cracking of structures under the action of low stress in a certain environment. When material and medium match, the time required to produce stress corrosion cracking is inversely proportional to the stress. The greater the tensile stress, the shorter the time of stress corrosion cracking. Stress corrosion cracking can be avoided if residual compressive stress layer can be created on the surface of components that contact with corrosion media.

The Influence on Welding Precision and Dimension Stability

For the component with residual stress, if the original residual stress field is damaged during machining, and the residual stress can be released or redistributed, the component will be deformed and the machining accuracy will be affected. After the component with residual

stress is placed for a period of time, the stress relaxation will occur due to aging effect or unstable tissue, leading to the instability of structure size. When the temperature rises, the relaxation will increase significantly and the material with unstable tissue is produced after welding. As the unstable tissue changes over time, the residual stress changes greatly, and the dimension stability of the welded parts is poor.

Fatigue Strength of Welded Joints

The marine engineering structure is affected by wind, wave, and flow, and the sensitivity of high strength steel to stress concentration is higher than that of low carbon steel. Therefore, the fatigue strength of ocean engineering welding structure must be considered (Yan et al. 2016).

Factors affecting fatigue strength of metallic materials (such as stress concentration, section size, surface state and load conditions, the application environment, etc.) also have influence on the fatigue strength of welded structure (Sankaran and Sarma 2003). In addition, some characteristics of the welding structure itself, such as the change of properties of area near weld zone in the joint, welding residual stress, and so on, could also affect fatigue strength of welded structure (Chapatti et al. 2003).

The main technological measures to improve the fatigue strength of welding structures are:

Specific measures to reduce stress concentration include:

1. Using reasonable structural forms to reduce stress concentration in order to improve fatigue strength.
2. Adopting the welding joint forms with small stress concentration coefficient possibly, such as butt joint.
3. Comprehensive measures should be taken to improve the fatigue strength of joints when fillet welds are adopted (machining the end of the weld seam mechanically, selecting the shape of the joint of plates reasonably, ensuring the penetration of the weld root, etc.).
4. Using surface machining to eliminate various grooves in the weld and its vicinity can reduce the stress concentration in the components to improve the fatigue strength.

5. Adopting TIG arc or plasma beam-shaping method instead of mechanical processing to make a smooth transition between the weld seam and the base material.

Adjust the Residual Stress Field

The fatigue strength of the joint can be improved by eliminating the residual tensile stress at the stress concentration or producing the residual compressive stress here. This method can be divided into two types: One is structural integral treatment, such as integral annealing or overloading prestretching; the other is the local treatment of the joint, such as heating, rolling, and local explosion of a part of the joint so that the residual compressive stress is generated at the stress concentration of the joint.

Improve the Surface Properties of Materials

The fatigue strength of the joint can be improved by surface-strengthening treatment, extrusion with small wheels, tapping the weld surface and transition zone lightly with hammer, or spraying the weld zone with small steel balls.

Special Protective Measures

The fatigue strength of materials is often affected by the medium, so it is beneficial to apply notched surface coating.

Requirement of Marine Engineering Structure for Weld Seam

The butt weld of marine engineering structure should be fully welded. The shape of the remaining height of the weld should meet the requirements, and be smoothly transferred to the base material.

Fillet welds are usually used for the connection between plates and stiffeners, the fixation of bracket plates, etc. If there is an allowable assembly gap between welds, the weld thickness should be increased accordingly.

The important fillet welds, as well as those of strong stress components which may cause fatigue, should be fully welded. The technological measures such as alternating symmetric welding and presurfacing weld bead on groove surface can be adopted when welding (Calle and Alves 2015).

Reasonable assembly steps and welding sequence are adopted to control the deformation

caused by the weld to avoid excessive residual stress and crack (Zhao et al. 2017).

For the joints of thick components and pipes:

1. Low-hydrogen electrode and reasonable welding technology should be adopted to ensure complete penetration.
2. Preheating should be done before welding, and postwelding heat treatment should be done according to the regulations after welding.
3. The pipe joints shall not be burnt through during welding, the weld surfaces shall be continuous and even, and the joint between the two pipes shall gradually make a smooth transition. It is recommended to use small diameter electrodes for cover welding to improve the fatigue properties of joints. If the project requires polishing the joint of the pipes, the radius of curvature of the surface of the weld seam after grinding shall comply with the relevant design and manufacture regulations.

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Winch

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Definition

Winch is a mechanical device that is used to pull in (wind up) or let out (wind out) or otherwise adjust the tension of a rope or wire rope (also called “cable” or “wire cable”).

Scientific Fundamentals

Historical Development

The earliest literary reference to a winch can be found in the account of Herodotus of Halicarnassus on the Persian Wars (Histories 7.36), where he describes how wooden winches were used to tighten the cables for a pontoon bridge across the Hellespont in 480 B.C. Winches may have been employed even earlier in Assyria. By the fourth

century BC, winch and pulley hoists were regarded by Aristotle as common for architectural use (Coulton 1974).

The yacht *Reliance*, American defender of the 1903 America’s Cup, was the first racing boat to be fitted with modern winches below decks, in an era when her competitors relied on pulley systems (block and tackle).

In its simplest form, it consists of a spool and attached hand crank. In larger forms, winches stand at the heart of machines as diverse as tow trucks, steam shovels, and elevators. The spool can also be called the winch drum. More elaborate designs have gear assemblies and can be powered by electric, hydraulic, pneumatic, or internal combustion drives. Some may include a solenoid brake and/or a mechanical brake or ratchet and pawl device that prevents it from unwinding unless the pawl is retracted (Smith 1999) (Fig. 1).

Marine winch is an important lifting and pulling equipment used on boat/ship, on the dock or shore, in the port, on offshore platforms, etc. for anchoring, mooring, towing and weights lifting, or pulling, and there is also subsea winch used for some submarine operations. The main structures of marine windlass comprise drum, gypsy wheel, chain, cable, manual band brake, manual clutch, power system and control system, etc. In addition to drum, many winches have one or two small warping heads. Take marine drum winch, for example, it can have one or two drums with or without warping head. Clients can customize by their needs. Speaking of the commonly



Winch, Fig. 1 The form of winch (<http://cn.bing.com/>)

used marine winches, they can be anchor winch, anchor and mooring winch, towing/tugger winch, mooring winch, combined windlass like electric single drum combined winch and electric double drum combined winch, and so on (Blanc et al. 2006).

Ship winch refers to the winch mounted and used on a ship and it is a necessary deck equipment for the ship, which can be used for ship anchoring, mooring, towing, cargo lifting and cable pulling, and so on. According to drum quantity, there is single drum ship winch, double drum ship winch, and triple drum ship winch; the drum brake device should be able to withstand the winch support load, and it should not rotate after braking. According to drive type, ship winch includes electric ship winch, hydraulic ship winch, electric-hydraulic winch, diesel winch, steam winch, and manual winch; according to the type of drums, the winch drums can be split or undivided as needed; as for the brake type and brake application, there are mainly band, disc, mechanical screw, and spring applied. The main specifications of hydraulic winch consist of rated

load, support load, rope speed, rope capacity, power mode, etc.

Ship winches are very useful whenever the ships need to be anchored, stopped, towed, or there are something on ships needed to be lifted or pulled. When a ship reaches a port or pier, it needs to stop, and the ship anchor and mooring winch will be used to help with the ship anchoring and mooring to make sure it stops and keeps in the place safely without moving or drifting; according to ships with different sizes, shapes, and rope capacities, they will choose windlasses with different rated loads and drums; when the ship starts to set sail after the stop, the winch will let in the line attached to the anchor to make the ship leave the port or pier; when a ship needs to be towed under some special circumstances, the winch will connect the ship to the tug boat to implement the towing operations; when there are some loading and unloading operations on ship, the winch will also be used to finish the weights lifting work (Blanc et al. 2006) (Fig. 2).

The most common use of winches on ships is for mooring, which means the winches are

Winch, Fig. 2 Ship winch



usually located on the fore and after the decks at both sides of the ships. The mooring system can hold a ship in position and prevents the ship from drifting away from a berth. The ship mooring winch performs variety of functions, for example, the winch serves to store the mooring line when it is not in use, to haul the ship in place against environmental forces, or it can also act as a kind of safety device which releases the line load in a controlled way once the force in the mooring line increases to the limit of breakage (Chakrabarti 2005).

Mooring winch is the mechanical equipment used by ships to dock at docks; tie-up pontoons, alongside ships; to enter and leave docks; etc. It is commonly called mooring equipment.

Mooring winch is mainly composed of mooring lines, mooring piles, cable guide holes, winch, rope wagons, collision pads, etc. The ship can be moored by using a stranding machine to take up the stranding cable. At the bow, the mooring reel is usually driven by the same power as the windlass and can also be engaged or disengaged by clutches.

The type of winching equipment required in a particular mooring system depends on the type of mooring line to be handled and whether or not the floating vessel itself must initially tension the mooring lines or test load anchors. A floating vessel often has the means of adjusting mooring line tension, retensioning after anchor drag, and disconnecting individual mooring lines.

Main Types and Working Principles

There are two types of winch: normal type and automatic tensioning type (Smith 1999).

The composition of the normal type winch is similar to that of the windlass, and it can also share a set of driving device with the windlass. Its function is to tighten the cable and tie it to the cable pile. While the automatic tensioning cable winch is provided with a cable winding drum, the cable is wound on the cable winding drum, has a separate driving power, and automatically tightens or releases the cable by taking the tensile force of the cable as a signal, so as to adapt to the influence of tide level and cargo loading and unloading on the ship, and is beneficial to

reducing the labor intensity of the crew. However, one winch can only hold one rope by itself, so the number of mooring winch need more.

The working principle of the automatic tensioning hydraulic cable winch is: when the tension on the cable increases, the pressure of the hydraulic system increases, and when the pressure is greater than the setting value of the constant tension valve, the valve opens, allowing the inlet and outlet of the hydraulic motor to communicate. Under the action of the tension of the cable, the hydraulic motor rotates to release the cable, the tension decreases after the cable is released, and the constant tension valve closes. When the tensioning force on the cable is reduced, the hydraulic pump supplies oil to the hydraulic motor so that the hydraulic motor rotates to tighten the cable in the cable direction. When the pulling force on the cable is balanced with the set value of the constant tension valve, the oil supply of the hydraulic pump bypasses through the constant tension valve. In order to reduce the loss of hydraulic energy, large and small hydraulic pumps are permanently installed in the hydraulic system of the automatic tensioning cable winch. The large pumps are used in normal cable collection and discharge conditions. The small pump is used for automatic tensioning.

The load capacity, rope capacity, rope diameter, rated speed, and dimension of the winch vary according to different needs and conditions; they are in the different ranges and can be customized. A marine winch is a piece of marine deck equipment applied for handling ropes or wires, and works by spooling or winding the ropes or wires on a drum with a horizontal axis. Powered by electric or hydraulic motors, the winches on ship are fixed and used for various purposes.

The winch can also be divided into the following four types according to the material and characteristic of mooring lines.

Windlass

The most common method of handling and tensioning chain is through the use of a windlass. The windlass consists of a slotted "wildcat" which is driven by a power source through a gear-

reduction system. As the wildcat rotates, the chain meshes with the wildcat is drawn over the top of the wildcat and lowered into the chain locker. Once the chain is hauled in and tensioned, a chain stopper or brake is engaged to hold the chain. Windlass has proven to be a fast and reliable method for handling and tensioning chain (Luo et al. 2015) (Fig. 3).

Chain Jack

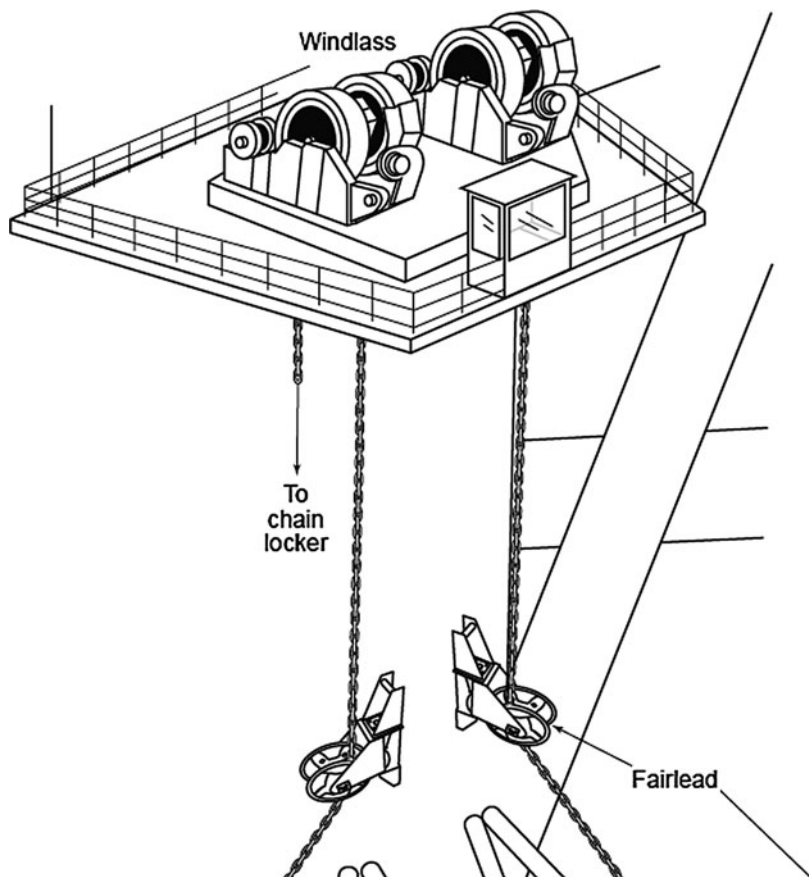
Chain jack is a device which reciprocates linearly to haul-in and tension chain. Usually powered by one or more hydraulic cylinders, chain jack engages the chain, pulls in a short amount of the chain, engages a stop, retracts, and repeats the process. Although chain jack can be a powerful means for tensioning chain, it is very slow and is recommended for applications not requiring frequent line manipulation (Luo et al. 2015).

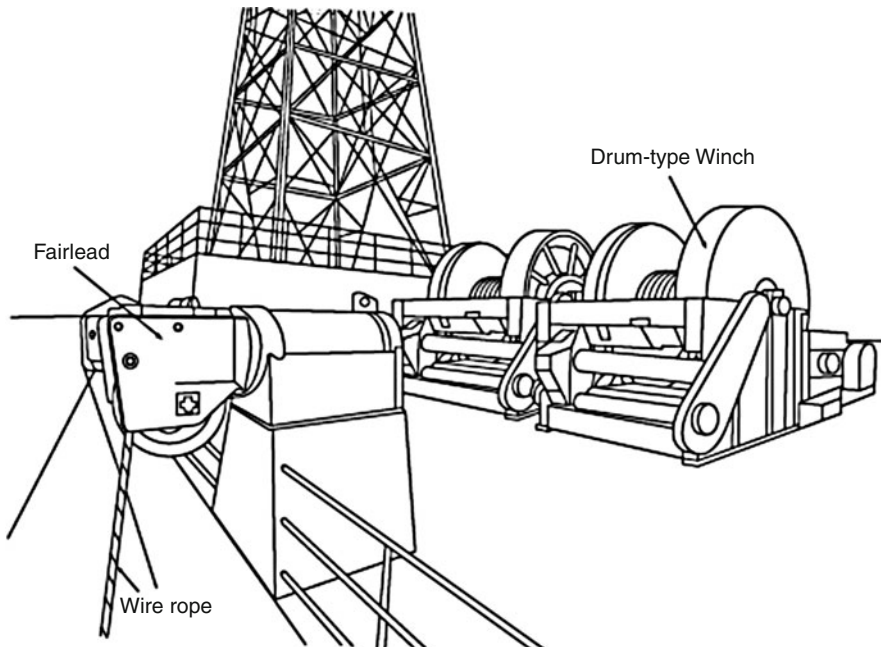
Drum-Type Winch

Conventional drum-type winch is the most common method used for handling wire rope. Operation of drum-type winch is fast and smooth. - Drum-type winch consists of a large drum on which the wire rope is wrapped. The base of the drum is often fitted with special grooves sized specifically to the size of wire rope being handled. The groves control the positioning of the bottom layer of wire rope on the drum. For subsequent layers of wire rope, an external guidance mechanism such as a level-wind is often used to control positioning of the wire rope on the drum. The tensioning capacity of the winch is a function of number of wraps on the drum.

Drum-type winch can be a cumbersome method of handling wire rope for deepwater or high strength mooring systems. As the requirement for line sizes and lengths increases, the size

Winch, Fig. 3 Windlass





Winch, Fig. 4 Drum-type winch

of the winch can become impractical. In addition, when wire rope is under tension at an outer layer on the drum, spreading of preceding layers can occur causing damage to the wire rope (Luo et al. 2015) (Fig. 4).

Linear Winch

Linear winch is similar in principal to chain jack. Two sets of grippers, one stationary and one translating, are used to haul-in and tension the wire rope. Linear winch is available in a single-acting form, in which the wire rope moves intermittently as the gripper is retracted to begin another stroke, and in a continuous double-acting form, two translating grippers are used alternately for continuous smooth motion of the wire rope. Linear winch is most applicable in a permanent application when high tension and large-diameter wire rope is required. A take-up reel is necessary in this case to coil the wire rope after it passes through the linear winch (Luo et al. 2015).

Traction Winch

Traction winch has been developed for high tension mooring applications as well as for

handling combination mooring systems. It consists of two closely spaced parallel mounted powered drums, which are typically grooved. The wire rope makes several wraps (typically 6–8) around the parallel drum assembly. The friction between the wire rope and the drums provides the gripping force for the wire rope. The wire rope is coiled on a take-up reel which is required to maintain a nominal level of tension in the wire rope (typically 3–5% of working tension) to ensure the proper level of friction is maintained between the wire rope and the traction winch. This system has been favored for use in high tension applications due to the compact size, capability to provide constant torque, and ability to handle very long wire rope without reduced pull capacity.

Key Applications

The rope may be stored on the winch. When trimming a line on a sailboat, the crew member turns the winch handle with one hand, while tailing (pulling on the loose tail end) with the other to maintain tension on the turns. Some

winches have a “stripper” or cleat to maintain tension. These are known as “self-tailing” winches.

Winches are frequently used as elements of backstage mechanics to move scenery in large theatrical productions. They are often embedded in the stage floor and used to move large-set pieces on and off.

Off-road vehicles often carry a winch which may be electrically or hydraulically powered and is wound with rope or a wire cable. If the vehicle loses traction, the winch is used to pull it back to firmer ground.

The purpose of windlass/winch is to control the speed of chain (cable) movement when releasing and recovering, and pre-tightening and adjusting the tension of chain (cable) while anchoring.

There are some requirements for winch design (Piggott 1977):

1. To ensure that the ship can still tie up when it is subjected to wind force below level 6.
2. The tensile force of the winch shall conform to the requirements of “code for the classification and construction of steel sea-going ships.
3. The wringing speed is usually 15~30 m/min, with a maximum of 50 m/min. When the rated pulling force is reached, it can be reduced to the minimum.

Cross-References

- [Design of Mooring System](#)
- [Mooring System](#)

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Wind Propulsions

- [New Technologies in Auxiliary Propulsions](#)

Wind Speed

- [Ship Operational Environment](#)

Wind Tunnel Test

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Synonyms

[Aerodynamic drag](#); [Artificial air flow](#); [Hydrodynamic coefficients](#); [Submersible](#)

Definition

Wind tunnel tests are tests performed in a tunnel with aircrafts, including wing-in-ground-effect ships, other surface ships, offshore structure models, or submarine and submersible models in different speed artificial air flow, which are the most popular tests in aerodynamic and hydrodynamic research. As the similarity between water and air, wind tunnels are also useful and important tools compared with towing tanks (<https://www.en.wikipedia.org/wiki/windtunneltests>).

Scientific Fundamentals

Wind Tunnel Types

Different types of wind tunnels can be divided by the top wind speed in the tunnel. These include low-speed wind tunnel, subsonic wind tunnel, transonic wind tunnel, supersonic wind tunnel, and hypersonic wind tunnel (Korsmeyer 1996). A division of various wind tunnels is given in Fig. 1.

The wind speed in the tunnel is V , and the speed of sound in the air is c , then the Mach number can be defined by the formula, $Ma = V/c$.

- Low-speed wind tunnel, $Ma < 0.3$. The air density keeps the same when as its speed is zero.
- Subsonic wind tunnel, $0.3 < Ma < 0.8$. The air density becomes different.
- Transonic wind tunnel, $0.8 < Ma < 1.2$.
- Supersonic wind tunnel, $1.2 < Ma < 5$.
- Hypersonic wind tunnel, $Ma \geq 5$.

Submersible or submarine models can be installed in wind tunnel and resistance measurement or flow-field measurement can be carried out. Usually the low-speed wind tunnel is suitable for their tests.

Wind Tunnel Test Types

A division of various wind tunnel tests is given in Fig. 2.

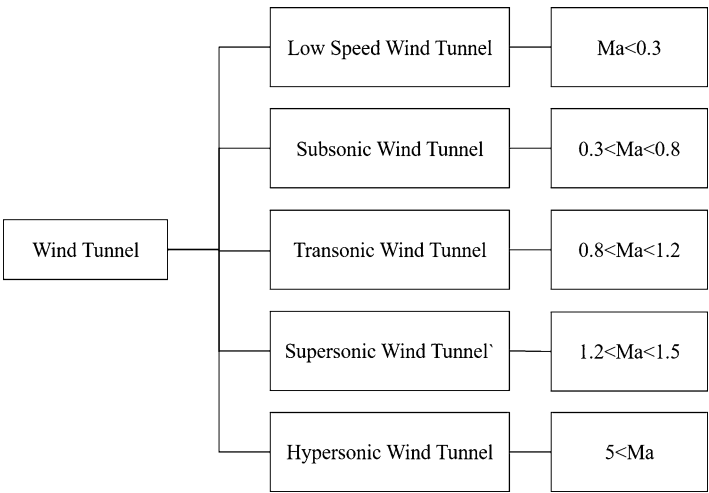
These include force measurement, pressure measurement, heat conduction tests, and flow-field observation.

- Force measurement tests are the basic wind tunnel tests. The resistance, trimming moment, lifting force, and so on can be measured of the model.
- Pressure measurement tests are important when people want to know the pressure distribution on surface of the model.
- Heat conduction tests are usually for high speed or super high-speed vessels. As the speed increases, the friction between the air flow and vessels become larger. When it is supersonic or even hypersonic, the heat is large enough to burn the vessel to ashes.
- Flow-field observation is usually carried out by using smoke or wires, which will show the direction of the air flow. Nowadays, new technologies such as particle image velocimetry are applied since the 1980s.

Historical Development

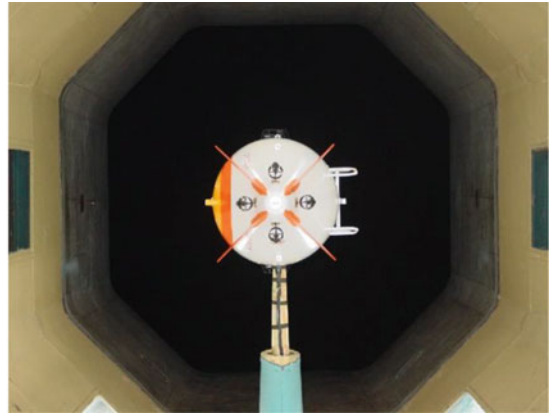
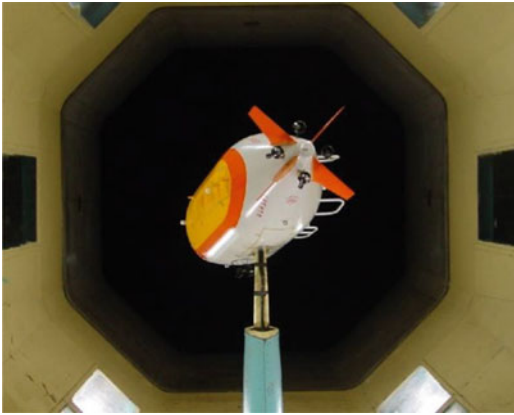
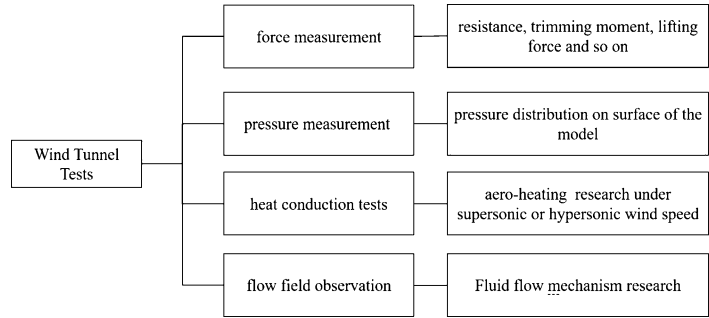
It is believed that the first wind tunnel in the world was built in 1871 by the British. The Wright Brothers built a wind tunnel in 1901 in order to carry out different tests for their aircraft model, whose section area was 0.56 m^2 and top wind speed was 12 m/s . With the help of the wind

Wind Tunnel Test,
Fig. 1 A division of
various wind tunnels



Wind Tunnel Test,

Fig. 2 A division of various wind tunnel tests



Wind Tunnel Test, Fig. 3 The Jiaolong sub model in wind tunnel

tunnel their plane flew in the sky 2 years later, which was believed the first plane in the world.

In the middle of the last century, wind tunnel tests became widely used by not only aircrafts manufacturers but other company, for example, the car industry. In the 1920s, Jaray started to do wind tunnel tests for cars. He did a good job to decrease the wind resistance of cars by carrying out a lot of wind tunnel tests. The aerodynamic drag takes more than 30% of the total resistance when a car drives at 180 km/h. After World War II, vessel engineers built hundreds of wind tunnels and have made great efforts to reduce the aerodynamic drag by more than 60% (Quix 2015; Forte 2012).

Scientists found that aerodynamic and hydrodynamic flow were always based on the same laws, which became the theoretical basis that ship and submarine model tests were widely carried out in wind tunnels. The hydrodynamic

coefficients of submarines or other under water vessels are studied.

During the design of the 7000 m deep manned Submersible “Jiaolong,” lots of wind tunnel tests were carried out (see Fig. 3). Different design was compared for better performance and higher safety.

Key Technology in Carrying Out a Wind Tunnel Test

There are four elements for successful wind tunnel tests. They are the wind tunnel and fans, the model, measurement device, and model vibrations.

1. The wind tunnel and fans

The wind tunnel should be large enough for the model. The cross-sectional area of models should be less than 5% of the wind tunnel.

The artificial aero flow produce by fans in wind tunnel should be fairly well distributed and stable for both the time dimension and space dimension.

As there is a lot of noise when the fans are working, it is suggested to build the wind tunnels far from living area.

2. The model

The model ship and the full-scale ship are geometrically similar. The model should be manufactured with high precision. And there are rudders or fins, which should be manufactured and installed to the model. Sometimes they are too small to be well manufactured. In other cases, manipulators, for example, are usually too complex to be manufactured. The model is simplified in most cases (Owens 2011). How to simplify the model should be recorded. Turbulence stimulators are usually installed on the model as the Reynold Number is not high enough to make the laminar flow turn to turbulence flow.

And the roughness of the model skin should be carefully paid attention to. Usually the models are painted with some standard methods.

3. Measurement device

Measurement devices are usually divided into three kinds, including force measurement balances, pressure and speed measurement sensors, and aero flow visualization devices. Five- or six-component testing system and the methods for resolving calibrating matrix are proposed in wind tunnel tests. Pressure sensors are developed to capture the high-frequency fluctuation aero flow. Particle image velocimetry technologies are widely applied in recent decades as flow visualization methods. Smoke and thin wires are the traditional and classical methods (Brooks 2014).

4. Model vibrations

During wind tunnel tests, aero turbulence, model structure, and aerodynamic forces will make the model vibrate with a high frequency. Model vibrations affect the measurement and the model tests cannot carry out in some cases. Model vibrations should be restrained and controlled.

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Winding Wire

► Cable

Winterization of Polar Engineering

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Synonyms

[Antifreeze of the polar ocean platform](#); [Polar Engineering antifreeze](#)

Definition

The polar offshore platform is the carrier of resource exploration, development, transportation, and scientific research activities in polar regions. However, the polar climate is extremely harsh, and the temperature is very low over the years. The Polar Engineering antifreeze is a technology that ensures the safe and efficient operation of the offshore platform in a complex and abominable low-temperature polar region and

provides a more comfortable working life environment for the staff.

Scientific Fundamentals

Polar Low-Temperature Environment

In the Arctic region, the winter time is long, the temperature is between -43°C and -26°C , the average temperature is -34°C , the air sea air exchange is strong and the humidity is very high, and the relative humidity is above 95% for most of the time, which shows the fog and thick fog. The ice overlying under the extreme climate has a great influence on the polar ocean platform (Horjen 1989).

According to the guidelines for the polar ships issued by the China Classification Society, the polar vessels operating at low temperature or season should take into account the exposure to the ambient temperature of the structure, equipment, and the surface of the system and take appropriate measures of protection or operation control to ensure that the ship and equipment have operational capability in the polar service temperature environment (China Classification Society 2016).

Ice Cause, Harm, and Ice Zone Specification

Cause of Icing

There are two main reasons for ice formation on the offshore platforms, namely, meteorology and ocean spray. The former is a secondary cause and the latter is the main reason.

The meteorological conditions mainly include snow, freezing rain, fog and hail, and so on. The overcooled droplets in the atmosphere form a large amount of icing on the surface of the structure, such as the icing on the cable, the lifesaving equipment, the air vent, and so on (Xie et al. 2017). When the temperature is less than or equal to 0°C , freezing rain will cause obvious ice formation on the platform surface. Freezing fog can also cause ice. The amount of ice generated by freezing fog is usually small, but if the duration of the fog is longer, it may cause an increase in ice accretion. Snow will cause icing

on the horizontal structure of the platform, such as deck.

The droplet is caused by the impact of the ocean wave platform. Due to the lower seawater temperature and air temperature, the droplet adheres to the hull, and the heat dissipated rapidly, resulting in ice formation. The conditions for ice spray formation are as follows: the air temperature is below -2°C , the wind speed is greater than 16 knots, and the surface seawater temperature is -1.8°C ~ -6.0°C (Yang 2015).

Icing Damage

Polar ice platforms can reach hundreds of tons of ice and may encounter the following phenomena and hazards when operating in low-temperature environments:

1. Ice will affect the platform body, the wet gas condensation, and freezing of the cold surface of deck and cargo equipment, and the splash water and rainwater on the superstructure of the platform are frozen on the exposed surface, which will reduce the stability of the platform. For floating platforms, the weight of icing will increase the heeling moment and reduce the freeboard height (Wold 2014). Ice formation will not only affect the floatability and stability of the platform but also increase the wind area of the platform and increase the wind overturning moment, which may lead to the overturning of the platform.
2. Icing can cause damage to the equipment in the polar working area, such as freezing winches, cranes, valves, covering windows, blocking air outlets, etc. Low temperature will affect the material properties of the platform structure, increase brittleness and thermal expansion, reduce allowable stress and structural reliability, and damage the equipment when used. The windows of the vents are mostly designed with mesh and lattice, which are easy to be covered with ice and snow plugging, which will bring problems to the health and mechanical operation of personnel. The lunar pool is an open area located in the center of the offshore platform; the drilling equipment is operated in this area. The icing will cause the valve to be

unable to work and the efficiency to be low (Xie et al. 2017). Icing may affect deck operation of platforms, such as the possibility that lifeboats/lifesaving valves can't be released properly.

3. Ice can affect the safety of the platform. Salt-water is likely to freeze on the antenna, GPS, radar, and other equipment, affecting the normal work of these devices (Gauthier et al. 1997). If the ice covers the cab windows, it will affect the operator's line of sight. Ice in the open deck passage of the platform will cause difficulties for the crew to walk. If the ice fall off the superstructure, it will directly threaten the personal safety of the crew (Xue 2013). Icing can reduce the self-rescue capacity of the offshore platform, such as covering fire extinguishing equipment, freezing fire extinguishing water gun valves, and blocking smoke sensors, all of which have a great impact on the safety of the platform.

Ice Zone Specification

In the 1930s, Finland and Swedish Maritime Administration first standardized the ice area enhancement of ships in the Finland-Swedish Ice Class Rules (FSICR). The polar classification requirements issued by the International Association of Classification Societies, including ten major classification societies including China Classification Society, were formally promulgated in 2006, and the hull structure was modified in 2010. In addition, Det Norske Veritas (DNV) also adds the WINTERIZATION entry symbol, which mainly considers the design requirements of the platform for the short-term working in the extremely cold environment and does not consider the ice in the surface. The aim is to ensure that the system and equipment can work properly in the design temperature range and reduce the threat of cold environment to operators (Det Norske Veritas 2008).

DNV has the following requirements for ocean engineering projects:

1. In ship or ocean engineering projects of the WINTERIZATION entry symbol, the calculation of space heating heat balance must assume

that the ambient temperature of the space is -30°C .

2. Heating power requirements for antifreeze and deicing are required: The open deck, helicopter deck, access, and staircase are not less than 300 W/M^2 ; the superstructure is not less than 200 W/M^2 ; the handrail is not less than 50 W/M^2 .

Antifreeze Design

The icing on the polar offshore platforms is mainly caused by seawater splashing. The effective ice protection and deicing design can significantly reduce the ice coverage. Polar antifreeze technology mainly includes five aspects: platform structure design, regional protection, selection of outdoor equipment, heat supply system heating, and coating protection. Polar offshore platforms can be designed frostproof from these five aspects:

Platform Structure Design

For the offshore platform, it is possible to design the pile foundation of the platform into large diameter and to make the bottom of the deck more flat to increase the possibility of the ice layer to fall off through its own gravity, in order to effectively reduce the amount of water splashing ice. Increasing the design of smooth surface and vertical surface and reducing the number of small components can reduce the mechanical interlocking of icing and equipment and improve the convenience of deicing and the possibility of icing and falling off.

Antifreeze Design in Open Area

The antifreeze design in the open area mainly includes the layout of the windbreak wall, the calculation of the heat load in the antifreeze room, the design of the electric tracing system, and the piping layout.

When the offshore platform works in the harsh environment, the deck platform is easy to freeze. It will not only affect the normal work of the equipment but also affect the operation of the crew, so the installation of the protective wall can greatly improve the working conditions. Considering the function of the equipment, the

platform can install a windbreak for some parts of the open deck machinery. The main equipment for installing the windproof wall is as follows: installation of windproof walls around the main deck, installation of windbreak storehouse in drilling pipe/drilling riser area, lifeboat/salvage yacht installation windproof wall, installation of windproof wall near the bow and stern lifesaving valve, and so on.

The crew sometimes needs to work in a room in an open area (such as a winch space). In order to operate the normal equipment in a relatively comfortable environment, a space heater is required to be installed in the room to prevent freezing. It can be used as an electric heater and calculates the power of the antifreeze space heater.

The escape channel is an important safety setting. In order to ensure effective identification and use of escape passages under severe cold, the activities and evacuation of personnel are not affected by the ice. According to DNV winterization basic, the escape passage in the open area belongs to the I type antifreeze area, which requires that the area can't have ice accumulation. Heat tracing is usually used to achieve the above purpose. Electric tracing has the advantages of simple installation, uniform heat dissipation, accurate temperature, high reliability, long service life, and energy saving (Guo and Li 2002).

In the layout of polar offshore platforms, the influence of low-temperature weather on the piping system should be considered in order to avoid the occurrence of liquid icing in the piping system. In carrying out piping layout, we should pay attention to the following aspects (Yang 2015):

1. As far as piping system is concerned, try to avoid direct exposure to outdoor lines below the waterline.
2. Discharge pipes shall be installed on icing pipes on the open deck and above the waterline.
3. It is necessary to increase the diameter of the drainage pipeline in the open area to avoid blockage.
4. For water tanks above the waterline, heating pipes should be added to prevent freezing.

Interior Antifreeze Design

The objective of interior antifreeze design for polar offshore platforms is to ensure that under harsh weather conditions, the living conditions of the platform personnel and the normal operation of the equipment are satisfied. The indoor antifreeze of the platform is mainly realized by heating, ventilation, and air conditioning system, and the three are collectively referred to as HVAC system.

The heating system of polar offshore platform can be divided into two parts: mechanical area and living area. Because the air conditioning in residential areas has the function of heating, the heating of living areas is a part of the air conditioning system. The main purpose of mechanical zone heating is to protect equipment and pipelines in low-temperature environment, and the way to use them is to use space heaters. The main function of the platform air conditioning system is to make the air to the required state, that is, the air temperature, humidity, air flow speed, and cleanliness are manually adjusted to meet the requirements of human comfort and process production (Yang 2015).

Design of Heat Source System

The heat source system is mainly used for the heating of fuel, oil, and other kinds of liquid needed by the power plant and provides the heat source for the heating equipment such as the cabin air conditioner and the heater. Therefore, the heat source system is one of the important systems of polar offshore platforms. The heat source system can be divided into four types according to the working fluids: steam heating system, hot water heating system, hot oil system, and electric heating system.

Steam heating has a simple and cheap working substance (water) and is widely used in heating occasions when experience is mature. There are three main types of steam users: room heating, cabin heating, and equipment heating. The hot water in the hot water heating system has a low temperature and a large flow rate. If the heating medium of the heating coil of the air conditioner is changed into hot water, the steady flow of water can be kept in the heating coil, and the danger of

the freezing cracking of the heating coil can be reduced. In the actual use of hot water heating systems, antifreeze may be added to hot water (Yang 2015).

Coating Protection

The special surface materials on the bare surface of polar offshore platforms can also effectively prevent icing, such as sacrificial coatings, superhydrophobic coatings, and water-lubricated coatings.

The sacrificial coating can release some small molecules that inhibit the crystallization of supercooled water irreversible and increase the lubricity of ice and material surface, reduce the adhesion strength of ice, and even make the icing naturally fall off (Ayres et al. 2007).

The so-called superhydrophobic coating is the coating with water contact angle greater than 150° and the rolling angle less than 10° . The coating has the dual properties of the surface and the multilevel roughness of the low surface energy material, which can reduce the contact area of water or ice to the material surface, enhance the stress concentration of the ice and the material interface, and reduce the adhesion shear strength of the ice. So as to achieve the purpose of delaying icing, and after freezing, the ice adhering to the surface of the coating can be removed by very small force or wind. Meuler et al. have studied the relationship between the ice adhesion strength of a series of materials and the water back contact angle θ_{rec} . It is found that the ice adhesion strength is strongly correlated with $1 + \cos\theta_{\text{rec}}$. The results show that increasing the water back contact angle of the coating helps to reduce the ice adhesion strength and has guiding significance for the design of superhydrophobic anti-icing coating (Meuler et al. 2010).

Dou et al. (2014) graft hydrophilic dimethyl hydroxypropionic acid (DMPA) onto polyurethane (PU) substrate to make the coating hydrophilic. The contact angle is only 43° . The mechanism is to separate the ice and the coating

surface by absorbing moisture. At -15°C , the adhesion strength of ice can be reduced to below 30 kPa.

Cross-References

- [Ice Breaking Vessel](#)
- [Navigation of Polar Vessel](#)
- [Polar Materials](#)
- [Structural Characteristics of Polar Engineering](#)

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Zero-Tension Interface

► [Shallow Foundations](#)